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Coal-Fueled Diesel Systems Research
Volume II
Engine Operation, Durability and Economic Studies

Final Report

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COAL FUELED DIESEL SYSTEMS RESEARCH - FINAL REPORT

EXECUTIVE SUMMARY

This report is presented in two volumes:
Volume I - Basic Fuel, Fuel System and Combustion Studies
Volume II - Engine Operation, Durability and Economic Studies

The work described in this report was performed by the Electro-Motive Division of General Motors Corporation (EMD), LaGrange, Illinois, Southwest Research Institute (SwRI), San Antonio, Texas, and Adiabatics, Inc., Columbus, Indiana, as subcontractors to Allison Gas Turbine Division of General Motors Corporation (Allison), Indianapolis, Indiana. Allison Gas Turbine Division served primarily in administrative and management capacity. The overall program was supported by the U.S. Department of Energy, Morgantown Energy Technology Center (METC), under contract No. DEAC21-85MC22123.

The Southwest Research Institute project team has been involved in combustion of coal in diesel engines since 1975. Much of the early work involved the use of coal slurries in diesel fuel and the use of coal powders. The coal/water slurry work was started in 1982-83 as a contract with the Bartlesville Energy Technology organization to provide technical assistance in the use of coal-water slurries, including the design and construction of a combustion bomb device at SwRI. In 1984, SwRI was contracted by METC to determine the effects of elevated combustion chamber temperatures on the ignition and combustion of coal/water slurries.

In preparation for the work described in this report, SwRI summarized the work of the above noted studies as well as the findings of other related earlier works to define the technology gaps in utilization of coal in diesel engines. The results of this literature assessment are presented in the Background Section of Volume I of this report. It became clear that the primary problem areas were fuel specification, fuel handling and storage, fuel injection, ignition and combustion, and engine and component wear. The goal of this project was therefore defined to be one of technology assessment in which each of the problem area was to be examined experimentally in order to define the best approach for system development. The goal did not include engineering development.

Electro-Motive Division , as a major world locomotive manufacturer, will be an ultimate user of any successful coal-fueled diesel technology developed. EMD provided medium speed (900 RPM) diesel engine expertise, hardware, economic investigations and evaluations. EMD also conducted test and evaluation programs with full cylinder size coal-fueled locomotive diesel research engines to determine operating and wear characteristics.

OBJECTIVE

The overall objective of this program was to investigate the economic and technical features for a coal-fueled diesel locomotive system. Commercialization merits of coal-fueled locomotives were examined. Bench scale screening tests and full scale verification tests were conducted to determine the effects of selected coal fuels on fuel specification, handling, injection, ignition/combustion, and engine wear. Technology deficits would be noted for recommended future evaluation.

APPROACH

The successful application of coal fuels in medium speed diesel engines requires technical development in five different areas. Namely:

1. Fuel specification
2. On-board fuel handling
3. Fuel injection and atomization
4. Engine combustion
5. Engine wear prevention

As applied to a true locomotive installation, all of these areas are related. The approach which SwRI used in this program involved the use of bench experiments (injection system stands and a constant-volume combustion apparatus (CVCA), where possible, to simplify the various programs to the basic elements. The various program areas were addressed in the simplest possible experiments in order to develop a technically sound data base. Because of the cost and complexity of running full-scale engine experiments (medium-speed EMD), a smaller engine, manufactured by the then Detroit Diesel Allison Division of General Motors (DDA)(now Detroit Diesel Corporation) which is identical in design to the full-scale EMD engine was procured. This engine was used in conjunction with the bench experiments to develop the technology required to perform successful full-scale engine tests. The overall approach involved the use of progressively more complex experiments, culminating in the last months of the program in experiments using a two-cylinder version of the full-scale EMD engine.

EMD, using the technology developed by SwRI early in the program, continued evaluations of the technology items noted above in a full cylinder size research engine as well as in a full scale 8-cylinder engine. Important new features were utilized and evaluated in the EMD effort:

- CWS storage system for long term storage.

- Systems for diesel fuel pilot charge, engine inlet air heating, and increased compression ratio (24:1) were integrated for enhanced CWS ignition and combustion.

- Incompatibility of mixing diesel fuel and CWS was eliminated by means of a system for controlled transition of injected fuels from No. 2 diesel fuel, to water, to CWS.

- A CWS fuel injection system was designed, fabricated and operated which is based upon using individual injection pumps and injector nozzles for both pilot fuel (No. 2 Diesel) and CWS injection. The CWS injector capacity is about two magnitudes greater than presently used unit injectors in order to attain CWS burning for equivalent levels for horsepower. The timing and fuel amount for both pilot and CWS injection are separately and independently controlled.

Economic viability of a coal-fueled diesel locomotive for the time period 1995-2010 was evaluated. The criteria used for determining the economic viability is the internal rate of return on investment (IRR). The coal-fueled diesel locomotive is compared to a baseline petroleum-fueled diesel. The coal-fueled diesel locomotive is still very much conceptual, therefore, many assumptions were made with regard to cost, operating procedure, fuel form, fuel cost, maintenance, etc. A spreadsheet calculation procedure was used for determining the IRR.

Late in the program six alternate coal-fueled locomotives are also evaluated with respect to economic viability. The evaluation procedure for the alternate coal-fueled locomotives is substantially the same as that for the coal-fueled diesel locomotive. The six alternate coal-fueled types evaluated are:

Steam turbine electric
Coal gasifier-spark ignition-electric
Coal gasifier-diesel electric
Fluidized bed steam reciprocating-electric
Coal burning diesel electric
Coal gas turbine-electric

Engine durability concerns were approached by Adiabatics, Inc. in a program which evaluated wear characteristics of a number of different candidate wear pair combinations. The wear determinations were conducted in a unique friction and wear apparatus which subjected the test specimens to reciprocating and rotary motion, with preload, in a coal combustion products atmosphere.

SUMMARY AND CONCLUSIONS

The technology assessment at SwRI was aimed at all of the problem areas, from fuel specification to engine design. The goal was not to design and demonstrate a system but rather to evaluate current hardware and technology and to make recommendations regarding the best approach for future development.

The result of the various laboratory and engine experiments are in general agreement with the results of similar experiments performed at other laboratories. The project team at SwRI disagrees with the approach which is being taken by the other research teams regarding the need for, and the unproven benefits of going to high-pressure fuel injection. As will become apparent from the conclusions, the results of the experiments performed at SwRI lead to the general conclusion that the best design approach for a coal-fueled diesel involves the use of low-pressure fuel injection into an indirect-injection engine.

The conclusions are divided according to the various problem areas which were examined in this project.

FUELS

Three different carrier liquids were used to formulate stable, low-viscosity coal slurries. The composition and the properties of these slurries have been thoroughly documented in this report. The experiences gained in preparing the evaluating these fuels have resulted in the development of several conclusions. These are as follows:

1. There are no generally accepted specifications for the coal slurries. The adoption of such a specification is essential to the development of a fuel of uniform quality which can serve as a guide to the fuel producers, and which can be used as the reference fuel by the engine designers.

2. Stable, low-viscosity slurries can be formulated, but they are non-Newtonian and generally tend to shear thin at shear rates up to 1000 sec-1, and may shear thicken beyond 1000 sec-1.

3. The rheological properties must be controlled, or at least characterized, at different shear rates. Injection performance is affected by the apparent viscosities at shear rates exceeding 10,000 sec-1. The flow properties in the handling system are probably best characterized in the range of 500-1500 sec-1. The storage stability is indicated by the apparent viscosity at extremely low shear rates.

FUEL STORAGE AND HANDLING

High shear rate pumping of the slurries will likely result in reduction of the particle size (breaking of agglomerates) and corresponding changes in the rheological properties. It is very likely that most fuel handling systems will incorporate one or more fuel pumps for transporting the slurry from one location to another. These systems are also likely to include some form of storage vessel. The two main problems in designing these systems have been pump selection and settling during storage.

1. Selection of reliable, low-shear rate pumps are an important aspect of designing slurry fuel handling and storage facilities. It appears that progressive cavity pumps meet these requirements.

2. The overall problem of long-term storage stability may be solved by the use of re-circulation storage tank systems like those developed in this project.

FUEL INJECTION

The problems associated with the injection of the slurries have historically included reliability of operation and achievement of the appropriate injection characteristics. All of the designs currently being examined and extolled as solutions to the slurry fuel injection problems have been evaluated previously for use with diesel fuel. All of these systems have been dropped from consideration by the industry for reasons which are related to one or more of the inherent operating characteristics of the systems. These undesirable traits are not likely to be mitigated by the use of slurries. The majority of the diesel engine industry has adapted the use of some form of pump-line-nozzle system.

Unit injectors can also be considered in this family, where the line length is extremely small in the case of the unit injectors. With this introduction, the conclusions which were drawn from the injection and atomization studies are as follows:

1. Reliable operation and acceptable performance could only be achieved in this program at SwRI using modified pump-line-nozzle systems. Two-fluid, diaphragm separated systems and pressure-time systems were examined experimentally in this program. Air assist designs were examined in a previous program at SwRI. Accumulator, common rail, and fluid intensifier system designs were examined analytically in the program. Each system was deemed to be unacceptable for further development because of inherent technical flaws or because the performance was not good enough to warrant further development.

2. Significant erosion of the nozzle holes was observed in the pressure time system. The problem was addressed using tungsten-carbide inserts in the tip. The manufacturing problems were successfully overcome. Selection of the appropriate formulation will be possibly only after the durability characteristics are assessed. Pintle nozzles were used in the CVCB experiments and demonstrated no erosion type wear problems.

3. The pump-line-nozzle systems which were successfully adapted to operation on coal all incorporated the following features which are essential for acceptable operation:

   a. Nozzle cooling
   b. Elimination of dead volumes
   c. High fuel supply pressure to the jerk pump
   d. Increased clearance and leakage rates.

4. The pump-line-nozzle systems offer the most promise for commercial development because all of the other approaches involve one or more inherent characteristic which are not acceptable for use in modern high-output, emission constrained diesel engines. Adaptation of existing hardware involved some tradeoffs which are acceptable for commercial systems, but which can be easily eliminated in new design.

5. A dry-powder coal injection system has been designed and successfully demonstrated in an operating diesel engine. The laboratory system which was built involved several mechanical components which proved to be unreliable, but which could be easily replaced with more reliable electrical or pneumatic equivalents.

6. The use of the slurries results in an increase in the spray tip velocity and a decrease in the spray cone angle as compared to equivalent operation on diesel fuel.

7. The higher apparent viscosities of the slurries results in increases in the injection pressures and corresponding increases in the mechanical loadings on the injection system components. Historically, engine injection system designers have attempted to minimize the injection pressures and thus keep the component loading as low as possible. Current trends to increase the injection pressures for the slurries may not be compatible with the limitations of commercially viable systems.

8. The optimum injection characteristic, in terms of the tip velocity, the cone angle and the drop-size distribution, have to be defined for the slurries. The performance of current diesel fuel systems may be a good target, assuming that the ignition and combustion characteristics of the slurred are the same as those of diesel fuel.

IGNITION AND COMBUSTION

The ignition and combustion characteristic of three different coal slurries were evaluated in a CVCA and in two different 2-stroke diesel engines. The results of these experiments demonstrated the differences in the ignition and combustion characteristics of the slurries as compared to diesel fuel. In addition, the experiments pointed out the limitations imposed by attempting to convert direct injection engines to operation on coal. The conclusions drawn from these experiments are summarized as follows:

1. The ignition delay time of the slurries are much longer than those of diesel fuel under identical conditions. This indicates that the slurries have extremely low cetane numbers and will all, including those formulated with diesel fuel, methanol, or water, require some form of ignition assist.

2. The methanol and diesel fuel slurries would autoignite in the engine under some operating conditions. The coal/water slurry required a pilot injections of diesel fuel under all operating conditions.

3. The optimum pilot injection timing was several crank angle degrees prior to the slurry injections. This indicated that the pilot was serving to increase the gas temperature at the time of slurry injection rather than serving as a direct ignition source for the slurries.

4. The combustion and thermal efficiencies were higher in the larger bore EMD engine than in the smaller DDA engine. This indicates that the larger heat transfer paths and the smaller surface to volume ratio result in higher temperatures. The gas temperatures in both engines were still too low, even with more than acceptable levels of pilot fueling. Additional ignition enhancement and/or higher gas temperatures are required for acceptable performance.

5. Preliminary experiments using glow plugs in the CVCA indicated that glow plugs might provide an ignition source which could compliment the use of higher gas temperatures. Glow plug durability would be of concern if this approach were pursued.

6. The data indicates that direct-injection engine designs may not be the best selection for conversion to coal fueling because:

   a. Coal particles come in direct contact with the lubricant-wetted cylinder wall.

   b. The injection system requirements for direct injection are too severe for operation on coal.

   c. It is not currently possible to increase the surface temperatures to the required values.

   d. The compression ratio cannot be raised to the required level in direct-injection engines.

   These are all limitations which must be overcome in a successful coal burning engine design. An indirect-injection design might overcome these problems.

7. A new indirect-injection engine design was developed, built and tested in the DDA engine. In this design the combustion chamber is formed in the piston and is separated from the main combustion chamber by a plug which protrudes from the head into the chamber. The engine was operated in this configuration for a short period of time in this project.

8. The new indirect-injection design was incorporated in the current EMD design. Implementation of the design in the EMD engine involved tradeoffs in the volume and the shape of the pre-chamber which were significant enough to compromise the basic approach. Proper implementation in the EMD engine would involve piston and head design changes which were beyond the scope of this project.

FULL SCALE ENGINE

The full scale engine operation at EMD is summarized as follows:

The single pump CWS fuel injection system as suggested from the SwRI work, did not perform with for any meaningful duration. EMD has determined that additional development is needed for the single pump system in order to attain acceptable service life.

The back-up injector system which is a modified EMD cam-driven unit injector apparatus performed 3.5 hours before having an operational problem. Further study is needed to increase the operational life of the injector. The following are important results from the 3.5 hours of operation:

- The combustibility condition of the CWS can be estimated from water injection into the cylinder.

- It is easier to ignite and burn CWS in a 2-cycle diesel engine than a 4-cycle diesel engine due to the high thermal loading of the 2-cycle engine operation.

- The effects of these parameters on the combustion of CWS are studied in detail.

  - Pilot fuel amount
  - Pilot fuel injection timing
  - CWS fuel amount

  - CWS fuel injection timing
  - Engine speed
  - Inlet air heating

- Higher combustion efficiency occurs with CWS injection timed near top dead center and pilot injection starting at about 28 degrees crank angle before top dead center. The timing of the pilot fuel is not so sensitive for the ignitability of the CWS and for a smaller range (about 12 degrees crank angle) for the combustion efficiency.

- The cylinder exhaust temperature can be a good indicator to monitor the combustion of CWS.

- The pilot fuel amount could be reduced to less than 5% of the total fuel energy at the maximum CWS fuel amounts possible with this injector system.

- Only one injector failure was observed after 3.5 hours of CWS operation accumulated in four days of operation. The failure was attributed to seizing of the injector piston as a result of CWS jamming in the injector/piston clearance. There was no visual evidence of injector tip wear and the injection characteristics of the unit did not change during the 3.5 hour of operation.

- The combustion of the CWS is not considered as a major problem. Attention should be focused on increasing the combustion efficiency.

- The CWS storage tank operation was monitored for ten months of continuous operation. The present storage tank system can keep the CWS in a mixed and usable condition for up to six months. Some change in the CWS concentration distribution within the tank was observed after six months of operation. Although the same CWS was used beyond this time limit, it is not known if the fuel characteristics were permanently altered.

- It is demonstrated that the concept of injecting both No. 2 diesel fuel and CWS from the same injector is feasible. The transient time in switching from water to CWS should be further reduced.

The piston ring and liner wear tests were not performed due to problems in acquiring reliably coated components from the suppliers within the timetable of the project. Further work is also required to determine if the high pressure single pump CWS injection system concept can be made feasible. As Electro-Motive did not perform tests with low pressure CWS injection systems, information is not offered regarding preference of one system over the other.

EXHAUST EMISSIONS

The exhaust emissions were measured during operation of the EMD engine on coal/water slurry. In addition, a study was performed to project the impacts of coal-fuel operation on the emissions of locomotive systems. The conclusions drawn from this work are presented as follows:

1. Coal-fuel operation of the EMD engine resulted in reductions in the CO and NO_x emissions and significant increases in the particulate emissions as compared operation on diesel fuel.

2. If the ash and the sulfur contents of the coal are 1% and no after-treatment devices are used, the particulate emissions would be an order of magnitude larger than with diesel fuel, and the sulfur emissions from locomotives would become a significant fraction of the total mobile source emission.

3. Several different approaches are possible for reduction of the particulate emissions from the mobile locomotive sources. Control of the sulfur emissions from mobile sources is currently not practical, although several different technologies are available for consideration.

4. Treatment of the exhaust gases prior to the turbo charger (hot gas clean-up) does not appear to be practical. Filters, scrubbers, lime injection and filtration, and all other approaches cause temperature and pressure drops which would seriously compromise turbo charger performance.

ECONOMICS

The economic viability of a coal fueled diesel locomotive is dependent on a number of considerations with respect to initial costs, fuel costs, maintenance, infrastructure costs and other similar factors. The conceptual nature of the coal-fueled diesel locomotive makes the identification of the costs of these factors somewhat arbitrary. However, based on reasonable assumptions for the factors involved, including a diesel fuel price of $5.86/million Btu and a coal-fuel price of $4.00/million Btu, the following conclusions were drawn:

- o Using 50/50 coal-water slurry, a sensitivity to wear rates shows that cylinder life must be two years to attain a 21.9% IRR.

- o If acquisition cost can be held to no more than 15% above that of the conventional diesel-electric, then the IRR would be 30.2%.

- o Efficiency with coal is assumed to be as good as, if not etter than, that with DF2. However, if efficiency were to drop by 10%, the IRR would be 15.3%.

- o Using coal slurried with methanol of DF2 produces a low or negative IRR. The worst case is a 50/50 coal-methanol slurry, which produces an IRR of -37.5%.

- Dry powdered coal is expected to be more difficult to handle than coal slurry fuel. However, the price advantage should make further study of this fuel attractive. The IRR for coal powder fuel is projected to be 39.3%.

The economic evaluations of the six alternate coal-fueled locomotives were based on information available from the proponents of the particular concepts and in the literature. The coal fuel price assumed for these studies was assumed as $1.52/MMBtu (Million Btu) for the concepts requiring untreated coal and $4.40/MMBtu for the concepts requiring treated (cleaned) coal. (Subsequent to the time the report was being prepared there is information that indicates the price of treated coal may well be substantially less than $4.40/MMBtu).

Based on the original assumptions, none of the concepts has an IRR that appears attractive. The gasifier locomotive with a diesel engine has the highest IRR at 8.6%, but this is short of its 30% hurdle rate goal.

If the price of diesel fuel were to increase from the $1.52/MMBtu used to $3.17/MMBtu or conversely, coal fuel were to drop in price significantly (as now may be the case) some of the concepts would look more attractive. In this situation:

- The steam turbine IRR of 36% surpasses its hurdle rate of 25%
- The gasifier/diesel IRR of 46.2% surpasses its hurdle rate of 30%
- The coal fueled diesel IRR of 41% surpasses its hurdle rate of 40%
- The coal fired gas turbine IRR of 38% approaches its hurdle rate of 40%

1.0 INTRODUCTION

On June 13, 1984, the U.S. Department of Energy, Morgantown Energy Technology Center issued a Request for Proposal (RFP) entitled, "Coal-Fueled Diesel Systems Research" (RFP No. DE-RP21-84MC21197). A research team, involving three different organizations, was formed to respond to the RFP. The lead role was assumed by Allison Gas Turbine Division of General Motors. Electro-Movtive Division of General Motors and Southwest Research Institute (SwRI) were subcontractors with the primary technical responsibility in the proposed effort. A proposal was prepared and submitted on August 3, 1984. Three contracts were awarded in March, 1985 as a result of the RFP. One contract was awarded to the General Motors team, one to a team consisting of A.D. Little and Cooper Industries, and the third contract was awarded to a team from General Electric.

The purpose for the requested research was to develop an environmentally sound integrated heat engine system which would produce a cost competitive energy using coal as the fuel. The RFP package included a work statement in which three specific tasks were identified. The first task involved an economic assessment aimed primarily at determining the economic boundaries for utilizing systems. The second task was to deal with the technical questions related to the use of coal in mobile locomotive systems. The objective of this task was to determine the effects of the coal on diesel engine combustion, to identify areas where technology developments were required, and to develop recommendations for further work. New technical data developed during the experimental work was to be used in the third task, which was to consist primarily of updating the economic assessment in an effort to determine the technical feasibility and the economic merits of direct utilization of coal in diesel engines.

The division of responsibility within the General Motors team consisted of Allison Gas Turbine as the prime contractor with overall administrative and technical responsibility in the completion of the economic studies. Originally, all of the experimental work was to be performed in the laboratories of SwRI. Electro-Motive Division was to provide technical and engineering support for the SwRI efforts and had primary responsibility for preparation of the spread sheets for the economic analyses. Both SwRI and EMD were responsible for providing data for inclusion in the economic analyses. A contract modification, approved in 1986, extended the contract to include significant expansion of the economic studies, completion of coal combustion studies in a specially designed single cylinder engine at EMD, and completion of a coal fueled diesel engine emissions study at SwRI. An experimental investigation to determine the friction and wear (tribological) characteristics of a selected group of candidate piston ring and cylinder liner materials suitable for a coal burning EMD 2 cycle engine was included in the contract modification. Adiabatics, Inc. of Columbus, Indiana was the subcontractor selected to undertake this task.

The work at Electro-Motive Division (EMD) as well as that of Adiabatics, Inc. is reported in Volume II. Engine component durability concerns, economic considerations of alternate coal-fueled diesel locomotives and coal-fueled operation of a EMD 2-645 research engine are addressed in Vol II of this report.

1.1 BACKGROUND

This report covers the work performed for the Allison Gas Turbine Division of General Motors (prime contractor) at the engineering and test facilities of the Electro-Motive Division of General Motors located in LaGrange, Illinois. This work was planned as a continuum of earlier research work for coal-fueled diesel engines utilizing the results from work performed by the Southwest Research Institute and Adiabatics, Inc. The work performed at Electro-Motive represents the final engine research oriented tasks to be performed for the Coal-fueled diesels: Systems Research endeavor.

1.2 PROBLEM STATEMENT

The overall goal of this work was to set up a test facility for a coal-fueled research diesel engine, based upon the best knowledge derived from the earlier project work, and implement a test program which would provide the most probable capability to run a diesel engine on coal water slurry (CWS) for at least ten cumulative hours of operation. Once achieving this capability, the relative wear characteristics of special coated piston rings and liners would be determined after having operated within this coal-fueled diesel combustion chamber environment for 10 hr of operating time. Decisions in planning this test facility and program were adjudicated according to the anticipated needs for the application of a CWS fueled diesel engine within the locomotive physical and environmental constraints. Such adjudication was to provide the most reasonable facilities and approach within the Statement of Work of the project. Results from this type of an approach were expected to yield important data and insight for determining the relative merit and justification for eventual technology development and commercialization.

1.3 OBJECTIVE

The objective of the work reported in this volume is threefold:

1. Evaluate the mechanical and thermodynamics operation of an EMD diesel engine with coal-water slurry as a fuel.
2. Evaluate durability of selected wear and corrosion treated components in a coal-water slurry fueled EMD engine.
3. Examine the economic viability of a coal-fueled diesel locomotive and alternate coal-fueled locomotive concepts as compared to the conventional petroleum fueled diesel locomotive.

1.4 TECHNICAL APPROACH

It is recognized that the ability to run a test for determining the wear characteristics for wear resistant coatings on piston rings and liners would be meaningless without first determining the compatibility of such coatings with the base metal of these components. Results from the work by Adiabatics, Inc. was used in determining the best candidate coatings. However, previous EMO experience with ceramic type coatings and knowledge of the importance to have materials with similar thermal and mechanical properties was used in making the selection of the final piston ring and liner wear surface coatings.

It was recognized that the most meaningful wear test which could be performed would be one in which the greatest amount of coal water slurry fuel could be injected and burned. This defined the approach to find a way in which we might be able to inject a CWS amount greater than the volume attainable with standard (DF2) injectors and determine the optimum combustion relationships (and engine equipment) for burning this amount of CWS with the least amount of DF2 pilot charge or, perhaps, no pilot charge at all. Calculations were made which subsequently determined the design of engine components and a test facility which would allow the greatest flexibility for controlling injection timing and fuel quantity for both Main-CWS and Pilot-DF2 fuels. In order to increase the thermal requirements for CWS combustion, provisions were made for heating of the engine inlet air as well as fabricating engine components to operate up to a 24:1 compression ratio.

2.0 COAL WATER SLURRY

2.1 CWS FUEL SUPPLY

The CWS fuel used at the EMD tests was obtained from SwRI. SwRI ultimately evaluated 13 different fuels in that program. The details of coal specifications (Ref 43), the companies for the request for bids, the properties of the coal purchased and the process for the prepapation of the coal slurries are described in Volume I in detail. Table 1 shows the properties of the coal used in making the slurry. This coal was a high volatile, bituminous and Pennsylvanian in age. It was from the Wellmore No. 14 mine of the Buchanan County of South Virginia, taken from the seams from Hagy to Clintwood. It was cleaned and ground to ultrafine powder form by AMAX Extractive Research and Development Company.

This coal powder obtained from AMAX was reground to smaller size, to the particle size distribution given on Table 2, and then the slurry was prepared by SwRI. The details of the slurry preparation can also be found in (1). The particle size distribution of this AMAX slurry is shown in Fig. 1. This slurry was used at SwRI tests.

Table 1.
Coal Properties

Property	Method	Amount
Proximate Analysis, wt%		
Moisture	ASTM D 3173	0.35
Ash	ASTM D 3174	1.00
Volatile Matter	ASTM D 3175	34.51
Fixed Carbon	ASTM D 3172	64.14
Ultimate Analysis, wt%*		
Carbon	ASTM D 3178	83.85
Hydrogen	ASTM D 3178	5.51
Sulfur	X-Ray Fluorescence	0.87
Nitrogen	Chemiluminescence	3.21
Oxygen (by difference)	ASTM D 3286	6.10
Heating Value		
Gross, MJ/kg (BTU/lb)		34.6 (14,905)
Net, MJ/kg (BTU/lb)		33.4 (14.390)
Particle Size Distribution	Photometry	
Log-Mean length, microns		10.2
Top particle size**		28
Standard deviation		1.85

* Moisture and ash free basis
** 95% of the volume has a smaller particle size

At the beginning of the EMD tests it was decided that the same slurry prepared and used by SwRI should also be used at EMD in order to maintain the EMD test schedule and to eliminate a possible additional parameter in comparing the results at EMD against the tests at SwRI. Therefore the same slurry prepared by SwRI is transported to EMD, after some reblending, and used at EMD tests.

2.2 SLURRY CHARACTERIZATION

To understand the shelf life of the slurry and the effects of the storage time on the slurry characteristics, a slurry monitoring plan was activated. This plan consists of two separate activities. First, the detailed characterization of the slurry at the beginning of the storage, and at the beginning and end of the wear tests were foreseen. For comparative purposes, these tests were to

Table 2.
Particle Size Distribution

Coal Grind	As Received (AR)	Ground Once (1X)	Ground Twice (2X)
Log-Mean length, microns	10.2	7.4	5.5
Top particle size, microns	28	18.5	12.5
Standard deviation	1.85	1.77	1.65

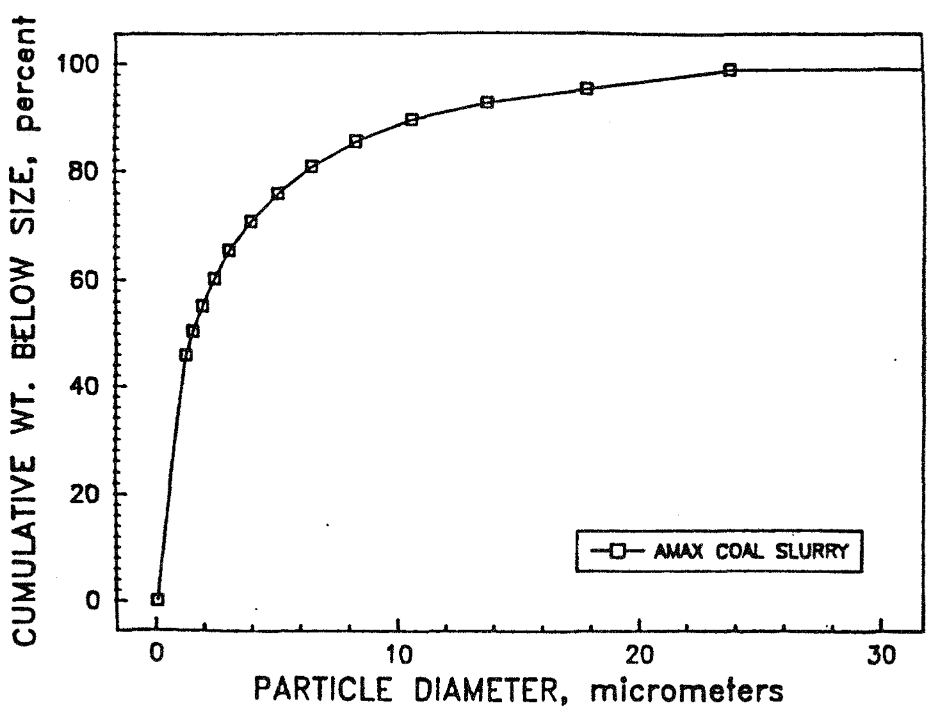

Figure 1. Coal particle size distribution in AMAX coal slurry.

be made by SwRI laboratories, using the same equipment and procedures as in previous characterizations. In these tests the slurry viscosity as well asthe particle size distributions are to be measured. Another group of tests were planned to monitor the slurry characteristics in the slurry storage tank at EMD which would measure the percent solids, percent ash, and the surface tension of the slurry sample taken from the circulation pump outlet of the slurry storage system at 15-day intervals.

Twelve drums of coal/water slurry were supplied to EMD. During the storage at SwRI, part of the coal had come out of suspension and had settled on the bottom of the drums. Each drum was reblended using the same procedure outlined in (114) and sampled. The composite sample from all drums was characterized with measurements of viscosity, surface tension and particle size.

On Fig. 2 the apparent slurry viscosity measurements are shown as a function of the shear rate for the blended slurry before shipment to EMD. The viscosity measurements were made by using different instruments and conditions to cover a shear range from 16 to 40,000 reciprocal seconds. Comparison of this figure against previous measurements at SwRI was reported to show that up to about 3000 reciprocal seconds, the viscosities were approximately the same. Above 3000 reciprocal seconds, the viscosity of the reblended slurry was lower than the original slurry.

The procedure for surface tension measurement requires removal of a ring from the surface of the liquid and when measuring it for the slurry, it was somewhat dependent on the rate of ring removal. Fast removal takes about one minute while slow removal takes about three minutes. The slurry surface tension measurement using slow removal was 73.42 dynes/cm and using fast removal was

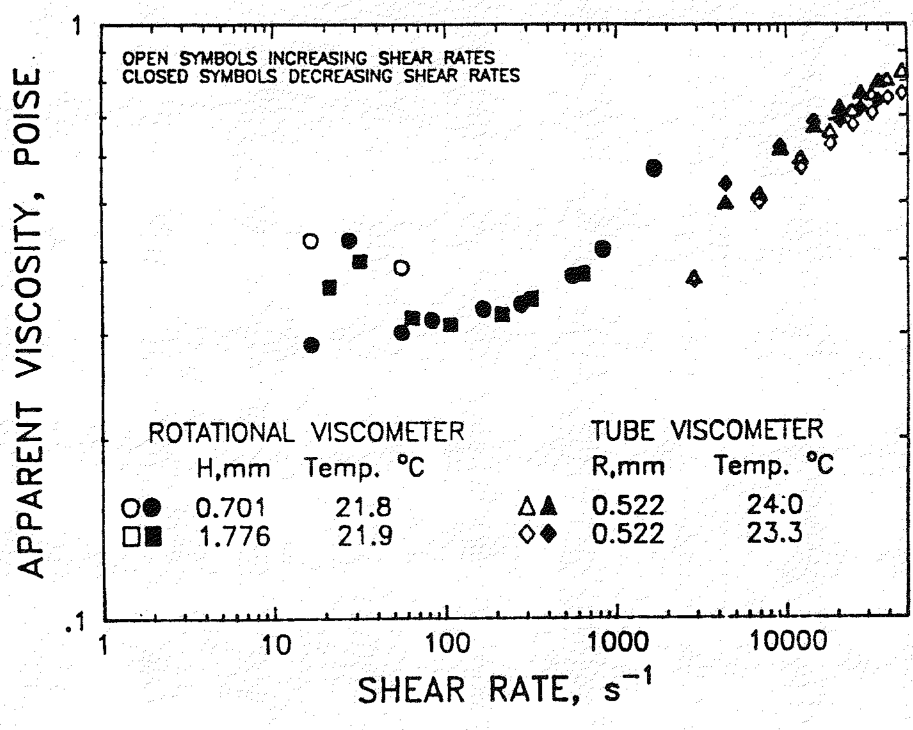

Figure 2. Apparent viscosity of coal slurry as a function of shear rate.

68.47 dynes/cm at SwRI. Hence the surface tension measurement of slurry is very sensitive to the details of the methodology. By comparison the surface tension of pure water at 20 Deg. C is 72.75 dynes/cm.

Particle size distribution of the reblended slurry was performed somewhat later than the above tests at SwRI. Samples of particles from the reblended slurries were photographed with the scanning electron microscope, and the particles were measured by using computer image analysis. The photographs showed that some particles had agglomerated. Since the agglomerated particles were easily distinguished from the single particles, their size distribution was measured separately and the results were combined to obtain the overall particle size distribution. The photographs and the data did not provide evidence regarding the cause of agglomeration; however, a separate sample of the slurry that had been kept in a continuously stirred tank since its arrival from AMAX to SwRI was available. The photographs of its particles also showed the presence of agglomerates. Therefore it was concluded that the agglomerates were not caused by settling and reblending at SwRI but was in the original coal. Bar charts of the size distributions for the single particles, agglomerated particles and for the combined distribution are shown on Figures 3, 4, and 5 respectively. The mean and top particle sizes are as follows:

	Particles Only	Agglomerates Only	Particles & Agglomerates
Log-mean Length, Microns	11	32	19
Top Particle Size	22	48	42
Standard Deviation	1.64	1.31	1.68

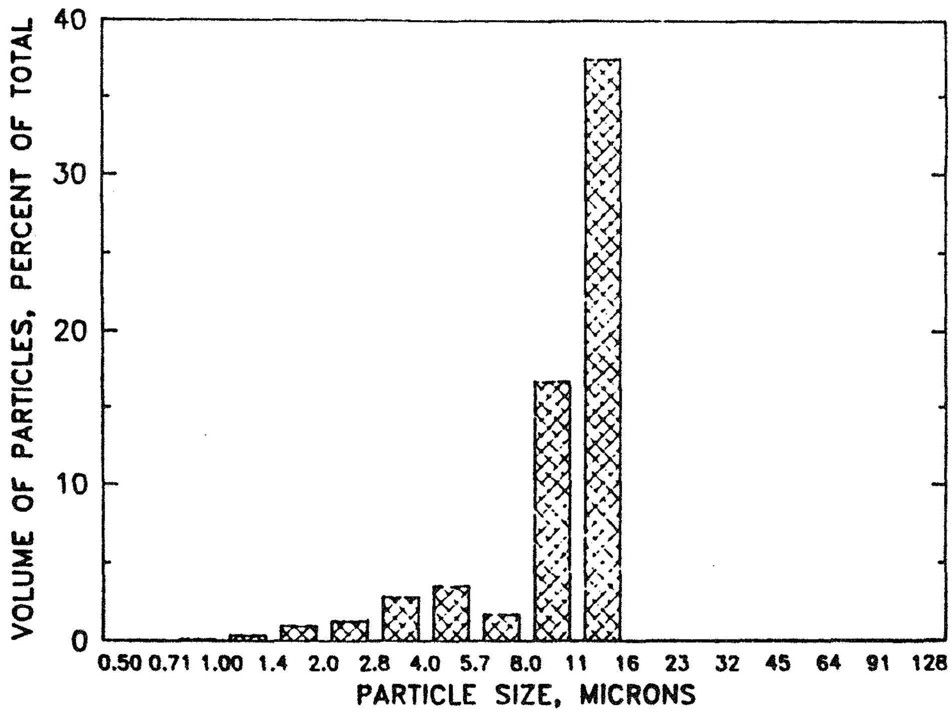

Figure 3. Particle size distribution in reblended slurry, single particles only.

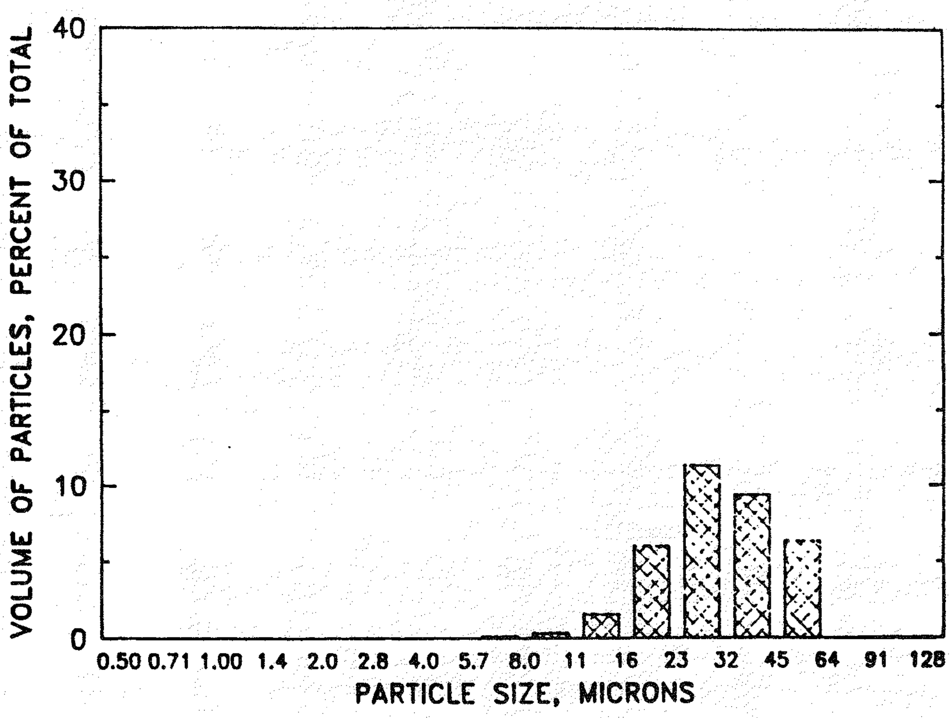

Figure 4. Particle size distribution in reblended slurry, agglomerated particles only.

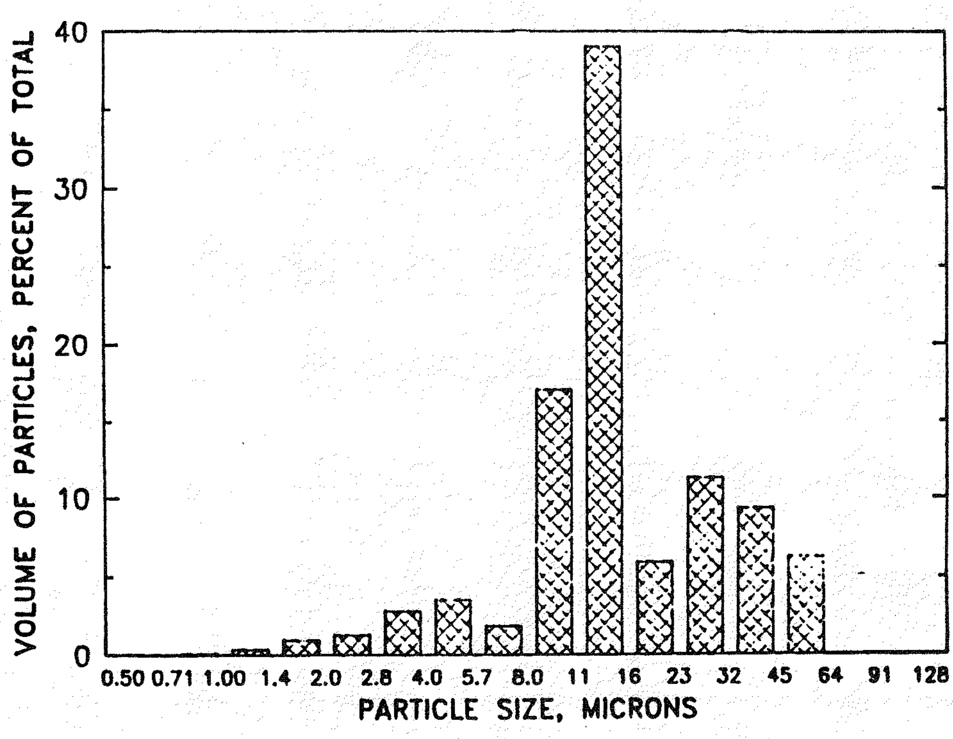

Figure 5. Particle size distribution in reblended slurry, all particles.

The combined distribution of the slurry, as shown in Fig. 5, is bimodal. The common, normal mechanism of particle agglomeration tend to shift the distribution toward larger sizes. It seemed more likely that the agglomerates were present in the coal before slurry preparation than they formed in the slurry afterwards. However the evidence was not conclusive.

The chemical composition of the slurry as transferred to EMD is reported to have the characteristics given on Tables 3 and 4 (17).

Table 3.
AMAX Coal/Water Slurry Fuel Characteristic

Coal Content of Fuel (wt%)			50.0
Ash Content of Fuel (wt%)			0.5
Coal Particle Top Size (micron)			15
Gross Heating Value of Fuel (BTU/lb)			7500
Viscosity of Fuel @ 100 sec-1 (cP)			40
Proximate Analysis of Coal (wt%)			
Ash			1.01
Volatiles			34.51
Fixed Carbon			64.48
Ultimate Analysis of Coal (wt%)			
Carbon			85.42
Oxygen			9.20
Hydrogen			3.22
Nitrogen			1.30
Chlorine			
			0.86
Ash Composition (wt%)	(ppm fuel basis)	Wt%	
SiO_2		1612	32.25
Al_2O_3		1315	26.30
CaO		340	6.80
MgO		58	1.15
Na_2O		213	4.25
K_2O		70	1.44
TiO_2		84	1.68
MnO_2		1	0.02
P_2O_5		12	0.23
V_2O_5		3	0.06
PbO		1	0.02

2.3 CWS STORAGE

Table 4.
AMAX Slurry Composition

Coal	50.00%
Stabilizer (Xanthan Gum)	0.03%
Surfactant (Nonionic)	1.50%
Preservative (Formaldehyde)	1000 PPM
Water	(Balance)

At the beginning of the EMD tests, it was decided to have one CWS batch prepared and maintained at the same condition so that the same fuel is used throughout the tests. This would eliminate the variation of fuel properties from one test to another as well as it would develop technology for storing the slurry on longer time frames. The CWS fuel requirement of the tests at EMD was estimated to be about 500 gallons, initially. Among various fuel storage tank possibilities, a horizontal tank configuration was preferred, as it would be a typical application for railroad tank cars, to be used as a tender to supply fuel to about 3 locomotive units.

The design development of the slurry storage tank was done by SwRI and described in (1) in detail. In summary, the design is guided to generate upward moving flow circulation pattern in the tank, by introducing the slurry from tank bottom, horizontally, and taking out from the top, vertically. The settling of coal particles are estimated from the gravitational and drag force considerations. The geometric parameters were optimized to eliminate the start-up of the secondary flow circulation pattern within the tank with downward flow velocities. The flow patterns in the tank were estimated by using the FLUENT code of CREARE (2) in dimensionless form of equations. Optimum flow pattern was obtained when the slurry flow rate was about 30 liter/min or the Reynolds Number, based on tank diameter, is about 1429. Application of this criteria to a 500 gallon tank resulted to a 1.2 m (4 ft) diameter by 1.8 m (6 ft) long tank with an optimum flow rate of approximately 45 gallon/min.

The storage tank and the circulation system were built by SwRI and delivered to EMD. Initial tests at SwRI indicated that a small layer at the top of the tank (about 3 inches) has less coal particle concentration, but otherwise the coal concentration was uniform throughout the tank. A schematic of the CWS storage and circulation system is shown on Fig. 6. The slurry is picked up vertically at the floating pickup, circulated by the circulation pump, and introduced to the tank at the bottom, through a flow distribution manifold. The pump is operated intermittently, with 15-minute on/off time. This timing is chosen for the reliable operation of the pump, without failure. The slurry is reblended at SwRI and transferred to EMD in 55-gallon drums. When the drums were opened, several thickened slurry layers were observed at different heights of the tanks.

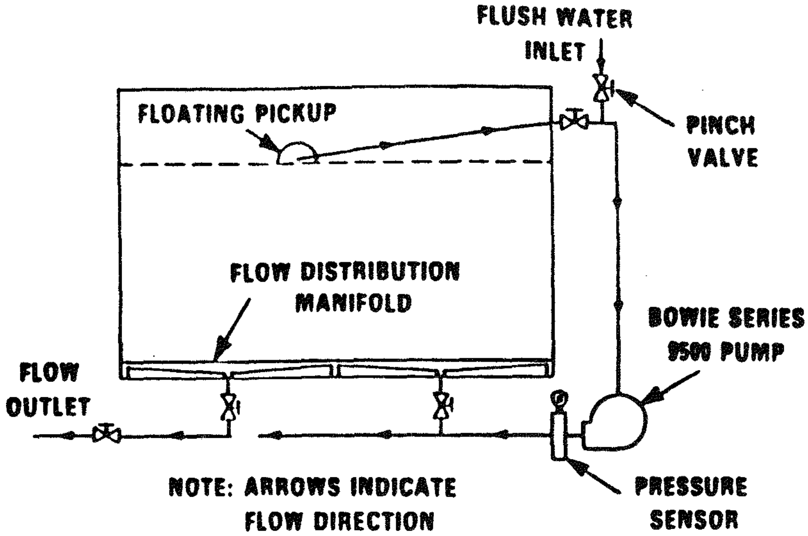

Figure 6. Schematic of slurry storage system.

The slurry was mixed within the drums by using a propeller, mounted on a long shaft, rotated at about 200 rpm, for a total time of about 1.5 hours per drum. The location and the dipped length of the shaft is changed in about 5-minute intervals. The presence of the thick CWS layers was checked by its resistance to a dip stick and by visual checks on samples taken. After this preparation, the slurry within the barrels were transferred to the storage tank and the tank operation was started on May 2, 1988. It stopped only once, for about 64 hours, at the beginning of August, 1988, when the bearing of the main circulation pump was failed due to seizure. Otherwise, it was continuously operational (with 15-minute on-off cycle) up to now.

The slurry properties in the tank were monitored continuously by taking samples from the circulation line, bimonthly. Previous tests at SwRI has indicated that the slurry concentration stayed constant for about 10 weeks. Our detailed tests started after 3 months of storage at the beginning of August, 1988. Three main properties of the slurry samples were measured. The Percent Solids were measured by weighing a sample before and after heating it in an oven at 212°F for 15 hours. This is expected to evaporate all the water and some volatiles content of the slurry. The percent ash content is measured by weighing another sample before and after heating it in an oven at 1100°F, for 4 hours. This is expected to burn all combustibles of the sample. The surface tension is measured by using a Cenco-du Nouy tensiometer (3) by the slow moving process described previously. For comparison, the surface tension of the tap water is also measured. These measurements are presented on Table 5 as well as on Figures 7, 8, and 9.

There are four trends expected to occur in the slurry tank. First is the evaporation of the water content through the free surface of slurry in the tank, with time. As the tank is not completely closed and vented to atmosphere, this evaporation may be important on a long time frame and would increase the percent solids and percent ash in the slurry sample measurements. The second trend is the separation of coal particles from the mixture by gravity. This would increase the slurry concentration near the bottom of the tank and decrease it near the top. As the suction of the slurry is near the tank

Table 5.
CWS Characterization During Storage
(Storage Started at 5.1.1988)

Date	8.4	8.15	9.1	10.6	11.1	12.1	2.15	4.1
% Solids (1.5 Hr 212 F)	51.8	52.0	51.5	51.7	51.7	50.0	45.2	45.15
% Ash (4 Hr 1100 F)	0.64	0.63	0.64	0.62	0.63	0.62	0.56	0.558
Surface Tension (dyne/cm)	79.1	79.5	78.6	77.4	77.4			77.3

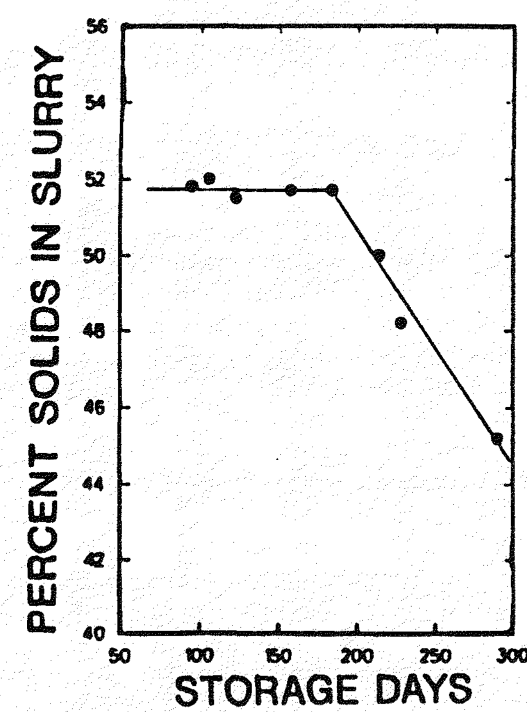

Figure 7. Precent solids in slurry as a function of storage time.

top, this separation would decrease both the percent solids and percent ash in the measurements taken routinely from the circulation pipe. The third phenomena is the possibility of the coal particles changing their shape by time due to the fact that they are continously subjected to high shear when they pass through the circulation pump. This may force the particles to take a more spherical shape by time. The fourth is about the additives used during the slurry preparation. They may be in the slurry solution or may be in the form of a surface layer over the particles. There is micrograph evidence in the literature (Ref 4) that the particle surface is covered with a solid-like layer, preventing coagulation of particles or keeping them in float. By the high shear of the circulation pump, this surface layer may be disrupted and broken, and the particle coagulation rate or separation rates may change with time. These may decrease the percent solid contents of the samples.

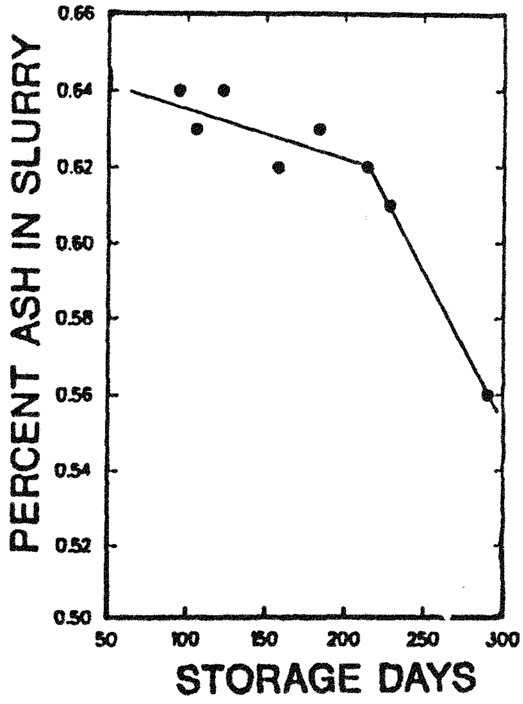

Figure 8. Percent ash in slurry as a function of storage time.

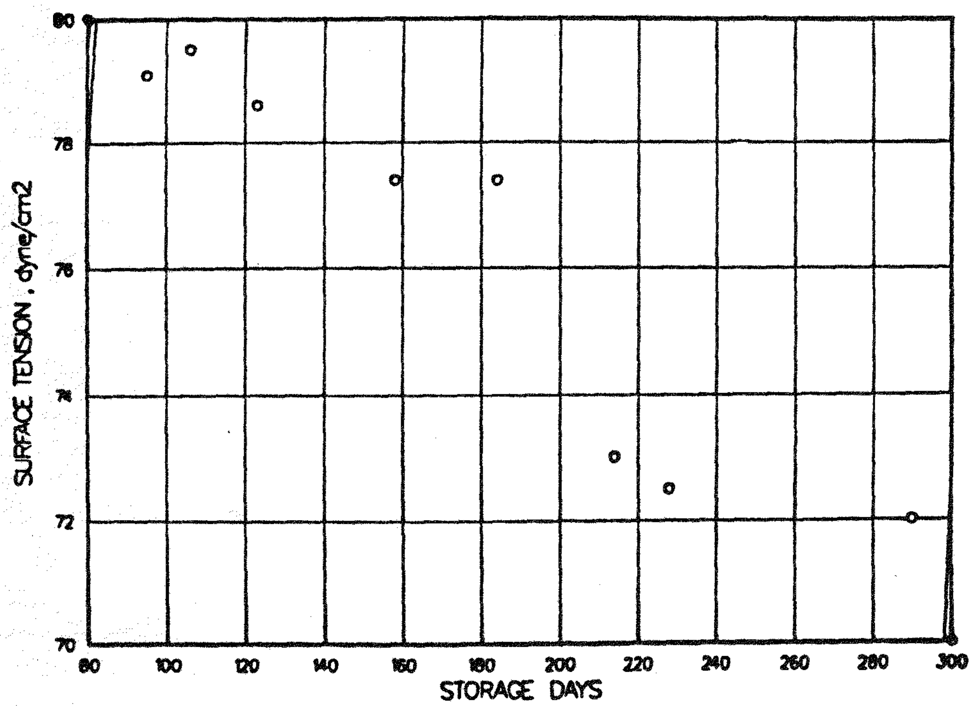

Figure 9. Surface tension of slurry as a function of storage time.

On Fig. 7 the percent solids measured in the slurry samples is shown as a function of time over about 10 months of system operation, indicating a trend that after about six months, the percent solid content of the samples begin to decrease. This implies that the storage system is capable of keeping the slurry intact for about six months. It also implies that the water evaporation is not an important and dominant factor in the operation of the system. On Fig. 8 the percent ash in the samples is shown as a function of time, which also indicates a large change in trends after six months. This also indicates that the water evaporation is not an important factor, because if it would be, the percent ash should increase during the first 6-month duration.

The surface tension measurements of the slurry samples are shown on Fig. 9. As the difficulty and uncertainty in the surface tension measurements are high, it is possible and more meaningful to discuss the general trends on a relative basis. The surface tension of the slurry samples are decreasing continually with time. The reason for this decrease is not known but it is speculated that it may be related to the surfactant coating properties of the particles changing by time. Comparing the surface tension measurements of the slurry samples against those of the water, measured at the same time, indicates that the slurry surface tension is not the same as water, but is higher.

After the analysis of the samples taken in April, 1989, it was felt necessary to have a survey of the slurry concentration within the storage tank. Using a special sampling probe, slurry samples are taken at different depths from the slurry surface. The results are shown in Table 6 (and in Fig. 10). This table indicates that the percent solids in the slurry is increasing from 45.15% to 55.25 % from 8 in. from the free surface to the bottom of the tank (33") for the samples taken while the pump was off. The coal concentration is almost constant from 8 to 24 in. but increases rapidly from 24 in. to bottom. There is definite concentration change as a function of distance from the top of the tank. The same test was repeated when the circulation pump was on and the results are also presented on Table 6. The concentration gradient is not as high under this condition. More over, both solids and ash measurements were considerably less than those of previous tests. Although the measured values were not consistent on a relative basis, they imply that flow circulation pattern in the tank is enough to disperse the concentration gradient during pump on period but the coal separation develops rapidly during pump off period. A possible mechanism for this may be the loss of surfactants from the surface of the particles. A particularly interesting point is when the measurements are taken at the pump off period, there is a section with concentration higher than 50% and another lower than 50%. Data taken when the pump is on only indicates percent water less than 50%, which may not be realistic. At this point, the slurry was not processed again and sampling was continued.

In these tests it was observed that the coal water slurry is a volatile suspension when it has a large free surface area and therefore to measure its surface tension is a difficult task. The surface tension measurements are more valid if the measurements are completed as soon as the sample is poured into the dish of the apparatus. Also, successive readings from the same sample bottle gradually decrease in surface tension readings. This could be due to the sample getting stratified in the bottle. The first sample taken from the bottle and measured within a minute or two after pouring the CWS into the petri dish is the most reliable of the successive measurements. The numbers reported are from this first analysis.

Table 6.
Slurry Concentration Survey in the Tank

Samples taken with the circulating pump off:

	Distance From Surface	% Solids	% Ash	Surface Tension dynes/cm2
A	8 inches	45.15%	0.558%	77.3
B	16 inches	45.0	0.561	76.4
(F)	20 inches	46.6	0.585	76.6
C	24 inches	45.32	0.559	76.0
(G)	28 inches	50.53	0.628	76.4
D	32 inches	54.42	0.653	79.1
E	Bottom (33")	55.25	0.669	79.0

Samples taken with circulating pump on:

1	16 inches	43.71%	0.547%	74.8
2	20 inches	45.27	0.570	77.2
3	24 inches	(71.76)	(0.886)	76.0
4	28 inches	45.77	0.572	76.6
5	32 inches	47.21	0.591	76.7

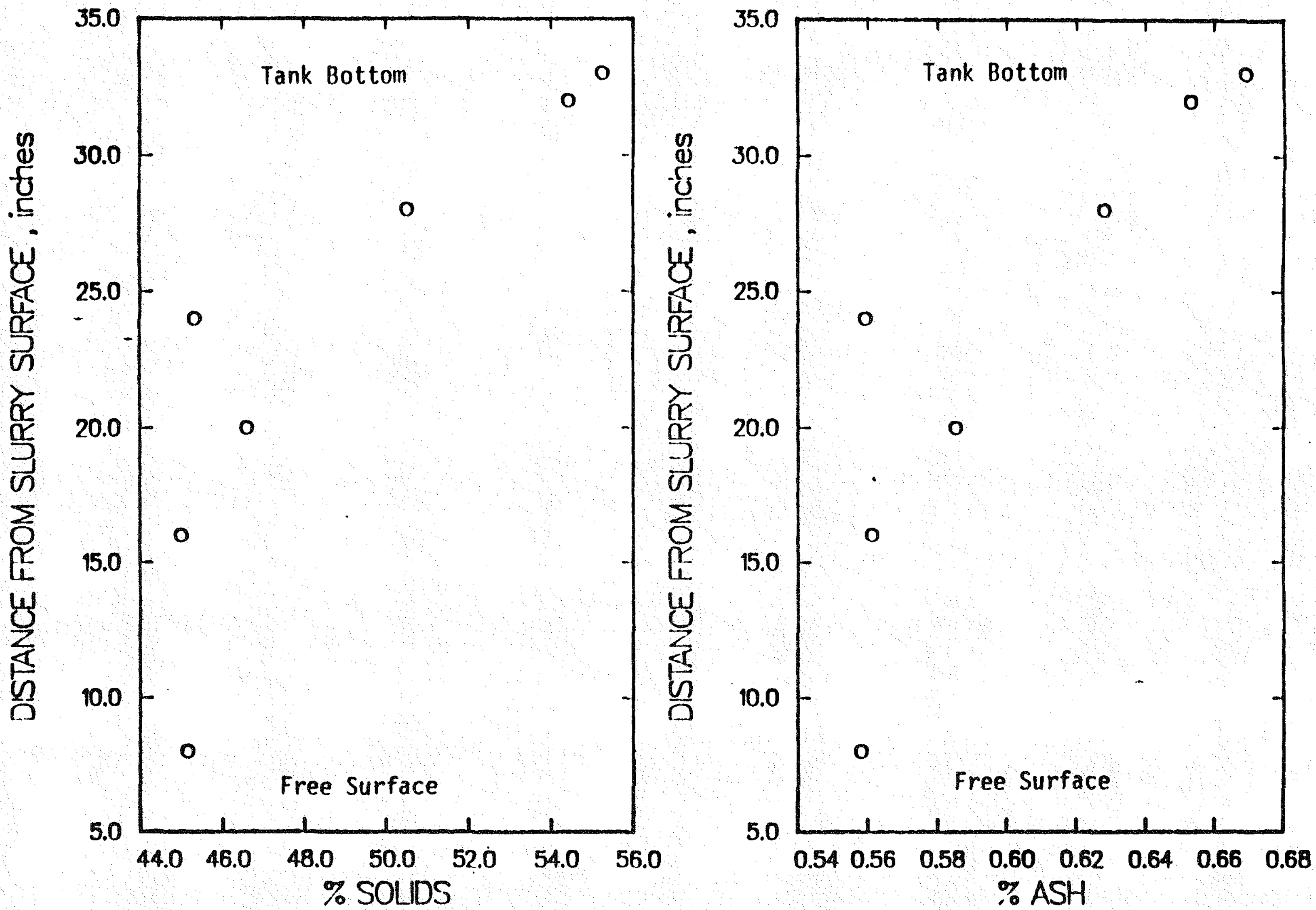

Figure 10. Solids stratification within slurry tank.

3.0 ENGINE TESTS SETUP AT EMD

At the end of the concept demonstration tests at SwRI, it was requested that the engine wear tests should be run at EMD. The decision of running tests at EMD has provided both a challenge and an opportunity. The challenge is to transfer the technology developed at SwRI to EMD in a timely manner so that the wear test objectives could be achieved in time. The opportunity is to be able to arrange a new test set up benefiting from the technology developed at SwRI and the information available in the literature up to that time. Both of these factors become an important issue at later stages of the project activities.

3.1 OBJECTIVES AND STRATEGY

A. OBJECTIVES

The specific objectives of the activities at EMD can be stated as follows:

* Identify conditions for sustained, reliable operation of the 2-645 engine with CWS,

* To perform 10 hr wear tests as per contract with DOE.

Other secondary objectives were as follows:

* To develop the EMD, in-house capabilities to run a coal water slurry burning diesel engine

* To develop an engine test set-up using the best available information at the beginning of this project with a distant goal of developing design information for locomotive applications

* To develop new information on the CWS combustion

* If possible, the amount of pilot fuel injected should be less than 10% of the energy of combustion at the maximum load.

B. STRATEGY

To reach these goals, a project strategy is chosen which would enhance the probability of success and increase the value of the outputs of the project. The overall project strategy was to use the experience developed by the tests at SwRI and at Adiabatics, Inc. to the maximum extent. To accomplish this, the decisions conclusions and recommendations of these previous activities are taken as the starting point for EMD activities. For wear rate tests, the three materials recommended most by Adiabatics, Inc. (5) were chosen for implementation without further tests and questioning. For engine combustion development, the following systems, components or technology suggestions of SwRI are used either directly (when applicable) or in principle.

* Same slurry is used after reblending
* CWS storage system recommendations
* Phase 2 Injection System Recommended
* Same Injector cooling system

There were two specific separate activities to reach the goals of the project. The first one is to design and develop an engine test system for burning CWS in a reliable, repeatable manner. The ignitability and combustibility conditions should be developed, the effects of several design parameters understood and engine operation near optimized condition is achieved. (These were not supplied by SwRI tests.) The second is to run the ten-hour tests. These two activities are assigned to two different groups. It was thought that during the development of the wear-resistant, coated components, the combustion studies could be completed. This report covers only the combustion-related studies of the EMD activities. Engine wear related studies are covered in a separate report. There were practical obstacles to achieve the combustion of the CWS in a diesel engine. One of the main technical obstacles for combustion of the slurry was identified as the water evaporation. This in turn would require the consideration of three distinct features. First, the enthalpy of the water evaporation is to be supplied before slurry ignition (or the gas temperature should be above a certain limit). Second, the air should be supplied to the coal particles but in a water vapor filled atmosphere. Third, these should be supplied in time so that the combustion of the coal slurry should generate heat at the efficient energy recovery portion of the piston motion. The third feature, namely the timeliness, is assumed to be achievable by adjusting the timing of the fuel injection. To heat the chamber gases three methods were previously tried by several researchers (Ref 6, 7, and 8) namely, the use of (a) pilot fuel (or another energy supplying combustion process such as gas igniters), (b) heating the inlet air before introducing into the chamber, (c) insulating the combustion chamber walls so that the heat losses during the compression could be reduced and thus the gas temperatures could be increased. There were two other methods that may be used for this purpose, which were not previously recognized or utilized by other researchers. They are: (a) heating the liners instead of cooling them, and (b) increasing the engine compression ratio. Among these alternatives, the use of the pilot charge, the inlet air heating and increasing the compression ratio are the methods chosen for implementation in this design. In order to supply the slurry heat of evaporation, in time, the control of both the main and slurry fuel amounts and their injection timings were considered essential, particularly to determine the ignitability conditions of the slurry for different engine speeds and loads. These strategic requirements become the basis of the design decisions.

Another major technical obstacle was the plugging and seizure of fuel pumps and/or injectors. The developments and tests at SwRI have shown that an EMD unit injector can be used for slurry injection after modifications for cooling and clearances. However, its capacity was limited, (850 mm3/slot fuel injection). In order to develop the second generation of injectors, with higher capacity (1500 mm3/injection) needed for locomotive applications, the pump and nozzle system suggested by SwRI was utilized with the unit injector as a back-up system. Details of these systems are presented in the following sections.

EMD was in the process of design and procurement of a single cylinder version of its 710 engine at the beginning of the decision of running tests at EMD. However, due to the delay in the delivery of this research engine, a two-cylinder 645 research and development engine was allocated to the present activity. As this engine was not turbocharged, the increase of the engine compression ratio to heat the chamber gases become more important, particularly at the high slurry loads with small amounts of pilot charge.

The use of the same slurry used by SwRI in the previous phases of the project was part of the strategy of utilizing the previously developed technology as much as possible. Therefore, it was decided to get the slurry from SwRI, ready to use in the engine, to maintain in mixed conditions during the tests and to check the CWS characterization at the beginning, middle and end of the tests. The use of a different slurry would be considered if this system would prove impractical or impossible.

3.2 THERMODYNAMIC CONSIDERATIONS

At the beginning of the combustion system design, analytic methods were used to ensure the attainment of ignitability conditions for the slurry. A literature survey indicated that there were two analytic modeling works applicable to the engine performance calculations with slurry combustion. The first one is the computer program developed by Kishan and Caton (9). A copy of the program, as available from Kishan's thesis, is generated at EMD, but attempts to run it failed, mainly because some of the program lines were missing. Attempts to get help from the senior author was also fruitless as the program has been deleted from their computers. To make it operational would necessitate a new project and require time. The second model was the use of the program KIVA COAL (18). As the original KIVA code was available at EMD, it was thought that the modification for CWS, detailed in this report, could be accomplished with reasonable effort. However, discussions indicated that the code is still not completed and the code predictions are not validated (11). As a result, much simpler and approximate thermodynamic calculations were the only analytic means available to the project at the time. A simple calculation indicated that for the modification of the present engine operation to CWS combustion, there are two basic properties of the CWS fuel affecting the engine design. They are the heat of combustion and the latent heat of evaporation of the CWS. Table 7 shows some of the properties calculated for the use of DF2 and CWS (for 50% coal-water slurry).

Table 7.
Comparison of Combustion Related Properties For 50% Coal-Water Slurry.

Fuel	Lower Heating Value BTU/lb	Latent Heat of Evaporation	Density	Energy per Unit Volume (Relative Basis)
DF2	18400	95	0.85	1.00 (2.12)
CWS	6900	524	1.07	0.471 (1.0)

Consequences of this data is informative. First, when the engine is switched to CWS from DF2 operation, the engine power will drop for the same volume of fuel injection. In order to have the same engine power level, the injector capacity should be increased by about a factor of 2.12, assuming the same engine efficiency is maintained. Therefore, for a CWS burning engine of the future, with the same power level of the present locomotives, a new injector system with about 2.12 times the capacity of the present one is needed. Even at the maximum capacity of the present injection system, the engine power, the cylinder pressure and temperatures would be about 0.47 percent of the present engine operation values at maximum load, approximately. Second, the latent heat of evaporation of slurry is expected to reduce the cylinder gas temperatures and (to a lesser extent) the cylinder peak temperature. As the peak power is already less, the difficulty in ignitability and combustion of the slurry in the desired crank angle period (to recover the piston work efficiently) becomes a more serious problem. To prevent longer ignition delay times and thus eliminate the possible loss in the piston power recovery efficiency, additional heating of the combustion gases during compression is essential.

On engine operation with slurry, alternative ways to heat the chamber gases are considered in detail from an engine thermodynamics point of view. The way the energy is added to the chamber gases is important because it may effect the thermal engine efficiency considerably. At the time of this design study, methods to increase the gas temperature by heating through pilot combustion, heating the inlet air, and by increasing the engine boost pressure was reported in the literature (6, 7). Heating by pilot injection is detrimental for engine thermal efficiency as it would require burning additional fuel during compression period, increasing the piston work needed for compressing the gases. Additional energy comes from the pilot diesel fuel chemical energy. Inlet air heating is detrimental as it would reduce the volumetric efficiency of the engine which would reduce the efficiency and the power level of the engine. This is particularly important for the 4-cycle engine operation because the engine efficiency and power is more sensitive to the volumetric efficiency. Possible heating sources for the additional energy to heat the inlet air may be the exhaust gases, or a separate air heater. The use of engine exhaust gases may be desirable from an efficiency point of view, but it would require a gas-to-gas heat exchanger, large equipment and also would increase the back pressure of the engine, which may cause some efficiency reduction. Increasing the inlet air pressure by increasing the engine boost is also detrimental to efficiency because the power for the increased compression ratio should be obtained either from the turbocharger turbine or engine shaft. The energy should come from the compressor work. This means redesigning the turbocharger (if it is a free floating type) with some increase in its size or reducing the power of the engine further if the turbocharger is driven from the crankshaft.

Other alternative methods of heating the combustion chamber gases are also possible. First, an increase of the engine compression ratio would also increase the gas temperature (and the pressure of the engine). In this case, the additional energy would come from the piston work which is expected to be more efficient than obtaining the same energy from the engine shaft through turbocharger, as in the case of increased boost pressure. This may also increase the CWS evaporation rate (and decrease the ignition delay time) because the latent heat of water evaporation decreases with an increase in the pressure. On the other hand increasing the compression ratio would also increase the ring friction which may offset some of the possible gains. Another method is to increase the cylinder wall temperatures by not cooling the cylinder as much. This will also reduce the engine cooling system size and increase the total system efficiency a little. A completely different way to achieve the heating of the cylinder gases is the use of exhaust gas recirculation. This would also decrease the unburned hydrocarbons and soot in the exhaust and increase the combustion efficiency of the engine. However, it is also detrimental to the engine efficiency, as it decreases the engine volumetric efficiency.

After these considerations, it was decided to have three different methods of heating for the experimental test setup: (1) The pilot charge, (2) the inlet air heating, and (3) increase in the engine compression ratio. Other alternatives were left for future work.

During the evaluation of these alternatives, another important feature of CWS usage is realized. That is, the suitability of the 2-cycle engine operation to CWS combustion more than the 4-cycle engine operation. It is a known fact that the combustion wall temperatures and the in-cylinder gas temperatures of 2-cycle engines (at least for the competitive locomotive engines) are higher. This is because of two reasons. In 2-cycle engines, there is combustion in every cycle and hence less time to cool the walls, and the cylinder gases cannot be cleaned off completely during the exhaust period due to scavenging characteristics of the engine, or in other words, there is an internal EGR. Therefore, for the similar engine power levels, the gas temperature at the possible point of CWS ignition is already much higher for 2-cycle engines than 4-cycle engines. Therefore, the evaporation of the water in CWS will reduce the temperature cylinder gases which in turn will reduce the engine wall temperatures (a limiting factor to increase peak pressures in 2-cycle engines) (12). This could be utilized as an important engine design characteristic to increase the engine boost or compression ratio further, to improve the engine efficiency.

As far as the CWS combustion is concerned, the evidence in the literature indicates that (Ref 13) it has the same combustion characteristics of diesel fuel, namely, with premixed and diffusion controlled combustion. Coal particles themselves do not show considerable ignition delay. On the other hand, the burning rate of the coal particles is of flame propagation type so that its control is a major problem. If the air is available, the coal particles could burn at a very fast rate, leading to very high peak pressures and temperatures. Hence the main problem, as far as the combustion of the CWS is concerned, is the long ignition delay times and the energy needs for water evaporation.

3.3 PREDICTION OF IGNITABILITY

In order to estimate and assure the ignitability of the CWS for a wide range of engine operation, a simple thermodynamic analysis is used to estimate the gas temperatures at the possible point of ignitability. For this purpose the engine cycle simulation code available at EMD is used. A general description of this code is available in (14), although it has been modified considerably for EMD engine applications. The code is modified for dual fuel operation simulating the pilot and main fuel injections. Both pilot and main fuels are simulated as diesel fuel. The fuel amount at the main injector is adjusted to have the same combustion energy as the CWS injected and the cylinder gas temperatures are predicted at the expected point of main fuel ignition. The thermodynamic model foresees that, at that point, the slurry water is suddenly evaporated and completely mixed with the cylinder gases. The predicted gas temperatures are considered as a conservative estimate of the cylinder gas temperature at the point of ignition of CWS in the engine and are compared against the CWS ignitability criteria available in the literature (15, 16).

As a sample, a nomogram is shown in Figure 11 which was developed by using this approximate CWS ignitability prediction methodology. For this plot, the simulated engine operating conditions are as follows:

Airbox Pressure	:	21 psia
Airbox Temperature	:	660°R
Cylinder Air	:	11.289 Grams
Airflow Rate	:	25 lb/min/cyl
Exhaust Manifold P	:	14.7 psia

In Fig. 11 the abcissa is the predicted gas mixture temperature at the point of ignition and the ordinate is the cylinder trapped air/slurry water mass ratio. The parameter (on the curves) is the engine geometric compression ratio. As the cylinder mass at the beginning of compression is known for a given amount of slurry injected, then the air water mass ratio at the point of ignition is known. Then, for the given compression ratio, the cylinder gas mixture temperature can be found from the nomogram. The ignitability limits for CWS as reported in the literature are also shown in Fig. 11. To compensate the inaccuracy of the model, conservative estimates are used which indicate that an engine test setup with an engine compression ratio of 24 can be able to ignite the CWS, without any other ignition enhancement.

This method of analysis does not have the desired physical details, particularly representation of ignition delay for the coal water slurry; however, it was found to be satisfactory in estimating test setup design. This model was used in the prediction of the engine performance under different engine operation conditions and was useful in learning the trends. Samples of calculations are given on Fig. 12 and 13 where the pilot and main fuel injection profiles

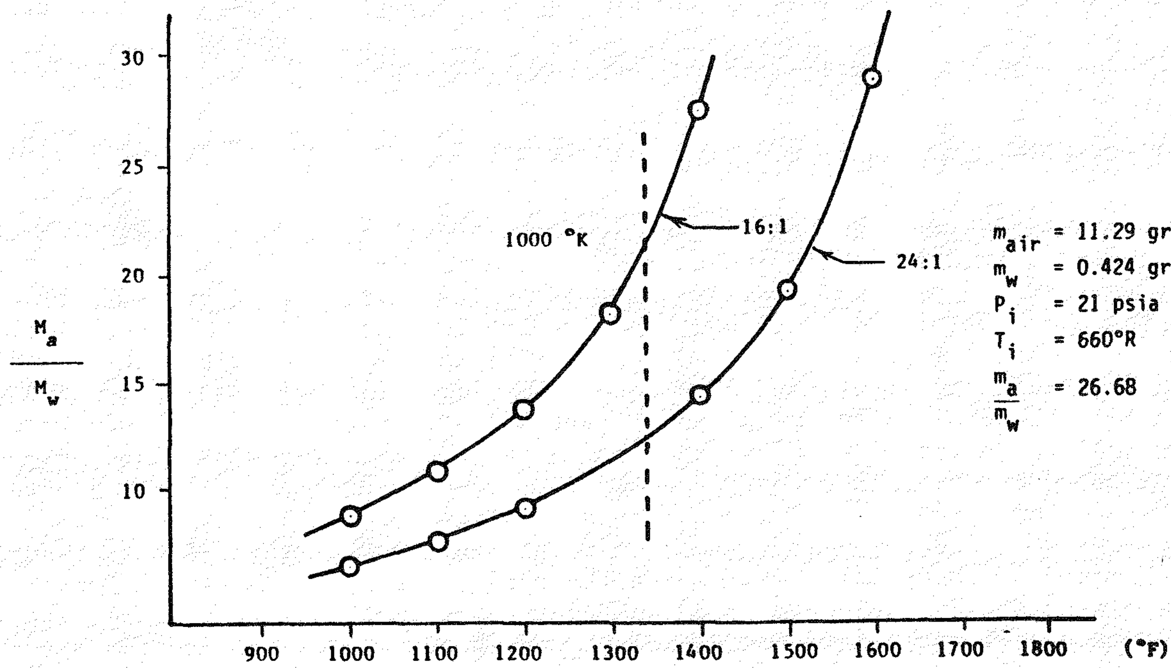

Figure 11. Example of predicted gas slurry mixture temperature at the expected slurry ignition point.

and the predicted fuel burning rates are shown for different injection advance and different engine compression ratio. The method is also applied to the question of whether the inlet port timing should be modified for CWS combustion. After some studies, it was concluded that within the sensitivity of the model, the port timings should not be changed.

The piston bowl shape is modified to increase the engine compression ratio. On Fig. 14 the piston bowl design modifications for 18, 20, 22 and 24 compression ratio pistons are given. Due to the time limitations the use of only 16, 20 and 24 CR pistons was planned in the tests.

3.4 SIMULATED STATE EQUATION FOR CWS AS A FUEL

Another approach to simulate the CWS in the engine performance prediction codes may be to consider the slurry as a pseudo fuel, determine its equivalent chemical formula from physical composition, assume equilibrium combustion, find its equation of state (as described by its enthalpy and/or specific heat as a function of temperature) from those of its constituents, and its equivalent ignition delay information from the data as available in the literature for CWS. Although it is quite simple, this approach is expected to have some instruction value. Of course it would have certain limitations as far as the information content about the details of the CWS related processes going on in the cylinder. However, from the basic thermodynamics point of view, it would give a reasonable estimate of cylinder gas temperatures for ignitability estimates. Such a method is tried but not fully implemented within the scope of this project. A review of CWS state representation is presented here.

For the purpose of this analysis, the coal composition is assumed to be the same as in (17). For simplicity the sulphur content is neglected, its weight is allocated to other components, and the composition of the coal, by weight, is assumed to be:

Carbon	86.13%
Hydrogen	3.25%
Nitrogen	1.32%
Oxygen	9.30%

As a result, the chemical composition of the dry coal is assumed to be represented by the chemical formula:

$$C_{12.4}\, H_{5.6}\, N_{0.16}\, O$$

The slurry is assumed to have 50 percent of this dry coal, 1 percent ash and 49 percent of water. The molecular weight and thermodynamic properties of water is classical but the treatment for ash and coal is not well known. These are available in (10, 18) where the ash is considered to be a mixture of several minerals and after some calculations it was represented by an equivalent material whose molecular weight is 664.04, heat of formation 0.0, and other thermodynamic properties as given in (10). As a result, the complete combustion of the slurry may be represented by the following equation, with an excess air ratio 1.

$$C_{12.4}\, H_{5.6}\, N_{0.16}\, O + 0.0026\ \text{Ash} + 9.6\ H_2O + 13.3\ (1 + \phi)\ (O_2 + 3.76N_2)$$

$$\rightarrow 12.4\ CO_2 + 50.088\ N_2 + 0.0026\ \text{Ash} + 12.4\ H_2O + 13.3\,\phi\ (O_2 + 3.76N_2)$$

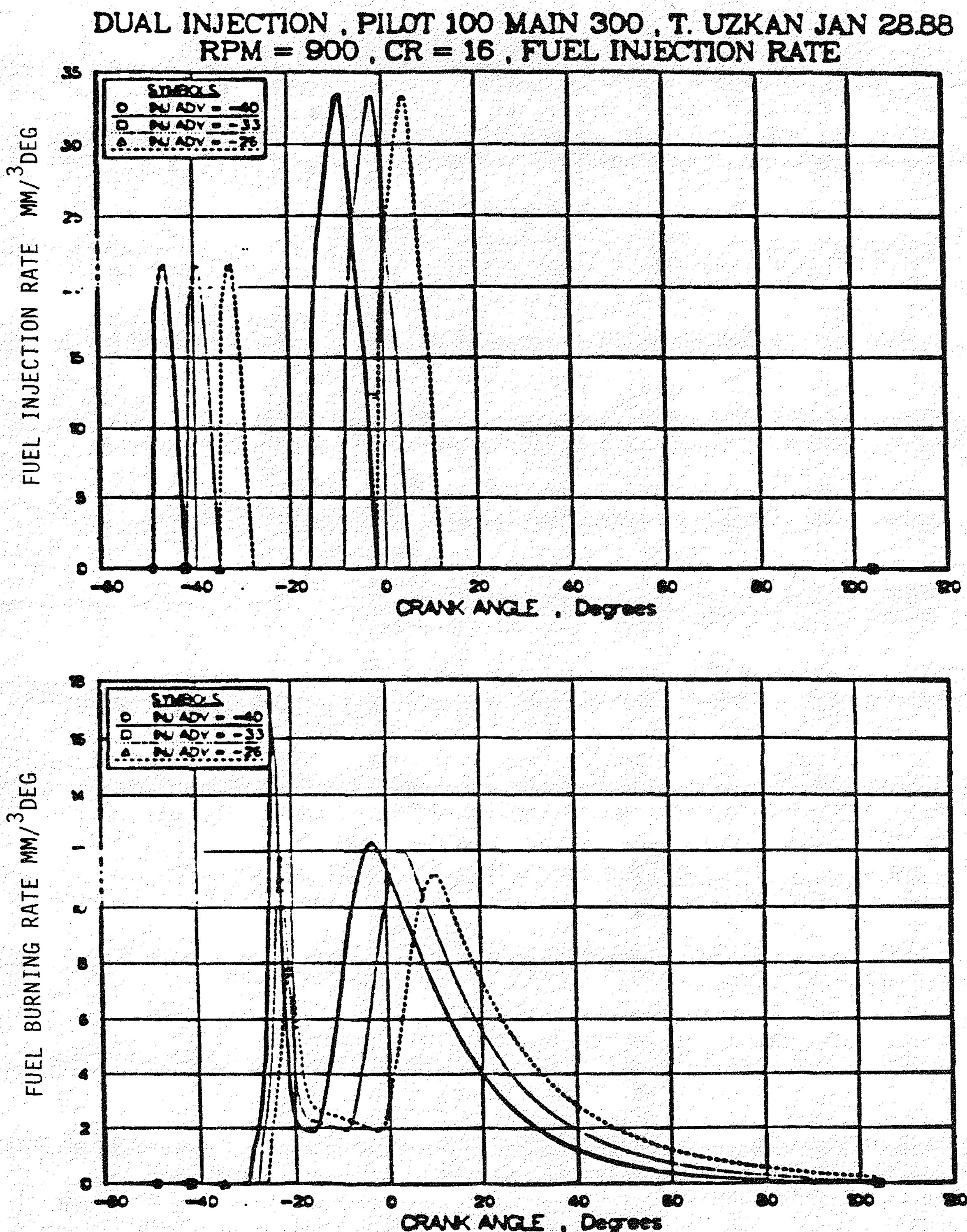

Figure 12. Examples of dual fuel injection code predictions. Fuel burning rates for different injection timings.

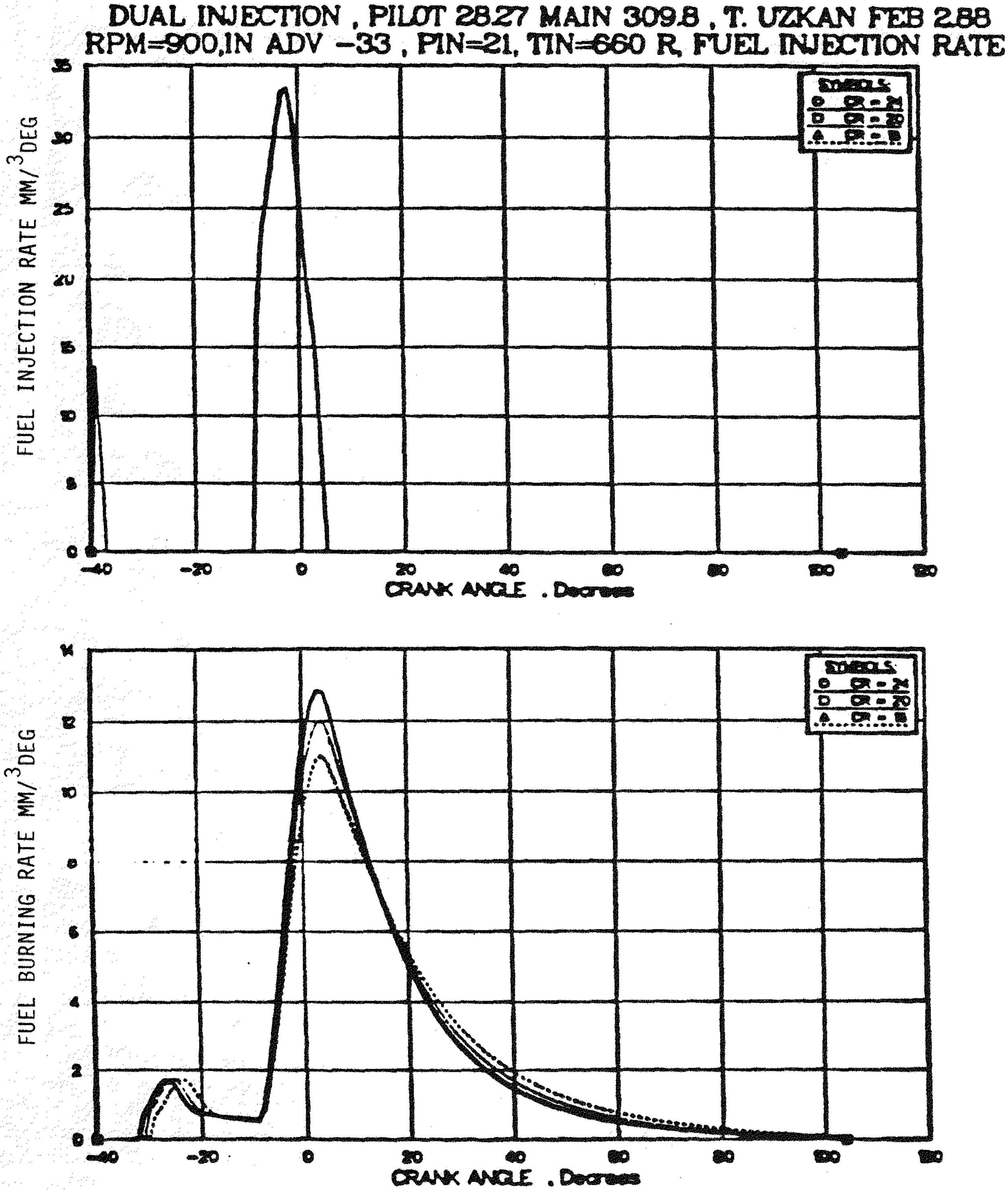

Figure 13. Fuel burning rates for different compression ratio.

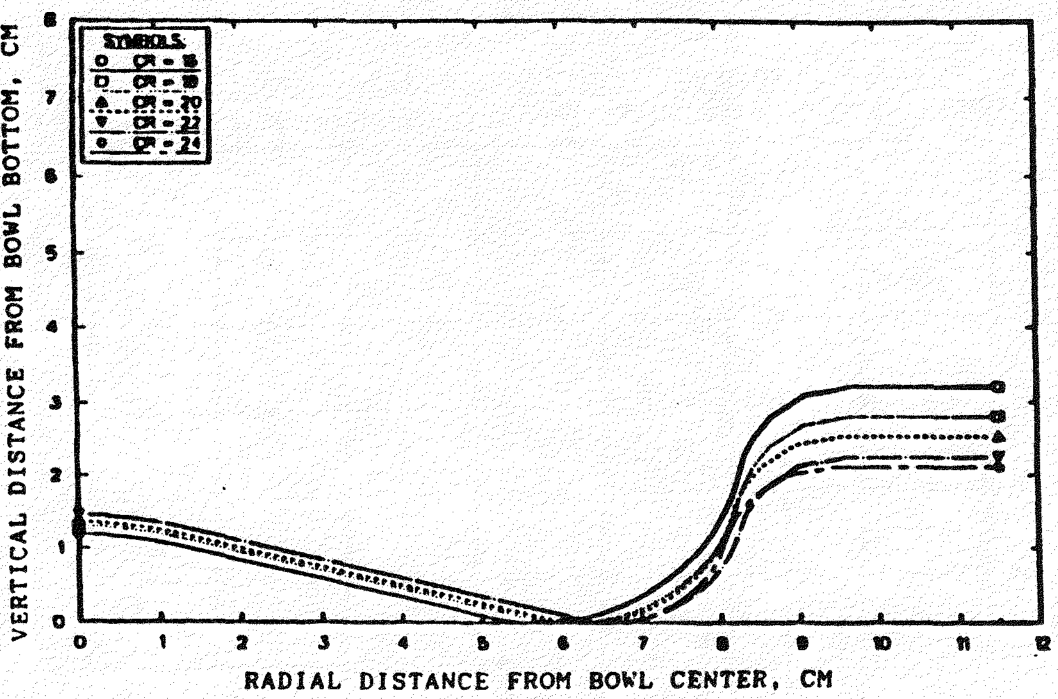

Figure 14. Piston bowl profiles for different comparison ratio pistons.

Assuming that the combustion of slurry from reactants to products can be represented by the representation and nomenclature of (19) and assuming that the enthalpy of combustion of slurry is 6675 BTU/lb, one can derive the following result to calculate the slurry thermodynamic properties from its constituents:

$$\begin{aligned} h^\circ_{f_{sl}} + \Delta h_{sl} = & -1288.03 - 13.3\,(h^\circ_f + \Delta h)_{O_2} + 0.08\,(h^\circ_f + \Delta h)_{N_2} \\ & + 12.4\,(h^\circ_f + \Delta h)_{CO_2} + 0.0026\,(h^\circ_f + \Delta h)_{Ash} \\ & + 12.4\,(h^\circ_f + \Delta h)_{H_2O} \end{aligned}$$

The thermodynamic properties of the slurry constituents are available in 10 and 18 in a form suitable for engine performance modeling. To make the heat of combustion of the slurry zero at the reference temperature of 298oK, the hfo of the slurry should be taken as +594.926 KCal/Mole. This set of data and equation should represent the slurry thermodynamics for the first approximation. The ignition delay data of the slurry available in the literature is not analyzed and put together and integrated with this representation to complete the model. The data reported in (20, 21) can be the first candidates for ignition delay characteristics of the slurry.

In review, it may be possible to represent the slurry in the form of a pseudo fuel, with a chemical equation based on its physical analysis, with a combustion representing the complete equilibrium combustion. The thermodynamic properties of the CWS fuel can be calculated from the properties of its constituents, using the tabulated properties of coal and ash in (20, 21), (and for other materials as available in JANAF Table (22)) as is presented above for 50% CWS. This method is not fully developed and utilized within the time period of this project.

3.5 ENGINE TEST SYSTEM DESCRIPTION

The engine test setup consists of the following subsystems or components:

* Fuel and Air Supply System
* CWS Storage Tank and Circulation System
* CWS Transfer and Changeover System
* CWS Injection System
* Emissions Instrumentation

The CWS storage and circulation system is described in Section 2 in detail and will not be described here further. Other systems and components are covered in this section.

A. ENGINE

The engine used in these tests is a 2-stroke cycle, 2-cylinder, EMD 645 type engine specifically designed and used for engine research and development. Essential features of this engine is given in Table 8. One cylinder of the engine is kept with the original configuration and its fuel rate is always under the control of the engine governor. The other cylinder was modified for dual fuel, CWS injection operation, where the pilot injector was operational with DF2 (Diesel Fuel #2) and the main injector was operational with either DF2, water or CWS. A specially designed changeover system switches the fluid supplied to the main injector to any one of these fluids. The CWS cylinder fuel system was not under the control of the engine governor. The fuel rates to the pilot and main injectors of this cylinder were separately controlled through their respective injector pump rack position controls.

Table 8.
Basic Features of EMD 645 Diesel Engine

Number of Cylinders	2
Engine Type	Blower Scavenged
Engine Cycle	Two cycle, uniflow engine
Bore, (in.)	9.025
Stroke, (in.)	10.0
Displacement, (cu.in)	645
Compression Ratio	16:1
Rated Speed, rpm	835
Maximum Speed, rpm	900
Combustion Chamber Type	Open Chamber, Direct Injection
Type of Injection	Unit Injector

The engine is attached to an electric dynamometer, which measures the engine BHP (total of both cylinders) through the measurement of output voltage and current.

B. FUEL SUPPLY ARRANGEMENT

The fuel and air supply to the test engine is schematically shown on Fig. 15. The diesel fuel supplied to the engine comes from the main fuel supply tanks of the engine test cells. Through the electronic fuel meter, DF2 is supplied both to unit injector of the DF2 cylinder and to the pilot injector of the CWS cylinder and to the CWS switchover system. This switchover system has also supply connections from city water and from the CWS storage tank. The switch-over system can supply DF2, water or CWS to the main injector of the CWS burning cylinder. The air to the engine is taken from the test cell, passes through the inlet air flow meter and the inlet air heater before admitted to the engine blower. Air flow rate is measured for the total engine not for individual cylinders, hence the air fuel ratio of CWS cylinder is not known, precisely. Separate flowrate measurements to each cylinder was not practical due to the engine blower arrangement. On the other hand, separate exhaust pipes to each cylinder were maintained so that exhaust emissions from the CWS cylinder could be measured independently.

C. CWS TRANSFER AND CHANGEOVER SYSTEM

There are several stringent requirements on the CWS supply system running from the CWS tank to the engine cylinder. During unoperational periods of the engine, the CWS should not be left in the supply lines against coal particle separation, accumulation at certain regions and plugging the system. Water also should not be left in these lines against rust formation and plugging of the lines. Additionally, during the engine startup or at partial loads, burning of DF2 fuel in the CWS cylinder may be necessary and unavoidable. Moreover, mixing of the CWS and DF2 is incompatible, as they would coagulate and form a heavy sludge immediately, plugging the fuel transfer lines, fuel pumps or injectors. Development of anticoagulant additives is an alternative to eliminate the last requirement, but it is part of the coal fuel development and reliable anticoagulant additives and the technology to use them effectively were not available at the time of the system design. Therefore, an operational procedure consisting of the following three steps: (a) having DF2 in the CWS lines during storage, engine start-up and at low engine loads, (b) flushing the lines off DF2 with water completely and then (c) flushing the water with CWS, could answer all these requirements. The fuel switchover system of this test setup is designed to satisfy these requirements.

The schematic arrangement of the switchover system is shown in Fig. 16. To simulate a more realistic system characteristic, distances of the components are chosen to approximately correspond to an expected locomotive application as practically as possible. The CWS storage tank and its circulation system described in section 2.3 are located outside the test cell (Fig. 17). The fuel to the main injector of the CWS burning cylinder is controlled by the remote controlled switchover valve 1 (Fig. 16). There are supply and return lines of DF2 to this valve, which also has inlet and outlet connections to the fuel circulation loop. There is a fuel cooling system on this loop, to keep the temperature of the slurry down, and there is a fuel circulation pump. When engine is not operational or during the engine start-up period, there is DF2

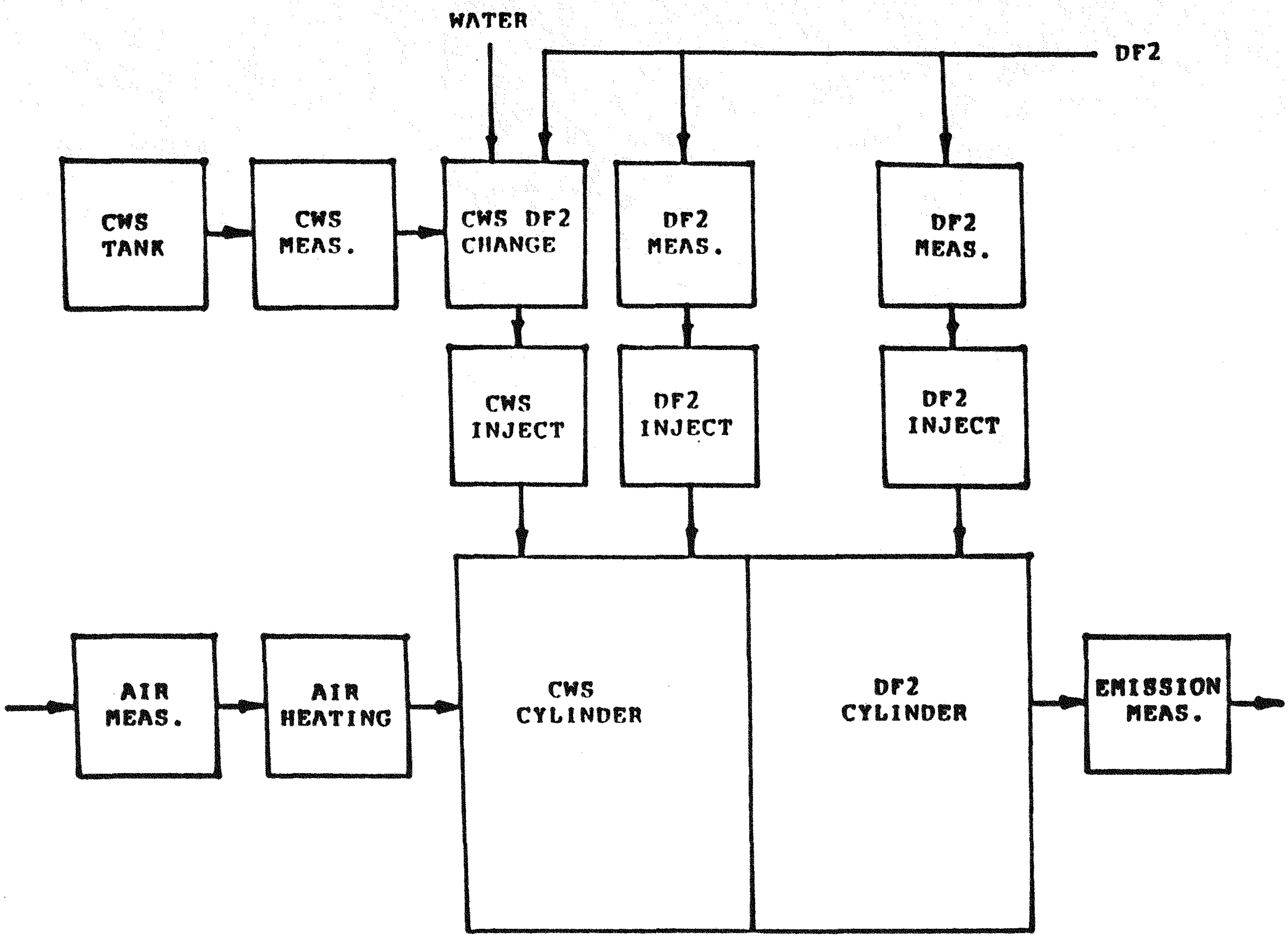

Figure 15. Engine test setup schematic.

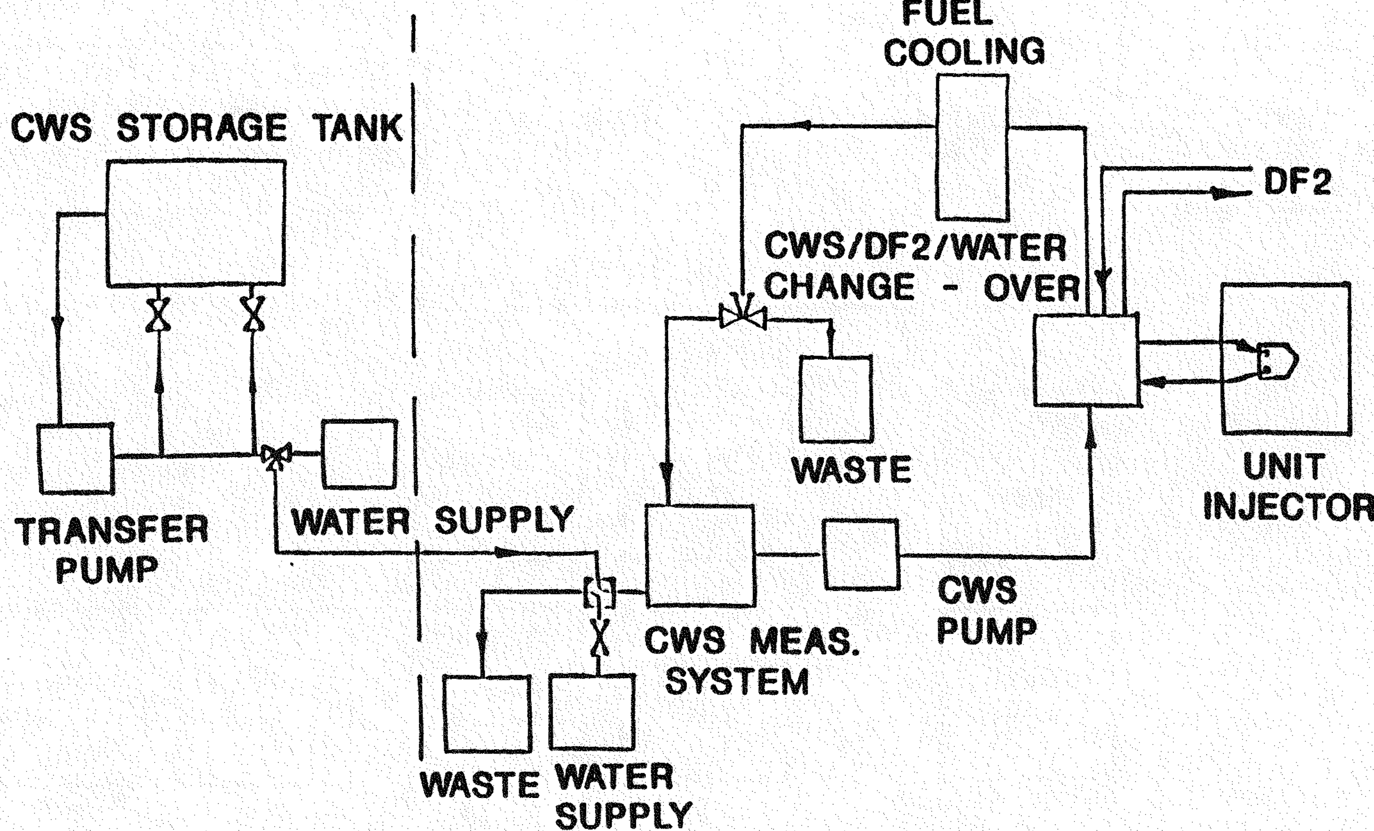

Figure 16. Coal water slurry storage and changeover system.

fuel in this circulation loop. To switch to CWS, the valves 1, 2 and 3 are activated so that the DF2 supply to valve 1 is shut off and the water is supplied to the circulation loop from the water supply 5. After all DF2 is flushed to waste 7, through the activation of valves 2 and 4, the CWS will replace the water in the loop. The control of the fuel injection rate at the main injector is through the main fuel pump rack position control. All these valves are remotely controlled from the control room as a prelude of simulating an expected locomotive fuel control system. The flushover time of the circulation loop from one fluid to another is a critical time constant for activating the remote control valves. It is determined during the switchover tests as will be discussed below in Section 4.4.

A dual strainer is used on the line from the CWS storage system to the fuel supply loop, to eliminate the possibility of plugging the injector from possible agglomerated coal particles. Certain amount of pressure is maintained in the supply line and the circulation loop to prevent the formation of cavitation bubbles through the use of relief valves. Necessary gauges are installed on the lines to monitor the operation of the system.

D. COMPONENTS

The details of the test set up and some of the components are shown in Figures 17 through 25. In Fig. 17, an overhead view of the slurry storage system outside the test cell is shown. The slurry circulation pump and the slurry suction pipe is observable on the tank (Fig. 18). The remote control valve is located near the wall and the electronic slurry flowmeter is the box on the wall. Fig. 19 shows the side view of the 2-cylinder 645 engine in the test cell. The DF2 cylinder is on the right and the CWS cylinder is on the left side while the dynamometer is at the back end of the engine. Two of the engine's separate exhaust pipe extensions are also observable on the top of this figure. On the left side, the exhaust emission sampling pipes are observable on the CWS cylinder exhaust pipe. On Fig. 20 the back side of the engine is shown where the dynamometer is apparent. On the dynamometer is the engine blower. The fuel injection system for the CWS cylinder is installed on the platform observable on the right side. From the end of the crankshaft, a belt system drives the main injector pump shaft. The details of this injector system are given on Fig. 21. There are two parallel shafts driven by the pulley-belt system from the engine crankshaft. On this picture, the top shaft is for the main injector and the bottom shaft is for the pilot injector. On each shaft, after the driven pulley, there is a flexible coupling, a timing differential gear and its remote control motor, a flywheel and a fuel pump are installed. On the top of the picture, only the bottom part of the main fuel pump is observable. It is a single-cylinder injection pump with maximum capacity of 1500 mm3/injection, at maximum injection pressure of 15,000 psi, made by United Technologies type 1-C-Q. Line pressure sensors were installed on the main and pilot injection lines from pumps to injectors. As the main injector nozzle, an EMD unit injector nozzle is used after modification to eliminate the cam activation and with the same injector body cooling system described in (23) in detail. The pilot injector was a stanadyne injector, having a capacity of 250 mm3/injection, with a fuel pump made by Majormec, SIMMS-SPEZN, (an original 6-cylinder injector pump, modified for operation of the first cylinder only). The amount of fuel injected by each pump is remotely controlled through their rack position controls, and the timings can be changed through the electric motor controls above the differential timing gears located on each shaft between the fuel pump and its driving gear. The rack position indicator sensors are observable on the figure. The rack position indicator sensors were calibrated during the start-up of the engine, against electronic fuel meters. During the test, the fuel is monitored through the rack position indicators, but the electronic scale data was printed out at the data logger. In Fig. 22 the CWS cooling tank is shown during the installation of the equipment.

The air inlet cooling tank section to the engine is shown on Fig. 23. The engine air is taken from the test cell, as observable on the upper left corner of the photograph, goes through an airflow meter (ACCUMASS Model 731-18-2) and an inlet air heater section, (designed with a capacity of 100 Degrees F over the ambient) and then enters the blower. On the right side are the two exhaust pipes from each cylinder. Details of the air heating section from the steam inlet pipe side is shown on Fig. 24. In Fig. 25 the engine control room is shown, where engine operation and slurry transfer can be controlled.

E. FUEL INJECTION SYSTEM

The fuel injection system of the CWS burning cylinder is one of the major components requiring new technology development. The requirements on the main injection system components can be grouped into several categories.

The essential and important characteristics of the new injection system is both the amount and timing of the fuel injected could be separately controllable for both the main and pilot injection. This is considered to be an important

feature for identifying the correct timing of the engine at different speed and load operation conditions. On the other hand the CWS injection pump was injecting both slurry, water and diesel fuel through the same injection pump, at high injection pressures, up to 15,000 psi. The operation of the pump with

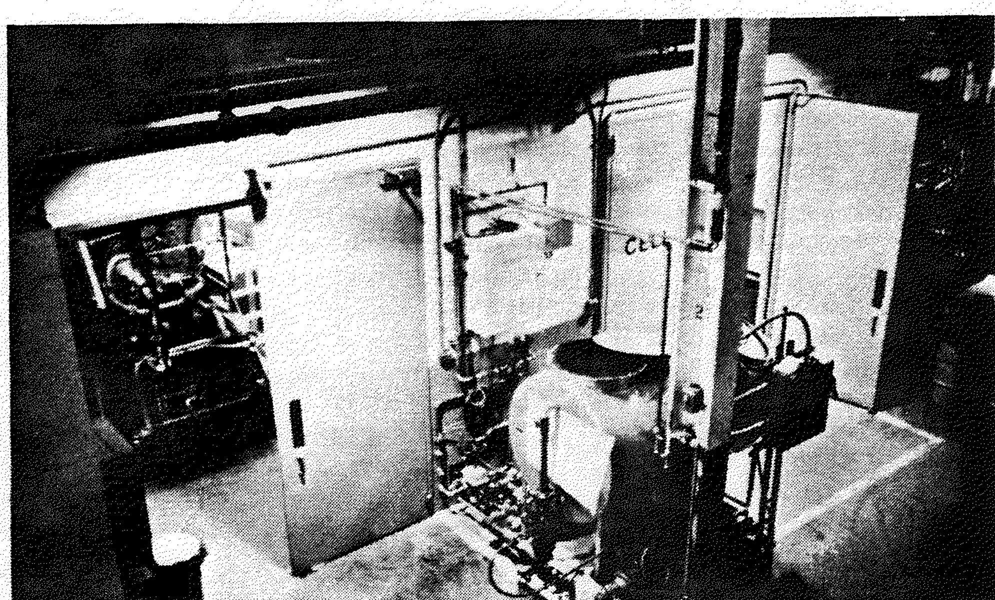

Figure 17. Slurry storage system and test cell.

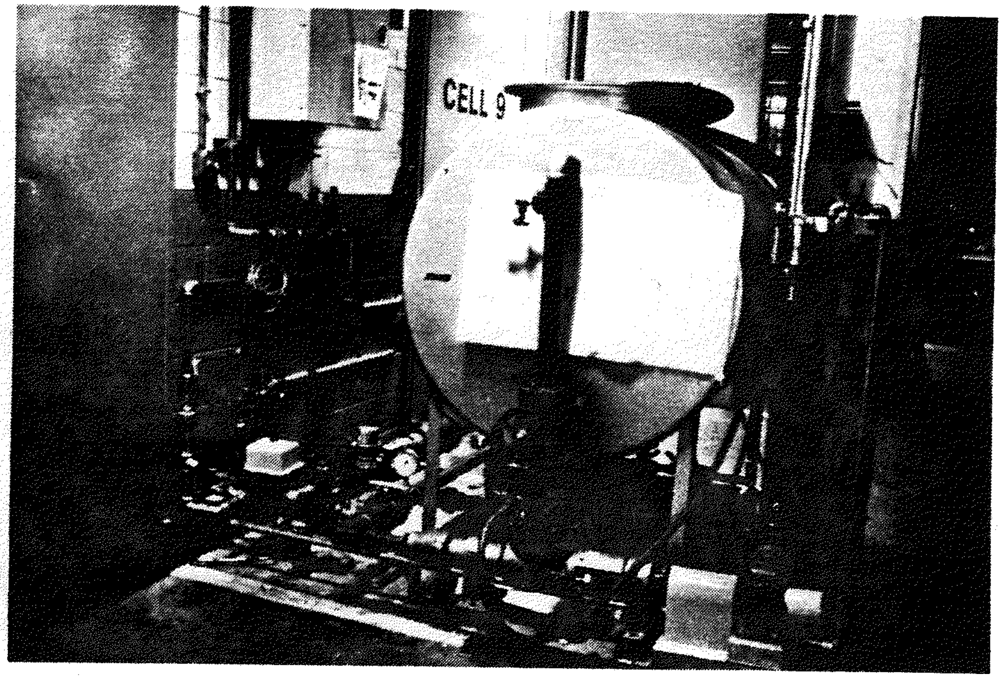

Figure 18. Close-up of the slurry storage system.

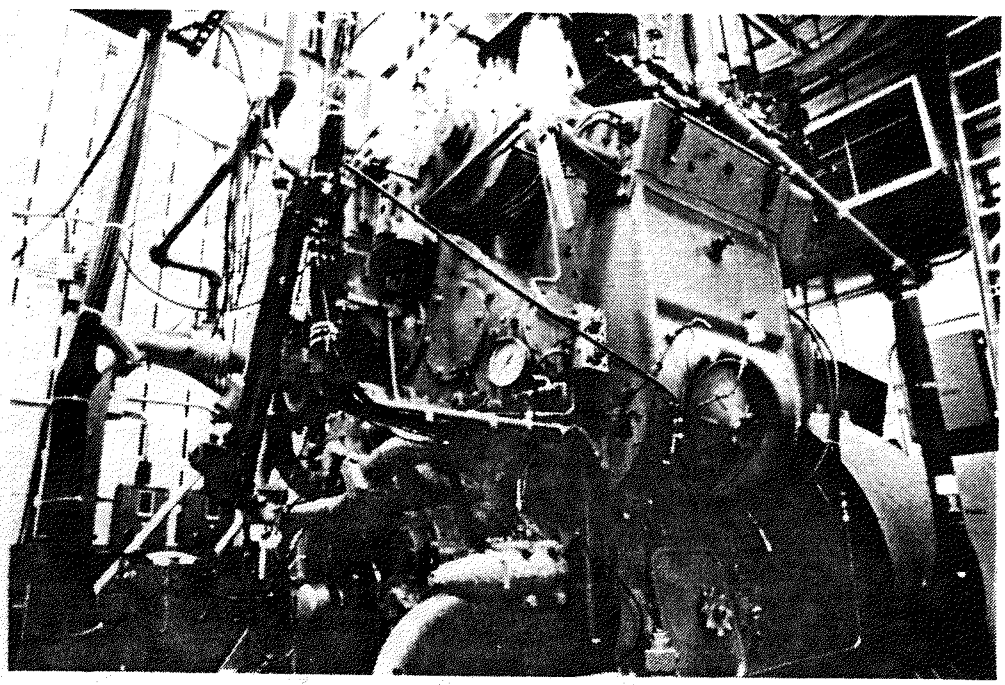

Figure 19. Side view of the 2-645 engine.

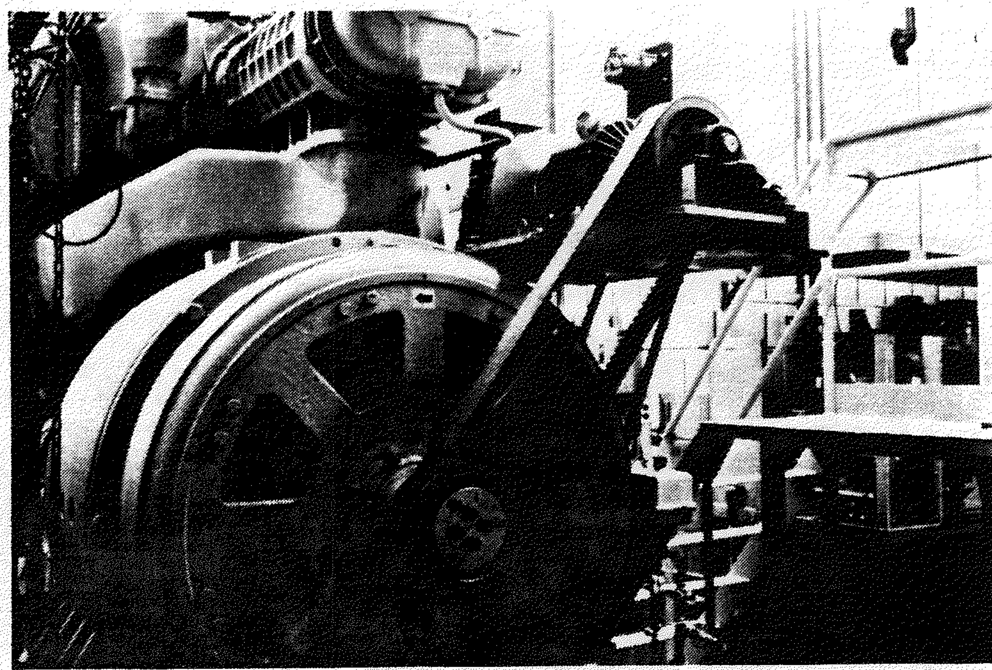

Figure 20. Injector drive mechanism.

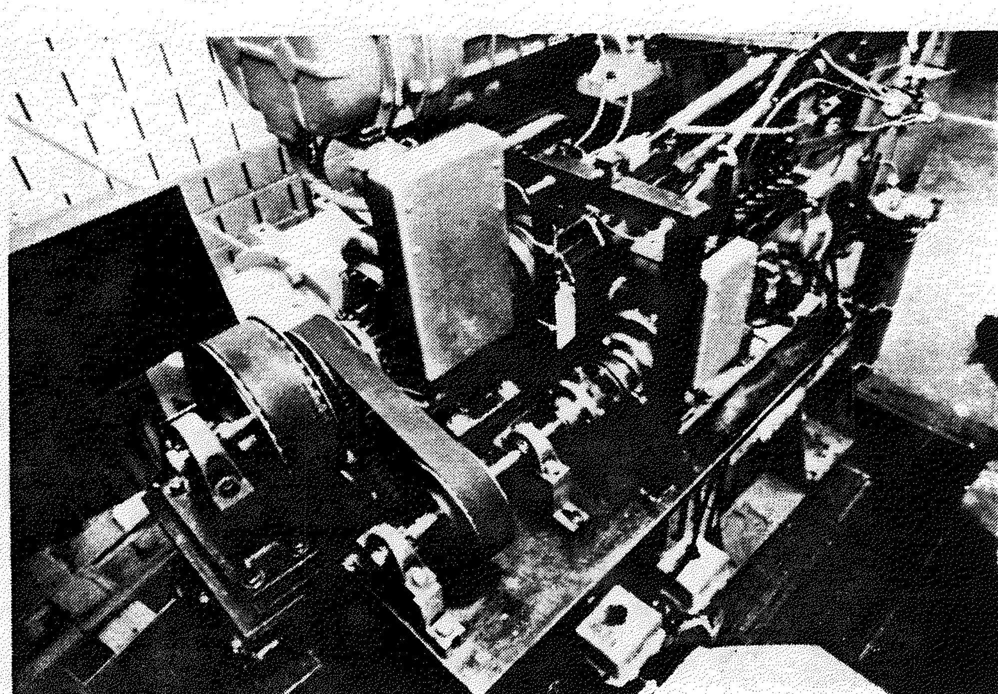

Figure 21. Details of the main injection system.

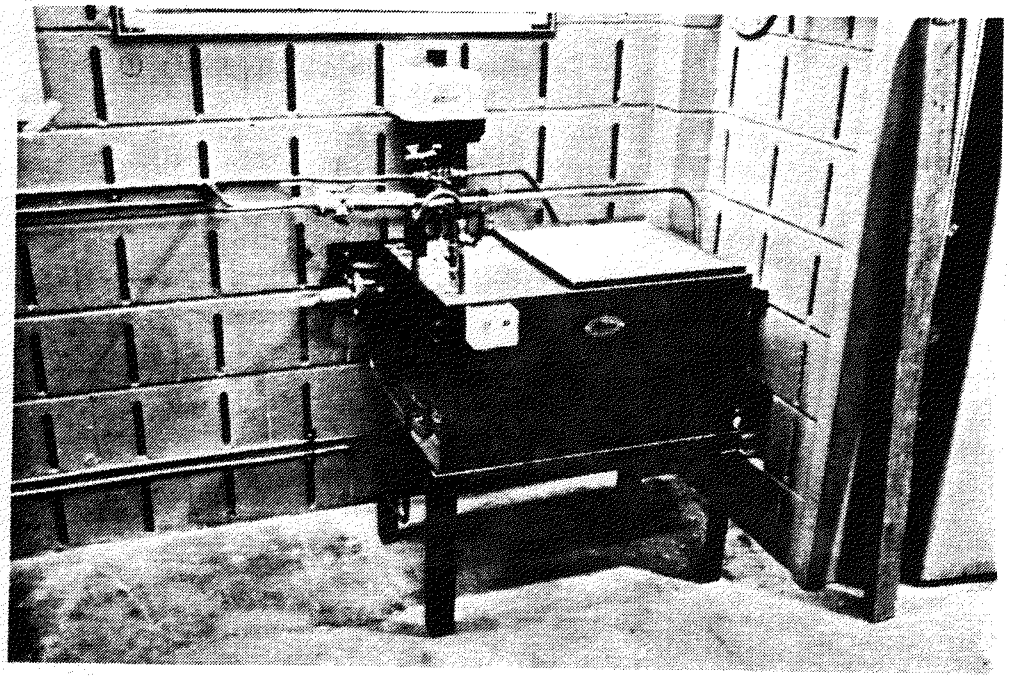

Figure 22. Slurry cooling tank.

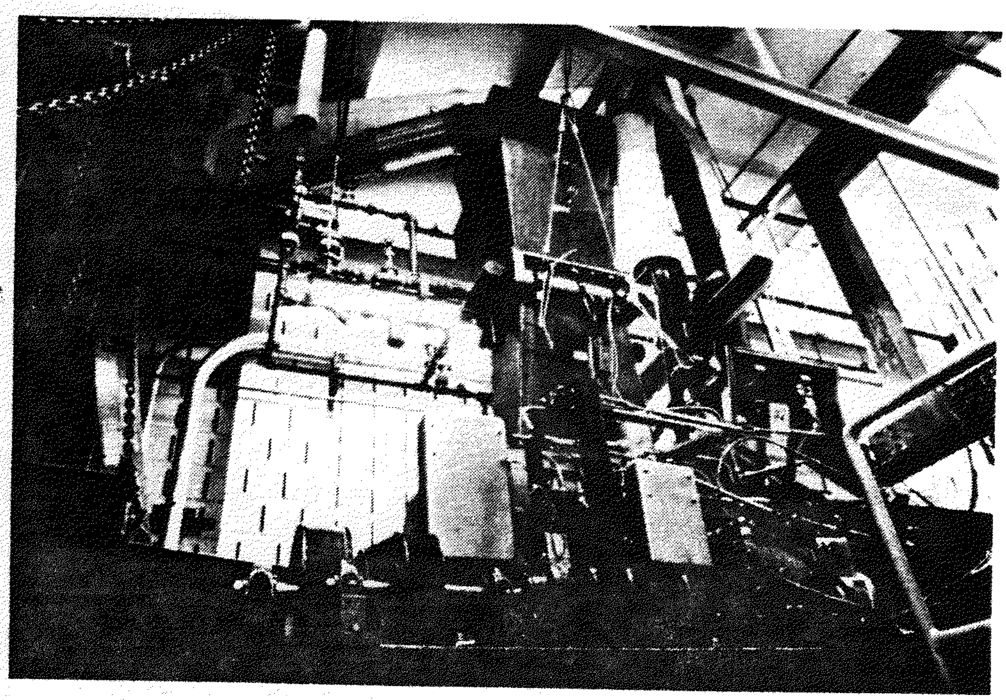

Figure 23. Air inlet section to the engine.

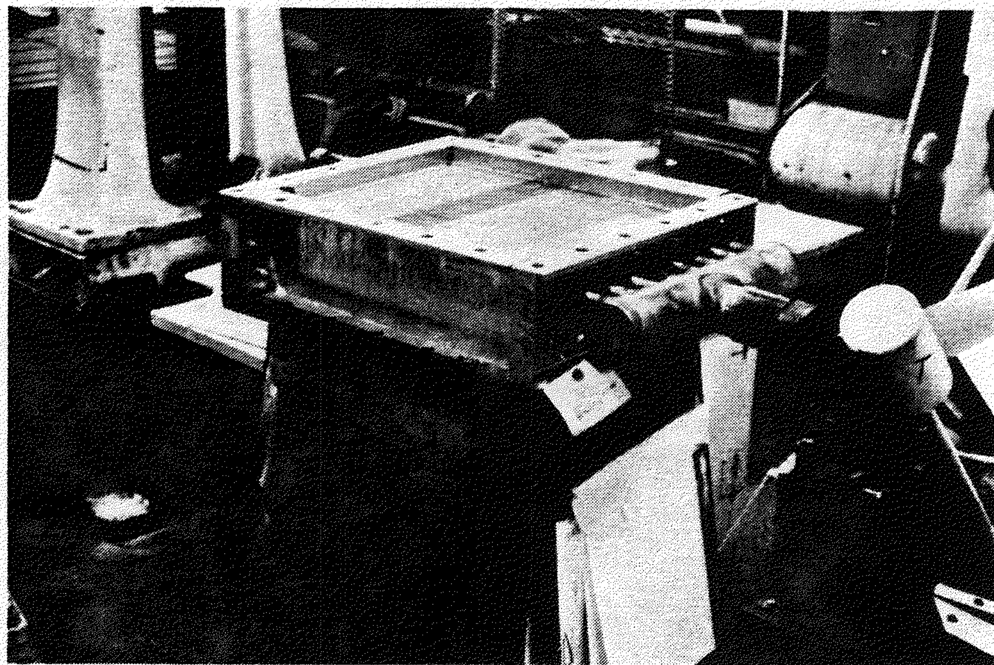

Figure 24. Close-up view of the inlet air heater.

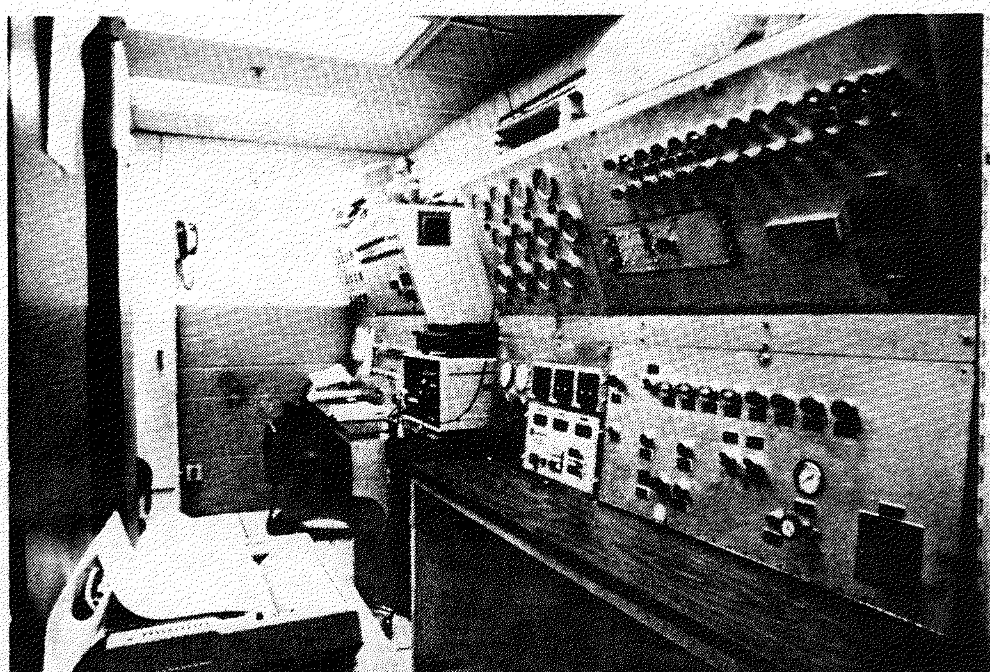

Figure 25. Engine control room.

fluids having so different properties becomes a problem. At the beginning of the test set-up, it was thought that by adjusting the clearances of the injector plunger and bushing this problem could be alleviated as was done by SwRI for unit injectors.

For injection systems with pump and nozzles, it is well known that when the fuel amount is changed, the injection pressure and, to a smaller extent the timing of injection, would also change. In Fig. 26 the pilot fuel injection line pressure traces are shown for three different fuel amounts, namely 100, 150 and 200 mm3/injection, at one operating condition of the engine. As the fuel amount is increased, the peak line pressure is increasing from 5600 to 10,000 psi and the injection timing is moved toward TDC about 2.5 degrees crank angle.

Two types of CWS injection system are used in this study; the main system and the backup system. The main system consists of separate fuel injection pumps for pilot and CWS, with capacities high enough to inject CWS up to full engine load (1500 mm3/injection), have separate controls for both the timing and amount of the fuel injected. This system was based and suggested on the previous CWS test experience at SwRI and described in the previous section in further detail. Although it has some very unique features, it was a new system under development and therefore involves some risks. In order to accomplish the project objectives, a backup system was also foreseen and planned at the test design phase. The backup system consists of the same pilot injector as in the main system, but has the EMD unit injector replacing the CWS injection pump and nozzle of the main system. This is the same unit injector used by SwRI tests, hence usability for CWS injection was proven. On the other hand

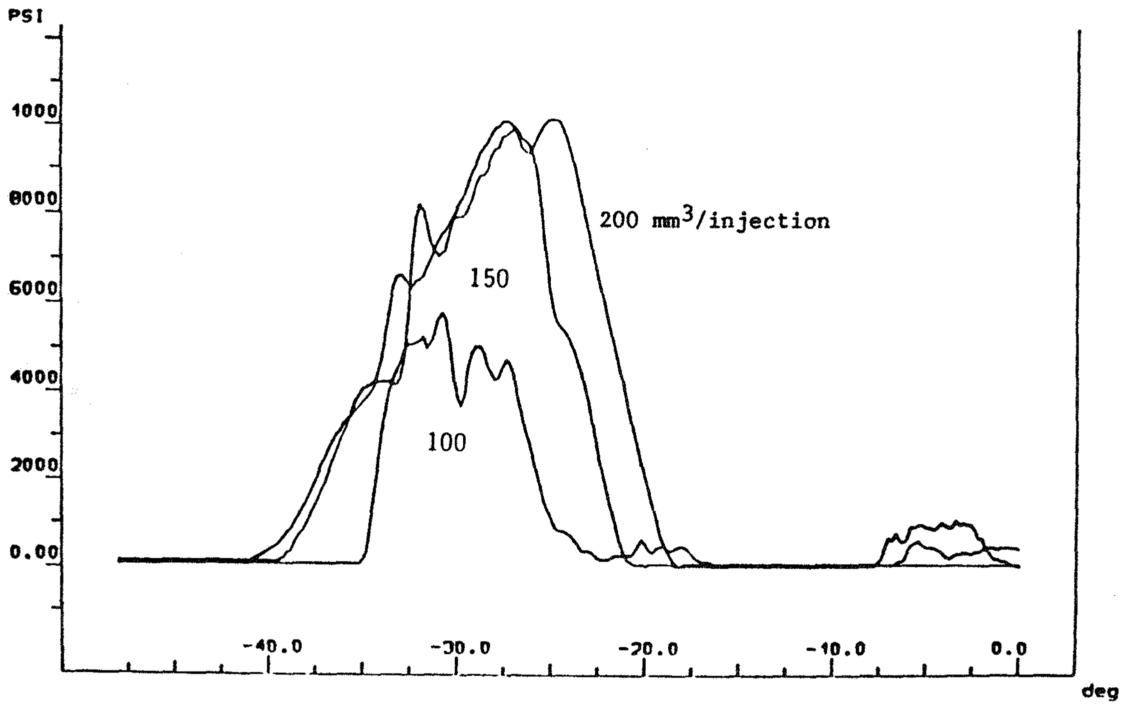

Figure 26. Pilot fuel line pressures for different fuel amounts.

it has two drawbacks. First, the timing of the CWS injection cannot be controlled from the control room, hence it would not represent the locomotive application as close as the main system. Second, the capacity of the injector was limited to about 850 mm3/injection, which would not represent the operation of the engine at full loads but only at partial load due to the heat of combustion of the CWS being about half of the diesel fuel. Although an non-production unit injector with 1400 mm3/injection capacity was also available, it was not modified for CWS operation hence would require further development. As the system was only considered as a backup, the production version of the unit injector was used. The modification of the original production type unit injector for CWS operation was described in the report of SwRI (1) and hence it is not described here again. Only one major modification was implemented. In SwRI tests, the timing of the CWS injection was constant because both the injector and the exhaust valve cams were driven from the same cam shaft. A change in injector cam position by turning the cam shaft would also change the valve timing. In the present case, a mechanical arrangement was designed so that the injector cam can be rotated while the valve cam is held at the same position. Thus, it was possible to change the injector timing while keeping the valve timing constant. Fig. 27 shows the mechanical arrangement used for this purpose.

F. EMISSIONS MEASUREMENT SYSTEM

Measurements of emissions in the exhaust of the CWS cylinder was considered essential to evaluate the effects on the wear rates of liner and ring materials. They were also helpful in evaluating the combustion efficiency of CWS in the cylinder. For these purposes an emissions measurement system was installed and described in this section. Emissions measurements were not taken during the combustion development phase. They will be reported as part of the wear tests.

Emissions measurement equipment consists of two groups.

CO, CO2, HC, and NOX Emissions

Exhaust emission samples for CO (carbon monoxide), CO2 (carbon dioxide), HC (hydrocarbons) and NO-NOx evaluation are taken from the engine's dual fuel exhaust stack via a 316 stainless steel probe. Through a series of sampling lines, pumps, filters, flowmeters as well as a refrigerated water condensing unit, and a low pressure gas purifier, the exhaust emission sample is properly processed for flowing into the test analyzers.

By opening and closing a series of solenoids and Whitney ball valves located throughout the emission test system, gaseous emissions would flow into either the double door rack containing the CO, CO2, NO-NOx analyzers and the HC control module or the single door rack containing the HC analyzer detection and sample preparation modules.

Gaseous and particulate exhaust emissions are to be measured once the engine was in a state (stabilized condition) for a minimum of thirty minutes. The following Beckman analyzers were utilized to determine specific emission component concentrations in the exhaust.

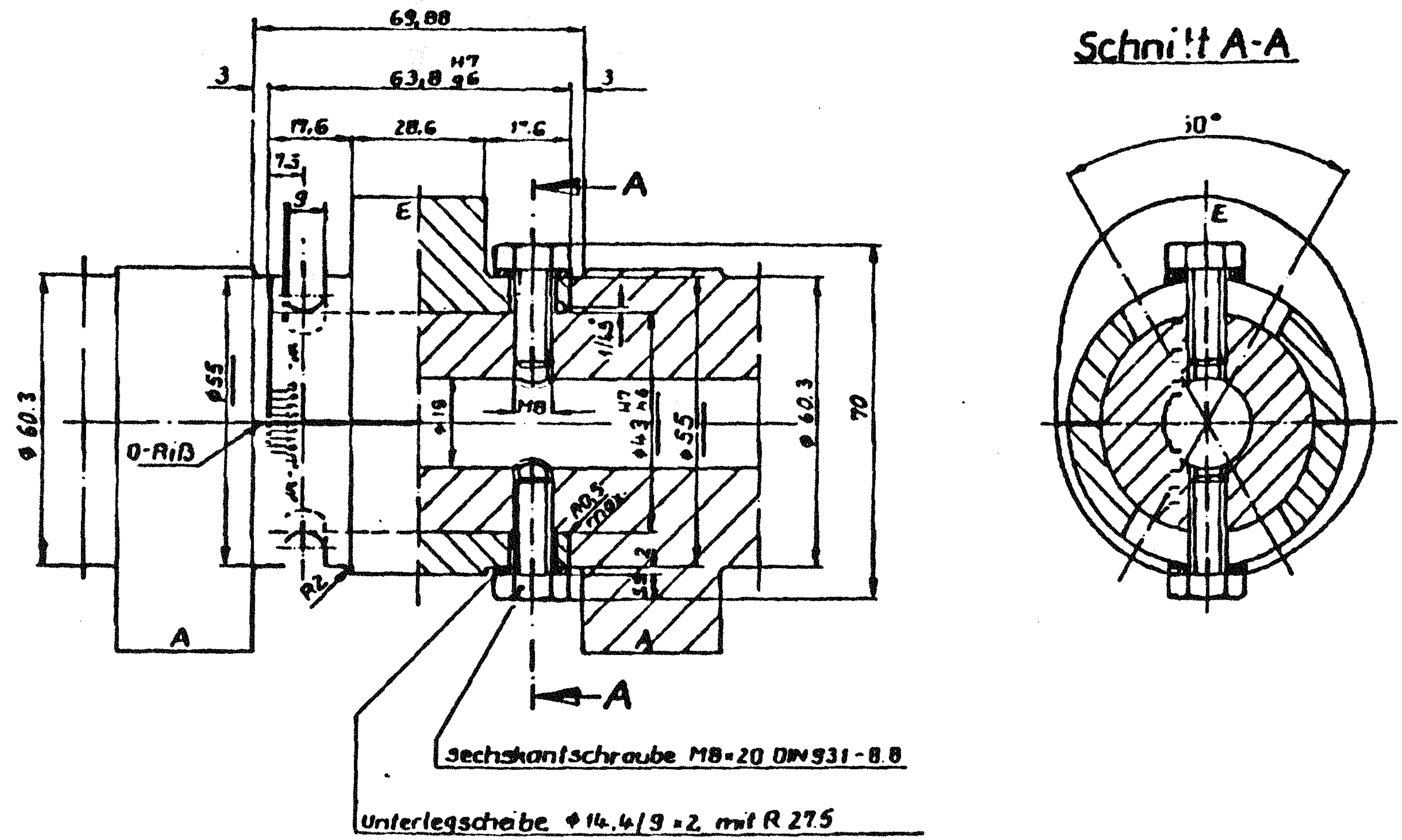

Figure 27. Details of the variable timing cam shaft.

Beckman 867	New dispersive infrared analyzer for CO
Beckman 864	Analyzer for CO2
Beckman 951A	(Combined NO and NO2) for NOx chemilluminescence method
Beckman 404	Flame ionization detection method for HO

Particulate Measuring System

For particulates, a special particulate measurement system developed by General Motors Research Laboratories was utilized. A special probe positioned in the engine's CWS exhaust stack allows an exhaust sample to flow up the heated sample line into the mini-dilution tunnel of the particulate measuring system to produce particulate samples on a specially prepared filter.

The GMR system dilutes only a portion of the total exhaust flow. Air from a low pressure compresssed air source draws the exhaust gas into the minidilution tunnel where the sample mixes with clean dilution air. A portion of the diluted exhaust is withdrawn from the tunnel through an absolute filter using a conventional particulate sampling network.

Mini-dilution Tunnel Procedure

The GMR mini-dilution tunnel procedure was broken down into several steps. To start up the system, it is required that the tunnel and exhaust tube be back-flushed, the background CO2 gas be measured and a specific dilution ratio be set, to establish the correct back pressure once the sample entered the dilution tunnel. A dummy filter is inserted and trial run of the system is conducted to ensure that the system is functioning properly. Once the trial run is successfully completed, a test filter would be inserted in the metal filter holder. Utilizing all components within the GMR system, exhaust emission would have been pulled through the tunnel, particulate matter would have been captured on filter and the filter would have been carefully removed so that no captured particulates would be lost. The test filter will be evaluated using the MCE (methylene chloride extraction) procedure by the Chemistry Lab personnel.

When a new test filter is inserted into the metal filter holder and/or the engine's load/power setting is changed, the entire dilution tunnel procedure is to be repeated.

The procedure requires that all filters are to be weighed prior to baking, baked for six hours minimum at 1000°F, stabilized for a minimum of 12 hr prior to being weighed on a Mettler balance. Filters were to be weighed every two to three hours after baking operation and before actual use on an emissions test.

Stainless steel tubing and fittings as well as a series of switches brings the required compressed gases into the racks and subsequently into the analyzers for the calibration, data taking and purging operations.

4.0 ENGINE TESTS

The test data taken with the engine system described in the previous sections of the report are presented in this section. The basic test data is presented in the form of the CWS cylinder pressure trace as a function of engine crank angle for the test points taken early in the test program. Later, test data points are not presented in a similar fashion due to the volume of the data. On the same figure, the cylinder pressure data of the DF2 cylinder and the injection line pressure traces for the pilot and main injectors are also plotted. Because of the amount of the data, it is given in Appendices A and B. A summary of the some data points is presented in Tables 9 and 10. Using these tables and appendices, different data points can be compared to identify trends with a parametric variation. Tables include the test date, figure number in the appendix, computer file number, pilot and main fuel rack position indicator readings, fuel type injected at the main injector, the exhaust gas temperature from CWS cylinder and short description of the test point. In the appendices, the heat release diagrams, as calculated by the fast heat release calculation procedure to be described below, are also given for some data points. To associate them with their pressure-time figure, the heat release curves are identified with the same figure number with an added suffix B. On the heat release plots there are four curves, the original cylinder pressure trace (in Bars or psia), the calculated cylinder gas temperatures (K) and the instantaneous and cumulative heat release rates (in KJ/KG-Deg and KJ/KG units). In general no smoothing is applied to the pressure trace or the calculated heat release diagrams. In some cases, three point smoothing procedure was applied to the calculated instantaneous heat release curves. In the evaluation of the test results, presented in the main text of the report, the smoothed instantaneous heat release curves are used.

4.1 TEST PLAN

In order to develop information about the essential combustion characteristics of CWS, in a diesel engine combustion environment, the effects of the following parameters are considered important.

a. Ignitability limits of CWS
b. Pilot fuel injection timing
c. Pilot fuel amount
d. CWS fuel injection timing
e. CWS fuel amount
f. Engine RPM
g. Inlet air heating
h. Engine compression ratio

The engine test program is adjusted accordingly. During the study of the ignitability limits of the slurry, the importance of the slurry water evaporation becomes dominant and this event was studied in some detail. The effects of all other parameters were covered, excluding the effects of the engine compression ratio, due to the delays in supplying the special piston designs. That test will be completed in the future.

Table 9.
Test data with original injection system.

Date	Run No.	Fuel Type	RPM	Exhaust Temperature (°F)	Pilot Scale (mm3/injection)	Main Scale (mm3/injection)	Comments
12 Jan 89	1	Water	835		119	186	
12 Jan 89	2	Water	835				DF2 - Full load
12 Jan 89	3	Water	835		119	186	Repeat of #4
12 Jan 89	4	Water	835		119	186	
12 Jan 89	5	Water	835		119	187	Repeat of #4
12 Jan 89	6	Water	835		185	325	Increase pilot
12 Jan 89	7	DF2	835		185	325	Same as #9
13 Jan 89	8	DF2	835	350.2	196	325	Same as #10
13 Jan 89	9	Water	835	197.7	197	325	
13 Jan 89	10	CWS	835	216	197	325	
13 Jan 89	11	CWS	835	221	197	325	
13 Jan 89	12	CWS	835	215.2	197	327	Repeat of #11
13 Jan 89	13	CWS	835	209.3	197	550	Increase Main-Run 5 min.
13 Jan 89	14	CWS	835	208	197	550	Increase Main-Run 5 min.
13 Jan 89	15	Water	835	160	197	550	Lost signal-came back
16 Jan 89	16	DF2	538		313	261	Pilot @+88 CA BTDC
							Main @ 30 CA BTDC
16 Jan 89	17	DF2	741		313	261	Pilot @ 90, Main @ 36
							Pilot @ 98, Main 2 30
16 Jan 89	18	DF2	741		313	261	
16 Jan 89	19	DF2	838		313	261	
16 Jan 89	20	DF2	835		313	261	
16 Jan 89	21	Water	835				
16 Jan 89	22	Water	835	219	489	258	
16 Jan 89	23	Water	835	219	489	258	Repeat of 22
06 Feb 89	24	DF2	835		202	200	DF2 Ref. points
06 Feb 89	25	DF2	835		153	200	DF2 Ref. points
06 Feb 89	26	DF2	835		102	200	DF2 Ref. points
06 Feb 89	27	DF2	835		205	275	DF2 Ref. points
07 Feb 89	28	DF2	836		150	275	DF2 Ref. points
07 Feb 89	29	DF2	832		95	275	DF2 Ref. points
08 Feb 89	30	DF2	835	667	206	289	
08 Feb 89	31	Water	836	375	206	289	
08 Feb 89	32	Water	836	375	206.5	290	Repeat of Run #31
24 Feb 89	33	DF2	835		27	200	
24 Feb 89	34	DF2	835		140	200	
24 Feb 89	35	DF2	835		140	200	
24 Feb 89	36	Water	835		140	201	
24 Feb 89	37	CWS	835		140	201	

Note: Up to RUN #5C, fuel measurements were on the electronic scales. After 06 Feb 89, fuel measurements reported are from calibrated fuel position indicators.

Table 10.
Test data with back-up injection system.

Pilot Amount: 100 mm^3/injection
Main Amount: 210 mm^3/injection
RPM: 834

Time	Run No.	Fuel Type	Peak Pressure (psi)	Exhaust Temperature (°F)	Airflow Rate (SCFM)	Dynamometer Power (H.P.)
9:42:33	.016	DF2	977	557.6	312.4	42.578
9:46:04	.018	Water	1156	318.5	316.0	
9:49:01	.019	Water		255.4	299.8	42.435
9:49:43	.020	CWS	877	267.4	311.0	41.922
	.021	CWS	878	270.0	299.4	42.188
9:52:15	.022	CWS	887	279.7	303.8	42.345
9:53:40	.023	CWS	872	291.9	286.8	42.170
9:55:04	.024	CWS	876	315.8	316.6	42.38
9:56:47	.025	CWS	874	325.8	312.2	42.77
9:58:32	.026	CWS	873	325.1	308.2	42.44
10:01:33	.028	CWS	903	344.1	308.4	42.67
10:03:12	.029	CWS	923	345.5	310.4	42.45
10:04:45	.030	CWS	929	343.7	304.4	42.12
10:07:35	.031	CWS	927	332.9	286.4	41.94
10:09:32	.032	CWS	918	317.6	303.0	42.58
10:12:55	.033	CWS	848	303.2	303.4	42.31
	.034	Water	913	259.2	303.8	42.26
10:20:18	.035	Water	907	259.1	306.8	42.36

4.2 TEST PROCEDURE

During a typical run, the following test sequence is used. After assuring that all modifications are applied and all engine systems are ready for tests, the engine is warmed up for about an hour by using DF2. During this time both pilot and main injectors were injecting fuel. When temperatures are stabilized, the fuel amounts and timings at the pilot and main injector are adjusted to the planned values and the test data is recorded both at the data logger and at the AVL Indiscope 647 at the same time. Usually several cycle averages and more than one cycle were recorded on the indiscope memory for later data processing. The engine operation was monitored through the control panel indicators and the AVL Indiscope. Then the fuel on the main injector line is switched to water. The exhaust gas temperature of the CWS cylinder was used as the indicator to establish completion of the switchover to water injection. It takes about 2 minutes 15 seconds to have stable exhaust gas temperature. The data for water injection is recorded after this equilibrium is established. Then the fuel system is switched over to CWS and the exhaust temperature and cylinder pressure trace was monitored for stable values. Initially, it took several minutes to reach this equilibrium. Later, procedures were established to bleed the water in the system faster and this time is reduced to about one minute. During that time the appearance of the CWS on the drain line is also observed through the control room window. In general, the main injector pump gets stuck during this period when the main injection system was in use. Therefore, there was no extensive and useful CWS combustion data with the main injection system described in section 3.5. With the unit injection system, the engine could reach the steady exhaust temperature, hence acceptable and meaningful CWS combustion data is taken with the backup unit injector system. After taking this data (if the main system is still operational) the timing and amount of the fuel charge is changed and new data point is recorded. If the fuel system gets stuck, it is easily identified from the exhaust temperature, the cylinder pressure and the main injector line pressure traces. In that case, the system is switched to water immediately, and after cleaning the system from the CWS for about 15 minutes, it is switched to diesel fuel.

4.3 PRESENTATION OF TEST DATA

As described before, there were two injection systems used in this study. With the main injection system, CWS combustion is achieved but only for a short duration. Before the exhaust gas temperatures stabilized, the main pump was stuck. Hence the reliable data with main injection system is limited to DF2 and water injection.

There were two different physical phenomena studied through these tests and the analysis will also be presented in two different parallel sections; the slurry evaporation and the slurry combustion.

4.4 SLURRY EVAPORATION EFFECTS

Evaporation of the water content of the slurry injected is a major parameter in controlling the ignition of it. On the other hand, for the development of an engine, operating at a large range of engine speeds and loads, the knowledge of the pilot and main fuel amount and their timings, necessary for the combustibility of the slurry is very important. The optimization of the engine performance at various speeds (and the corresponding loads) would also require detailed information on the ignitability of the slurry. To investigate this phenomena, changing the timing and amount of the fuel injections was considered an asset and an integral part to the design of the test set-up. Previous researchers (15, 16) has shown that for the combustion of slurry, the gas mixture temperature at the point of ignition should be about 800-1000 degrees. As the engine tests with slurry combustion are more difficult, time-consuming and expensive, it was thought that for a given engine operation condition, the minimum amount of pilot fuel needed to ignite the slurry can be estimated by injecting the water into the chamber (instead of slurry) and calculating the gas mixture temperatures from the chamber pressure time trace. As the present test set-up was very suitable to check this concept, the evaporation of water in the cylinder with different fueling conditions was studied in detail. Fig. 28 shows the results of such a study.

In Fig. 28 a, b and c, the pressure time traces of the cylinders are shown for diesel fuel, water and slurry injection at the main injector, respectively. The engine speed, the pilot fuel amount and timing, and the main fuel amount are all kept the same, but only the fuel type is changed.

Here an important feature of the present test set-up needs some explanation. The fueling rate in the second, the DF2 burning cylinder is under the control of the engine governor. Therefore, when the power obtained from the CWS cylinder goes down, the fueling rate of the DF2 cylinder goes up to maintain the engine speed. When the fueling rate of the DF2 cylinder goes up, so does its peak pressure, and associated gas temperatures.

In the tests, presented in Fig. 28, the pilot fuel timing is advanced too much to simulate a condition with no CWS combustion possibility. Following, the figures from 28a to b indicate that when the fuel in the main injector is switched from DF2 to water, the peak pressure of the CWS cylinder goes down while the pressure at the DF2 cylinder goes up, indicating the increase in the fueling rate of the DF2 cylinder to keep the engine speed the same. The main fuel line pressure also changes its shape and duration; this is interpreted as the clearance between the plunger and the barrel of the main injector has different behavior (i.e. amount of leakage) with water than DF2. In Fig. 29 the heat release diagrams obtained from these pressure time traces are shown for both DF2, H_2O and CWS injection. It shows that although the same amount of fuel is injected, the effected heat release in the cylinder is decreasing from DF2 case to CWS to water case, in that order. The reason that the effective heat release with CWS is higher than those of H_2O is attributed to the fact that CWS has less amount of liquid to evaporate, in the form of water and volatiles, and the heating capacity of coal particles is less than the heat of evaporation of water.

The calculated cylinder gas temperatures of these three cases are shown in Fig. 30, indicating that the peak gas temperatures are about 1200, 930 and 890°K respectively. This figure also shows that if the gas temperature at a possible ignition point with CWS is above 800-1000 K limit with water injection, it should also be so with CWS. Thus the estimates of the ignitability conditions from water injection tests should be conservative. Fig. 31 shows the cumulative heat release diagrams for these cases. The near constant portion of this heat release curve, after about 20 degrees ATDC, can be an indication for the completion of the evaporation process. It may also be possible that the amount of heat release is just equal to the amount of heat of evaporation of water at that crank angle so that total heat release would not

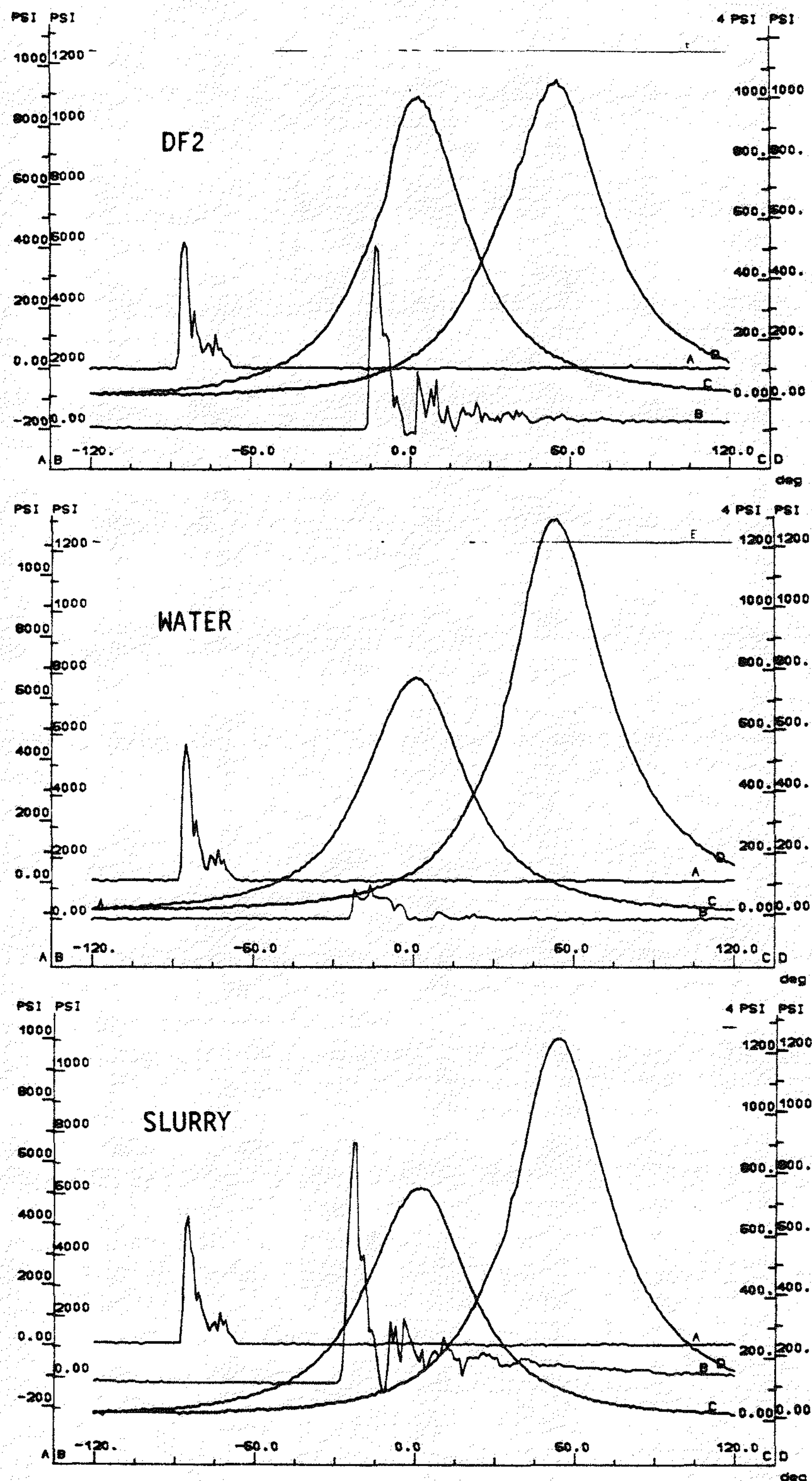

Figure 28. Cylinder and injection line pressures with DF2, water and CWS injection.

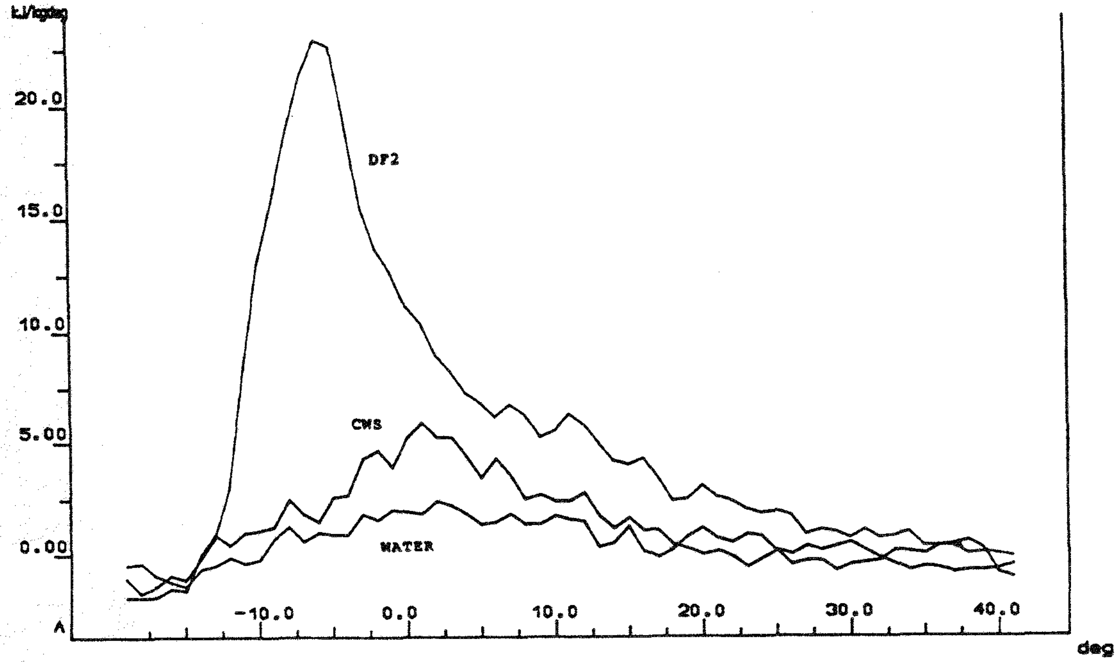

Figure 29. Heat release comparison for DF2, H_2O and CWS injection.

change. For this particular case, it was concluded that water evaporation was completed at about 20 degrees CA ATDC. In Fig. 32, the cylinder pressures around TDC are shown for these three cases. As the pilot injection was advanced too much (about 90 deg CA BTDC) the effects of the pilot fuel combustion is not observable in Figures 29-32. Fig. 32 does not show an increase in the cylinder gas pressure due to CWS injection. From these curves, together with the exhaust gas temperatures, it was concluded that the CWS combustion was not much for this case. A peak gas temperature of about 890°K did not cause CWS combustion. The engine performance for these runs is given in Table 11.

Table 11.
Performance with DF2, H_2O and CWS Injection

Liquid Injected	DF2	H_2O	CWS
RPM	835	835	835
BHP	63.6	Same	Same
CWS Cylinder Exh. Temp. F	350.2	197.7	221.0
DF2 Cylinder Exh. Temp. F	623.6	821.9	750.7
Airflow (SCFM)	312.4	313.6	310.4

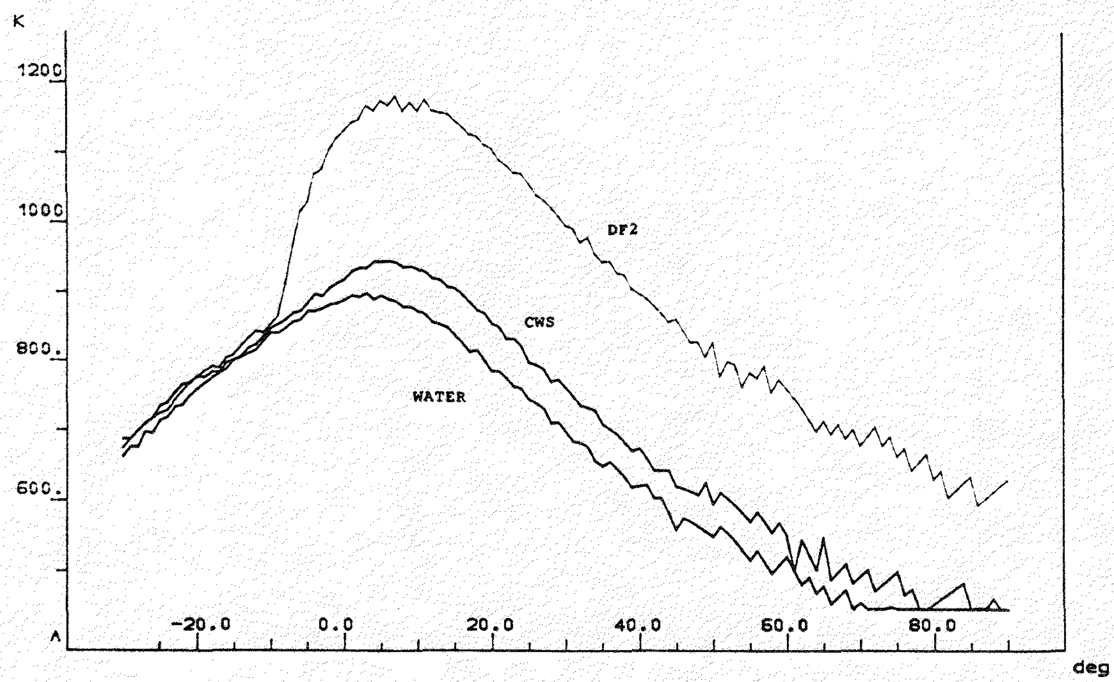

Figure 30. Cylinder gas temperatures for DF2, H_2O, water and slurry injection.

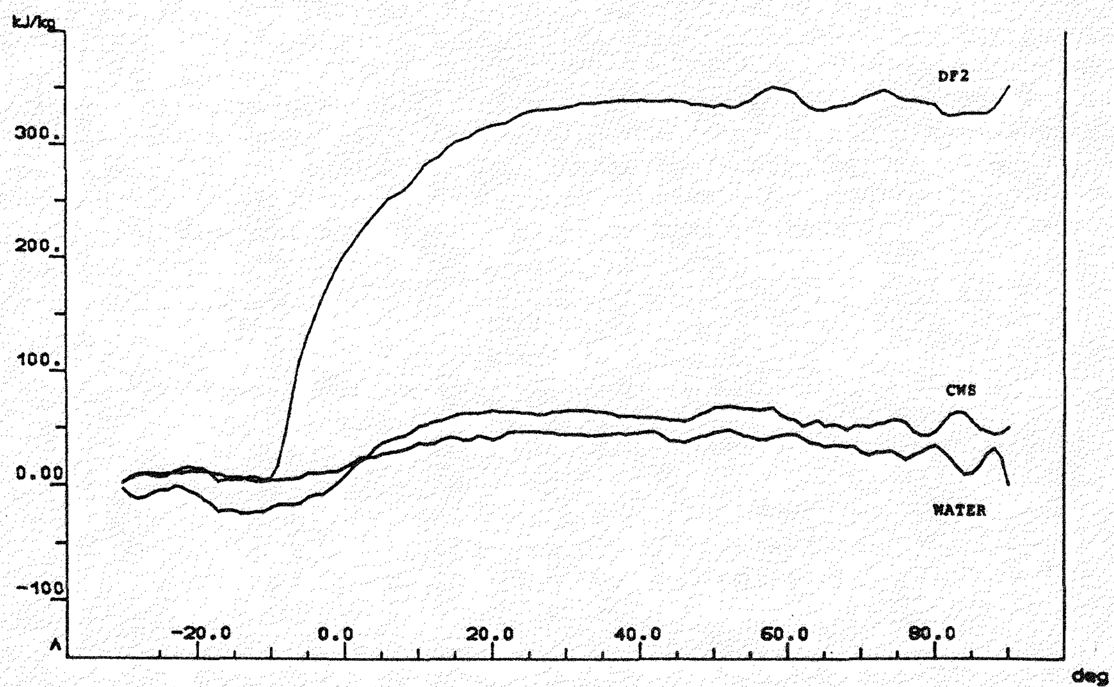

Figure 31. Cumlative heat release for DF2, H_2O and CWS injection.

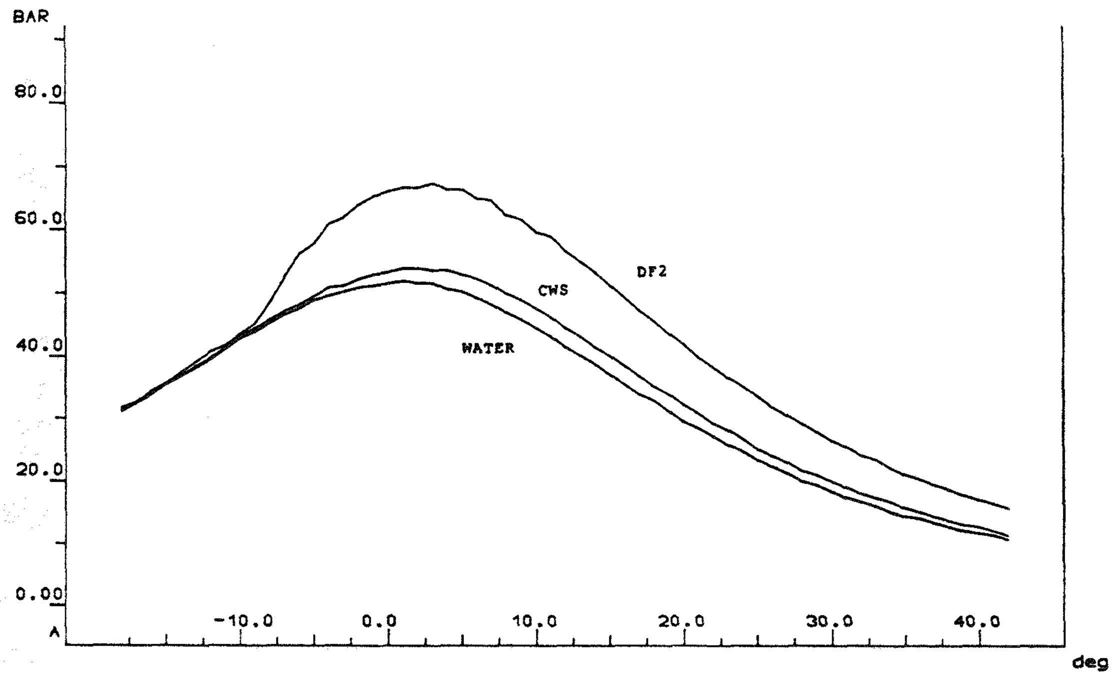

Figure 32. Cylinder gas pressure for DF2, H_2O and CWS injection.

Another example of water injection is given in Fig. 33, showing cylinder and injection line pressures. The calculated instantaneous and cumulative heat release curves and the cylinder temperature plots are also given in Fig. 34, 35 and 36. This operation point corresponds to the operation of SwRI engine, as reported in (17) with 200 mm3/injection main and 100 mm3/injection pilot injection. The results are not comparable directly against SwRI test reports, as the engine strokes and displacement volumes are different. The calculated gas temperatures are above 1000°K in the range of 15 degrees CA BTDC and 15 degrees CA ATDC, indicating the possibility of combustion with CWS. Actual combustion was achieved by SwRI as reported, and indication of combustion by a rapid rise of exhaust temperature was also monitored in our tests before the main injector pump stuck during the exhaust temperature stabilization period. Fig. 35 shows the cumulative heat release plots for DF2 and CWS injection cases. This figure indicates that the evaporation is completed at about 40 degrees CA ATDC and took about 53 percent of the heat released by the DF2.

4.5 SLURRY COMBUSTION

Test results with slurry combustion were made using the backup injection system (unit injector). As the unit injector was used, the line pressure trace for the main injection line is missing in these plots. Test procedure and the numbering system of the figures were the same as explained before. Table 10 shows the basic engine operational information for the data points 38 through 55. Other engine operational conditions which were held constant (approximately) during these tests are given below:

Engine speed	834 RPM
Airbox pressure	2.90 psig
Exhaust pressure	0.48 psig

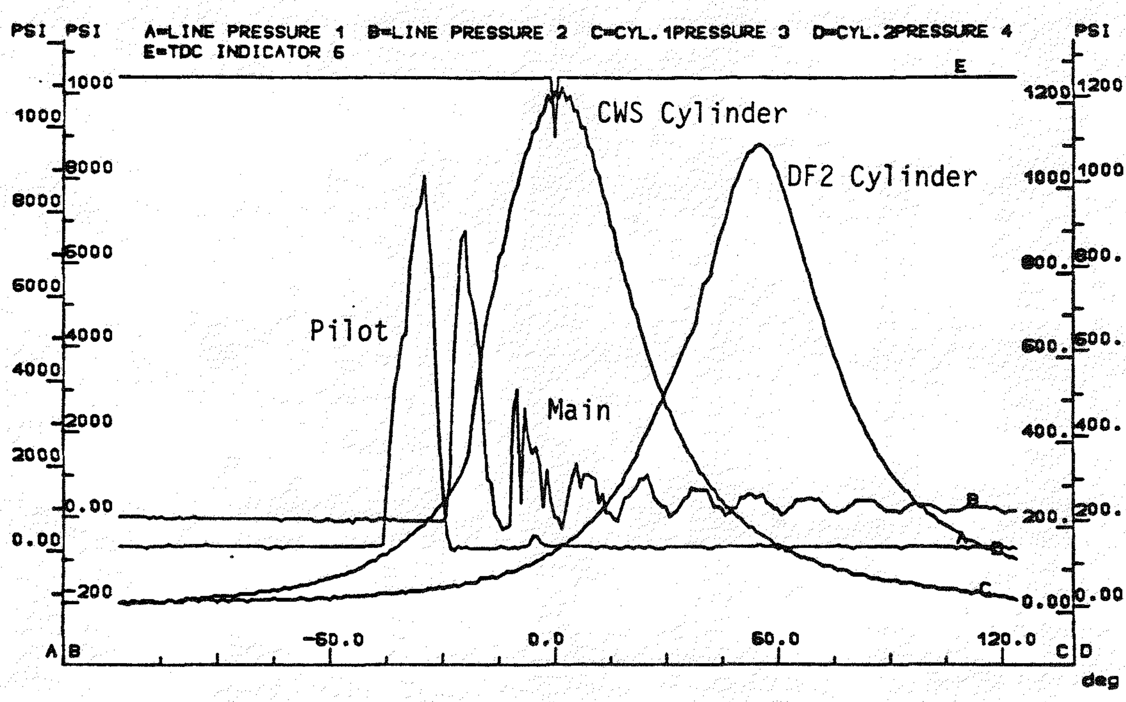

Figure 33. Cylinder and injection line pressures for DF2 injection.

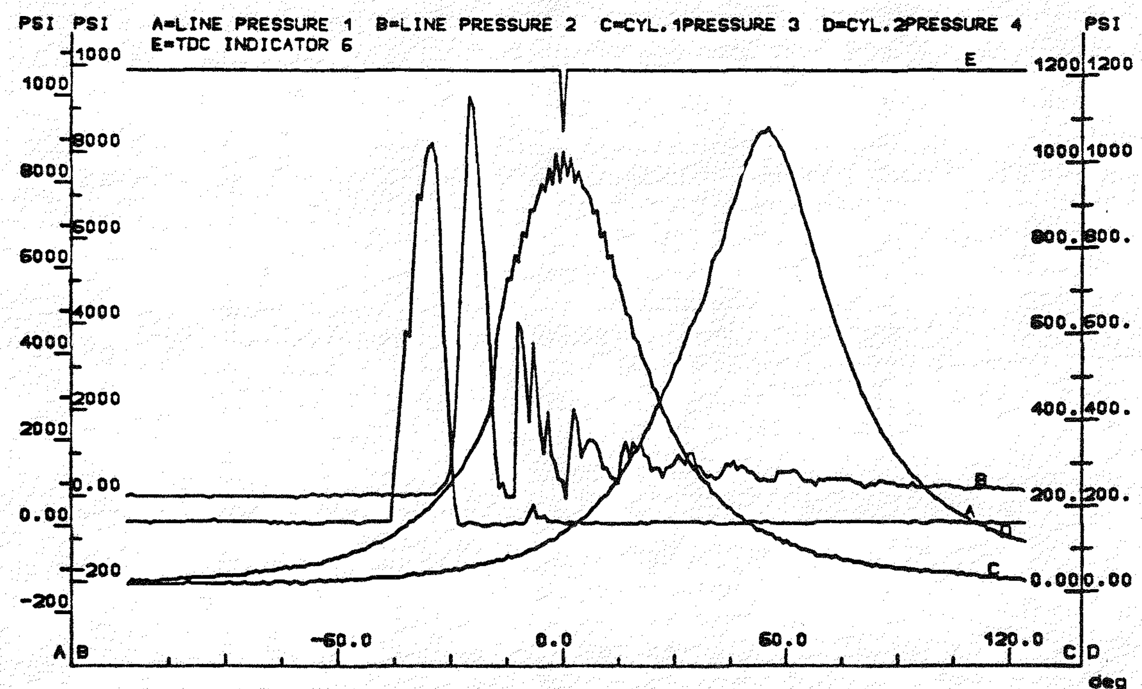

Figure 34. Cylinder and injection line pressures for water injection.

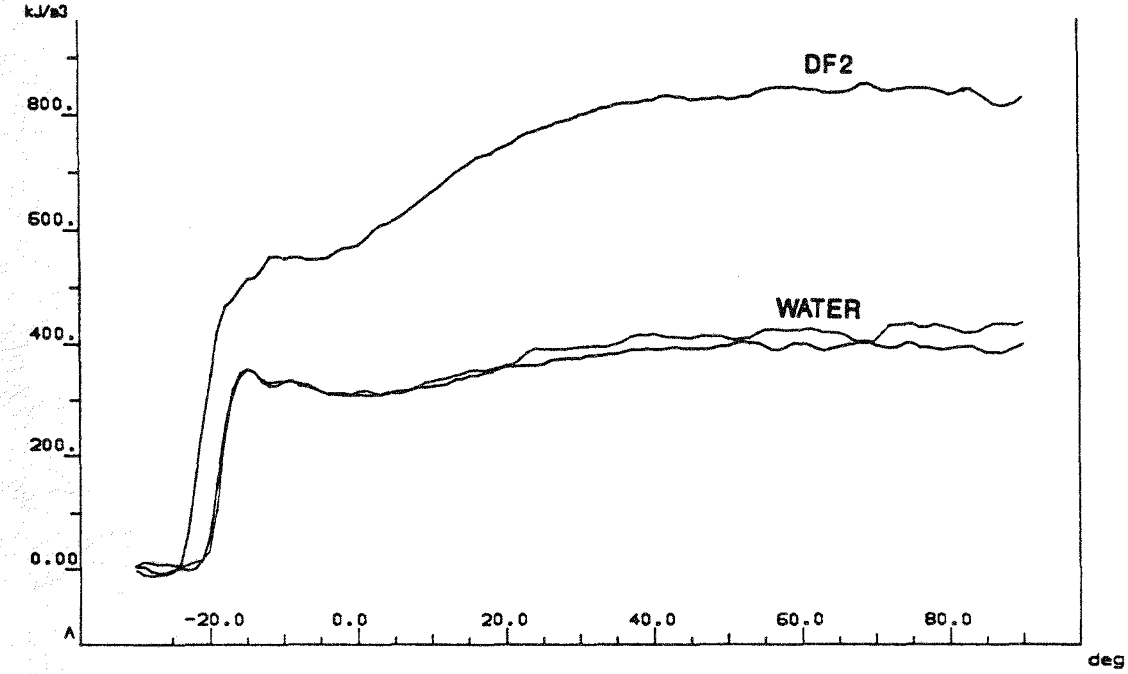

Figure 35. Cumulative heat release rates with DF2 and water injection.

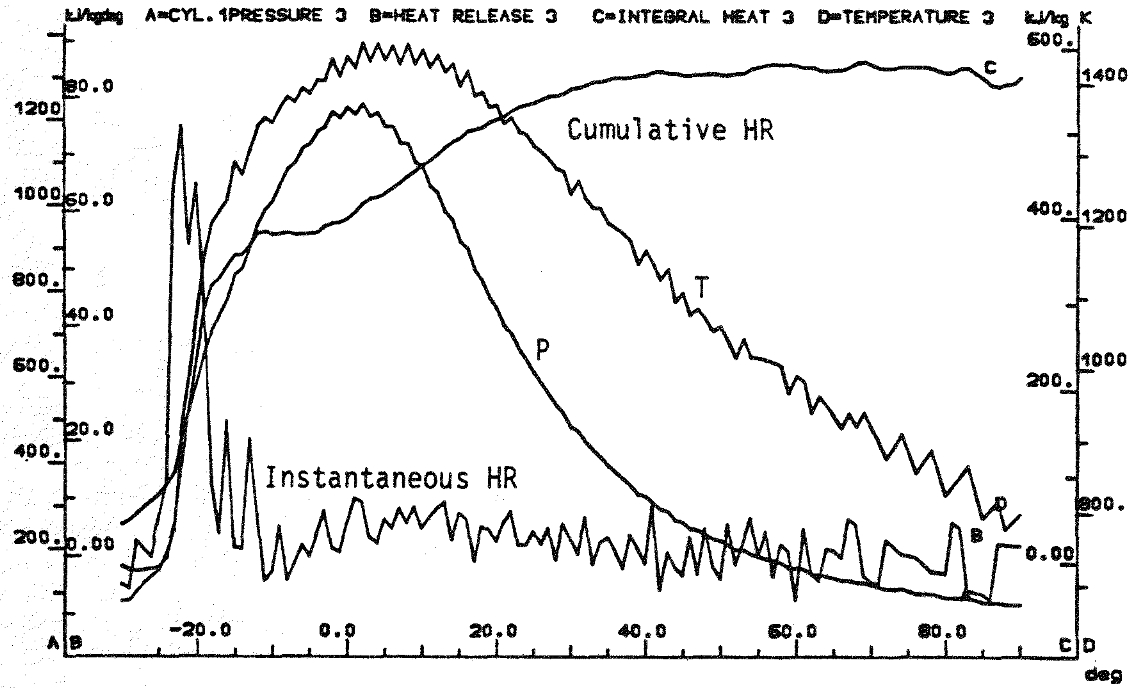

Figure 36. Heat release analysis with diesel fuel injection.

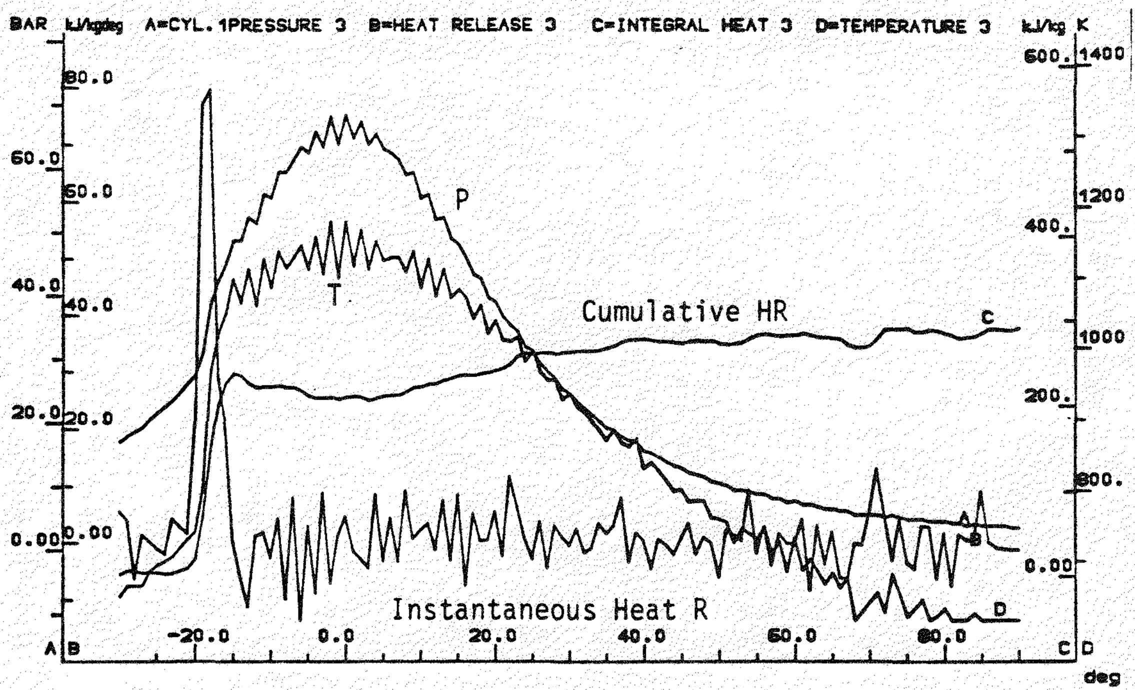

Figure 37. Heat release analysis with water injection.

Airbox Temperature	203 F
Main Injector Fuel	200 mm3/injection
Pilot Injector Fuel	120 mm3/injection

Several important trends are observable from the data presented on Table 10. First, the run time with CWS (from data point 40 to 53) is more than 23 minutes.

A. INJECTOR TIP WEAR

There was no apparent change in the performance of the injector characteristics during this 23-minute test. In order to check this point, the initial data with water injection (Run 19) is repeated at the end of the tests (Run 35). A comparison of Runs 19 and 35 is possible. The transients switching from DF2 to water (Run 16) and from CWS to water (Run 33) had not stabilized to the same exhaust temperature reading and pilot timings are a little different (4 degrees). Comparing of Runs 19 and 35 would give an indication about the change in combustion behavior due to the injector wear for 23 minutes of CWS injection. The PV diagrams of the CWS cylinder is shown for these two cases. The calculated gas temperatures, cumulative and instantaneous heat release rates are given in Fig. 38, 39 and 40 respectively. Although there are some differences, these are mainly due to the inaccuracy in matching the pilot timing; therefore, it was concluded that the difference due to injector wear was not so large to affect the results. Although the CWS injection rate was also low (about 200 mm3/injection, and thus the injector wear is not as high as it would be at full load), the fact that the injector tip survived so long without excessive modification of combustion behavior is encouraging.

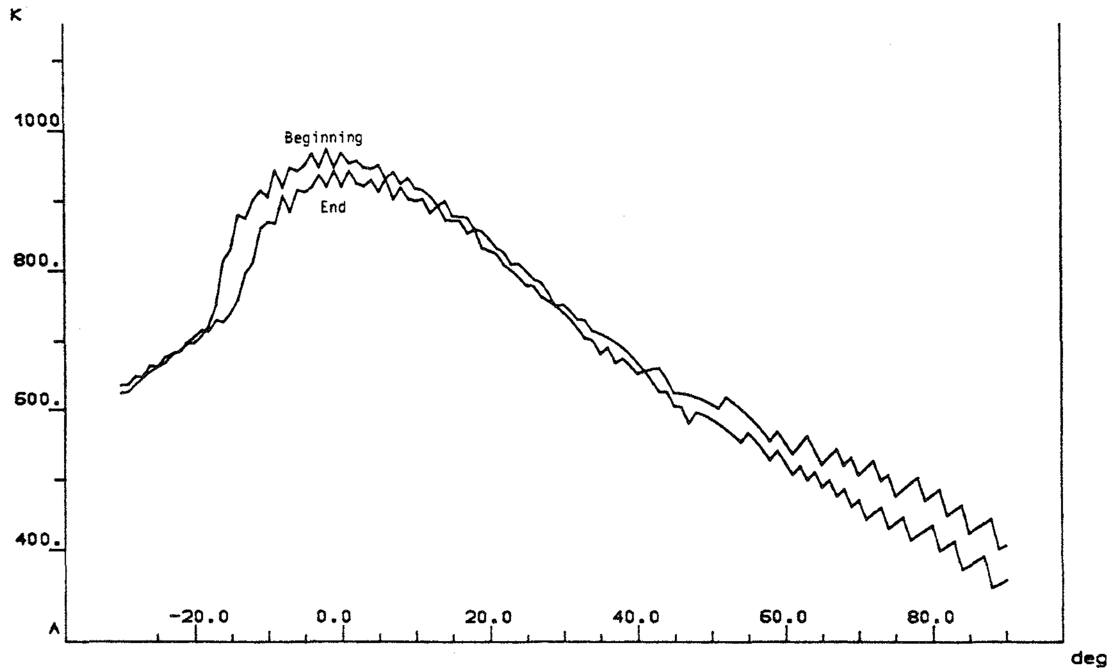

Figure 38. Cylinder gas temperatures with water injection at the beginning and end of 23-minute test. (Note: injection times are different).

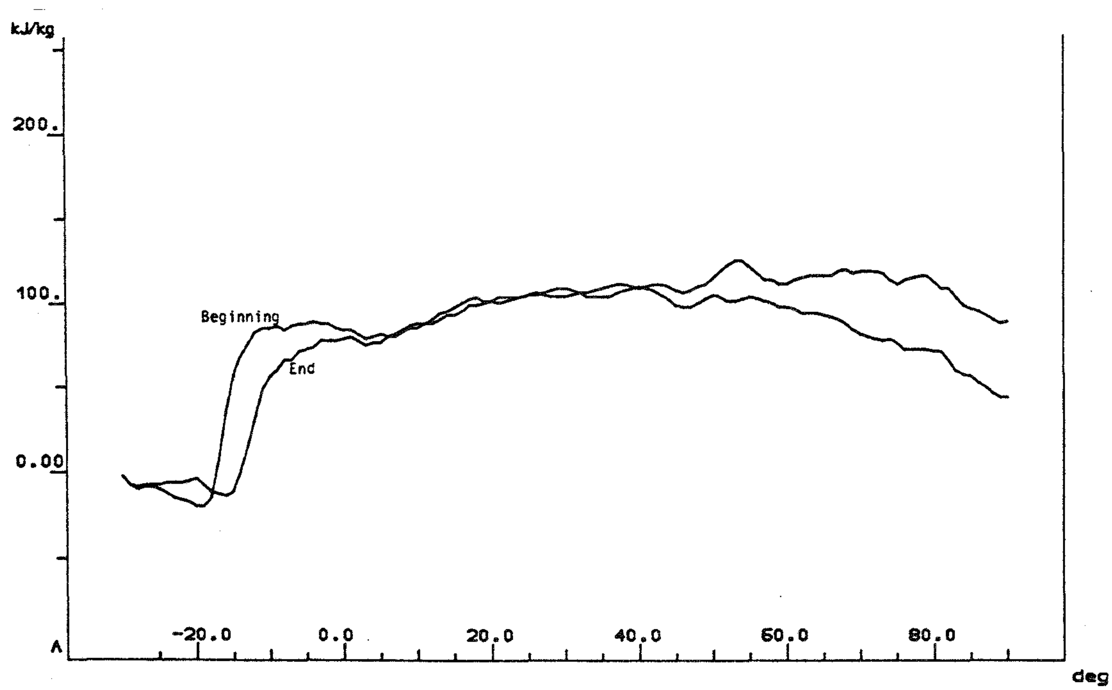

Figure 39. Cumulative heat release curves with water injection at the beginning and end of 23-minute tests. (Note: injection timings are different.

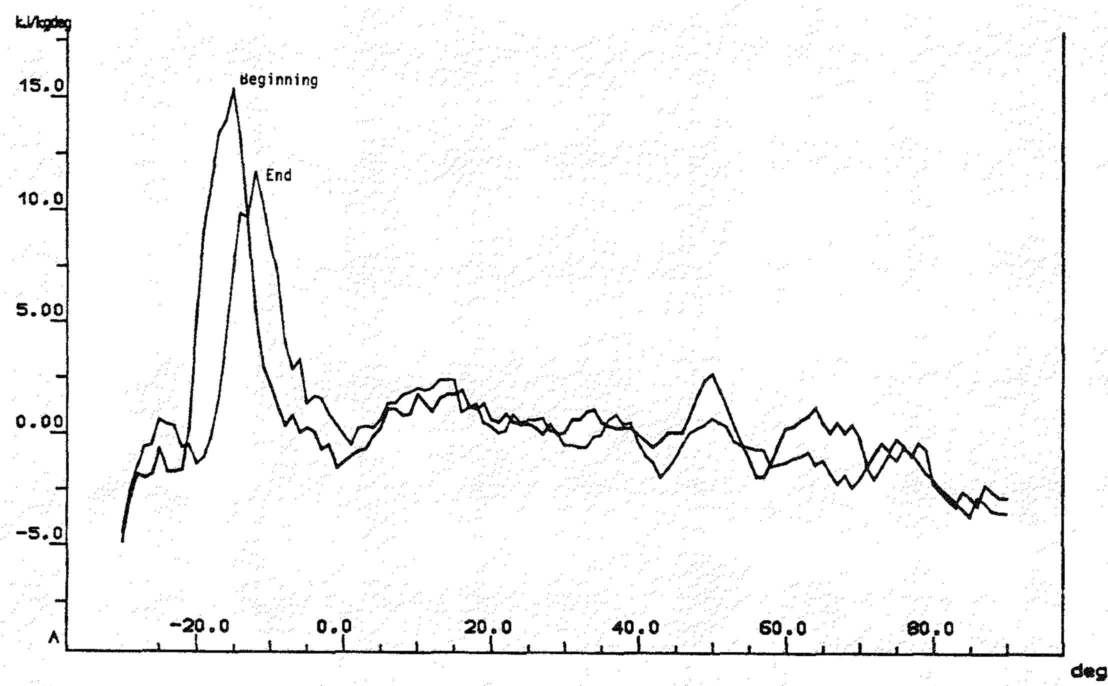

Figure 40. Instantaneous heat release rates with water injection at the beginning and end of 23-minute test. (Note: injection timings are different.)

Additional runs using the same injector tip were successful. On four different dates, the same injector tip was used to get data. The actual test durations with CWS combustion were over 23, 46, 86 and 57 min, which adds up to a total of 3 hr and 32 min of CWS operation. There were several water injection periods between the tests excluded from this time. The tests were not stopped due to any mechanical problems. At the end of the 3-hour, 32-minute testing, the injector plunger seized. During these tests, about 50 percent of the time the CWS injection rate was near the maximum capacity of the injector (about 800 mm3/injection); hence, injection conditions were relatively heavy. The fact that the injector did not fail for over 3.5 hr and did not cause an observable change in the engine output or performance is very encouraging. This is better than expected. There is hope to use the same unit injector design, materials and procedures and improve the injector life by incremental improvements. However, this conclusion is based only on one example. Further injector durability tests and more detailed wear evaluations are essential. The second injector, replacing the first one, also operated over 54 min without any problem.

B. SYSTEM TRANSIENTS DURING SWITCHOVER TO CWS

One of the combustion system characteristics is the transient time for the engine to come into stable operation after the fuel is switched from DF2 to slurry. This is important because a data point taken immediately after this process may not represent the actual operation of the engine when it reaches the steady condition due to temperature transients. In our tests the cylinder behavior is monitored through the exhaust temperature during the transient times from DF2 to water and from water to CWS switching periods. There is an important question about how well the stability of the exhaust conditions would represent the in-cylinder conditions. In order to answer this point, several data points are recorded during the transient period of switching from water to CWS injection. Test numbers 20 through 26 represent these data points. This data shows that under the present operating condition, the exhaust temperature stabilizes in about 8 min 51 sec. This transient time is too long from system dynamics point of view. In Fig. 41, the exhaust gas temperature of the CWS cylinder is shown as a function of time. The total power of the engine was changing a little but almost constant during this test as measured by the dynamometer.

The instantaneous and cumulative heat release rates for test data points 20, 22, 23, 24 and 26 are shown on Figures 42 and 43 respectively. Comparison of the heat release curves shows that during the switching transients, the incylinder heat release is almost completed by Test Run No. 24 or 5.3 minutes after the change. On the other hand, the exhaust gas temperature measurement (Fig. 41) shows that the steady values are reached in 7 min. From these measurements one may conclude that the measurement of completion of transients by the exhaust gas temperatures is conservative. The cylinder heat release pattern is reaching to an acceptable steady-state value in about 75% of the time measured by the exhaust temperature. The system was later modified to reduce transient times to about one minute and the exhaust gas temperature was used to monitor the tests while taking data.

C. COMPARISON OF DF2, WATER AND CWS INJECTION

Another comparison of the combustion phenomena in the cylinder with the diesel fuel, water and slurry injection through the main injector is detailed on Figs. 44, 45, 46 and 47 in terms of cylinder pressure, cumulative heat release, instantaneous heat release and cylinder temperature. In these tests the engine speed was 834 rpm, the pilot and main injection fuel amounts were 100 and 210 mm3/injection respectively. The percentage of the energy of combustion from the pilot charge is about 50 percent.

In Fig. 44, the cylinder pressure time traces are shown for these three runs showing that the cylinder pressure level is higher for DF2 case than CWS which itself is higher than those of the water case, during the expansion period. It also shows that the peak pressure with DF2 is at later crank angles and higher than the other two. The peak pressure levels with CWS and water injection is almost similar. The pressure time traces do not differentiate the CWS and water injection cases appreciably at this low fueling rates. The cumulative heat release rates are shown in Fig. 45 showing the five separate regimes, namely (a) pilot combustion, (b) water evaporation, (c) slurry combustion, (d) no combustion and (e) late water evaporation periods, to be described in Fig. 49. It also shows that the energy released by DF2 case is much larger than the CWS case, about 52 and 18 KJ/kg respectively. The energy ratio is about 2.90 while the energy content ratio of DF2 over slurry was about 2.1. This is an indication that the efficiency of slurry combustion is much less than those of DF2 under this low fueling rates. The exhaust gas temperatures for these cases were measured to be 557.6, 255.4 and 325.1°F respectively for the DF2, H_2O and CWS cases.

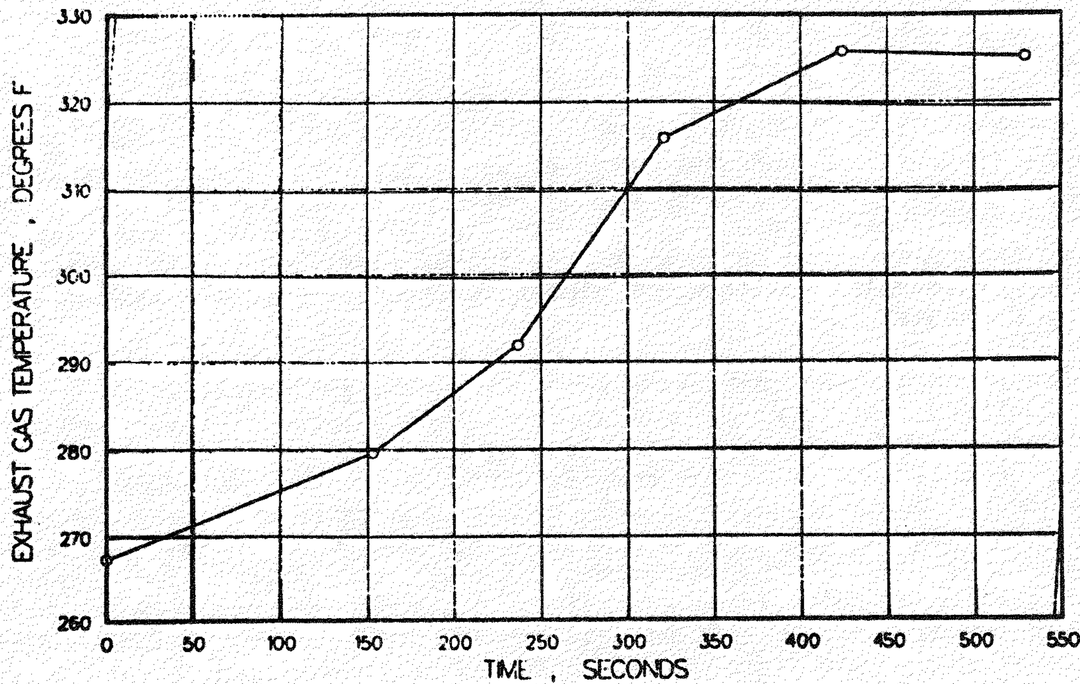

Figure 41. Exhaust gas temperatures during switchover period.

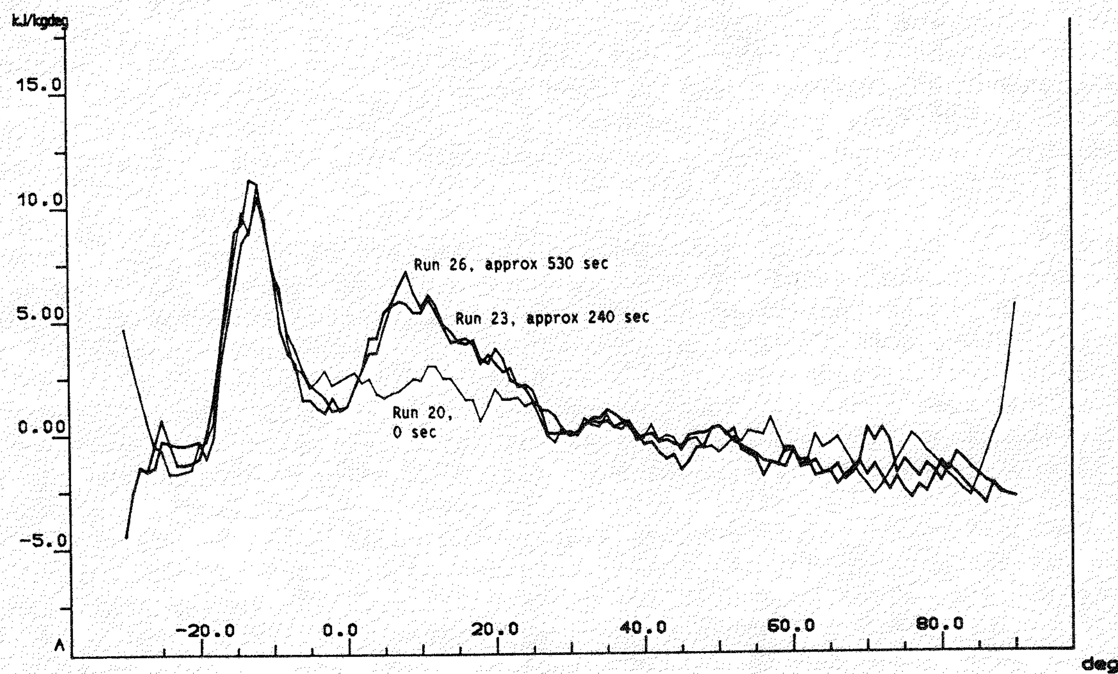

Figure 42. Instantaneous heat release rates during switching transients.

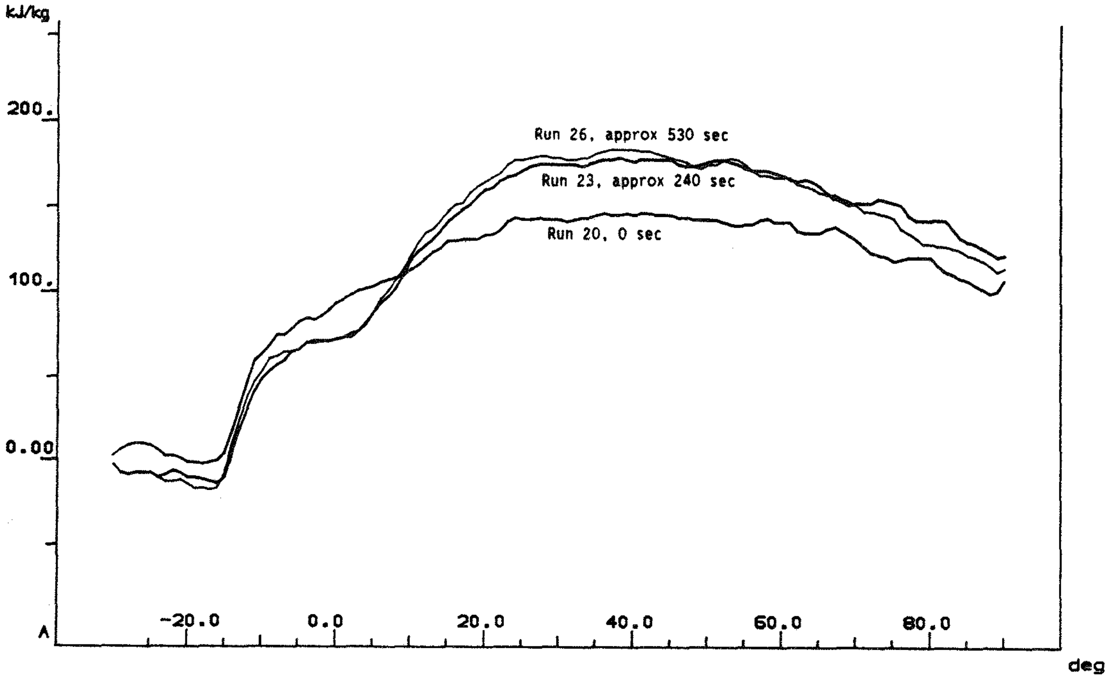

Figure 43. Cumulative heat release rates during switching transients.

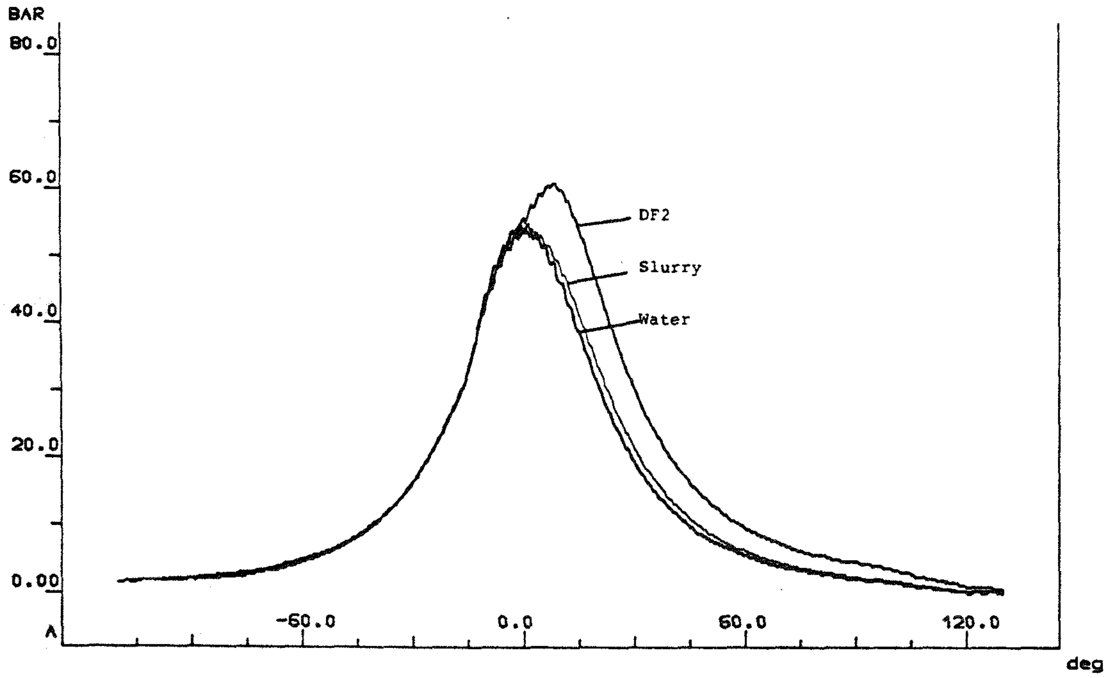

Figure 44. Cylinder pressure curves with diesel fuel, water or slurry injection.

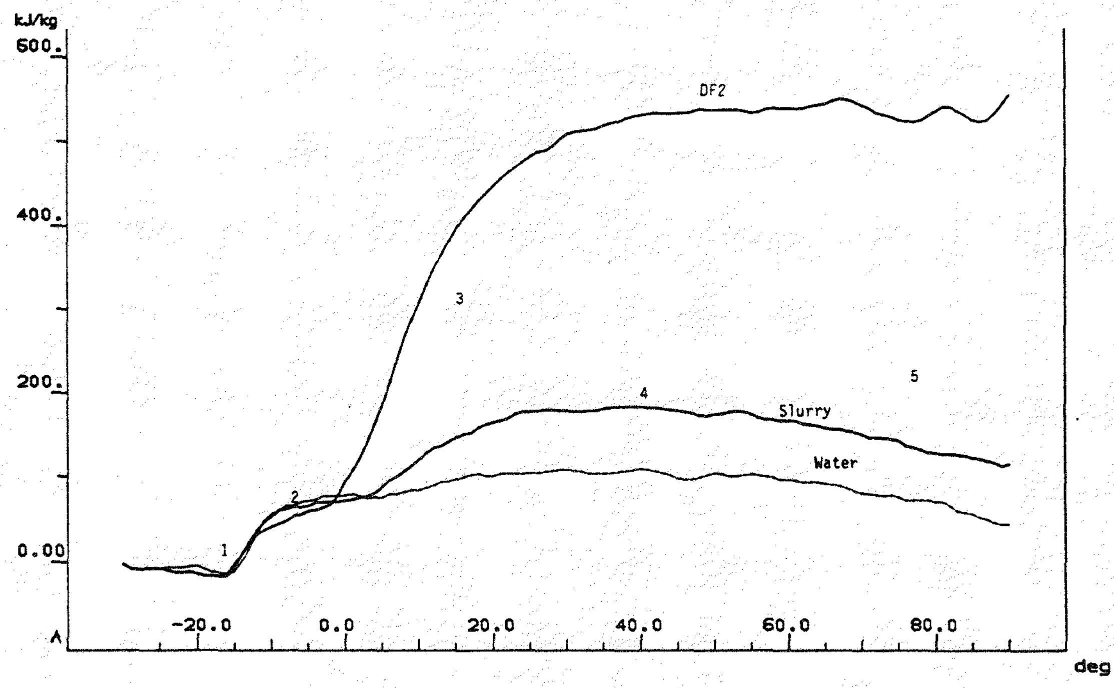

Figure 45. Cumulative heat release curves with diesel fuel, water, or slurry injection.

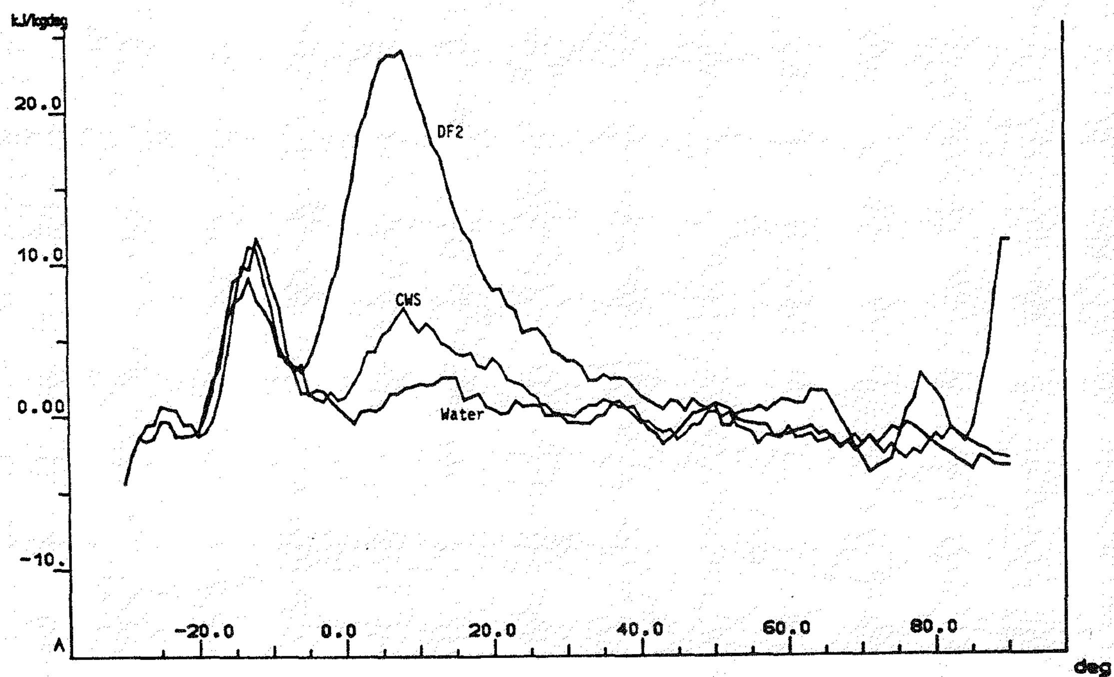

Figure 46. Instantaneous heat release rates with diesel fuel, water, or slurry injection.

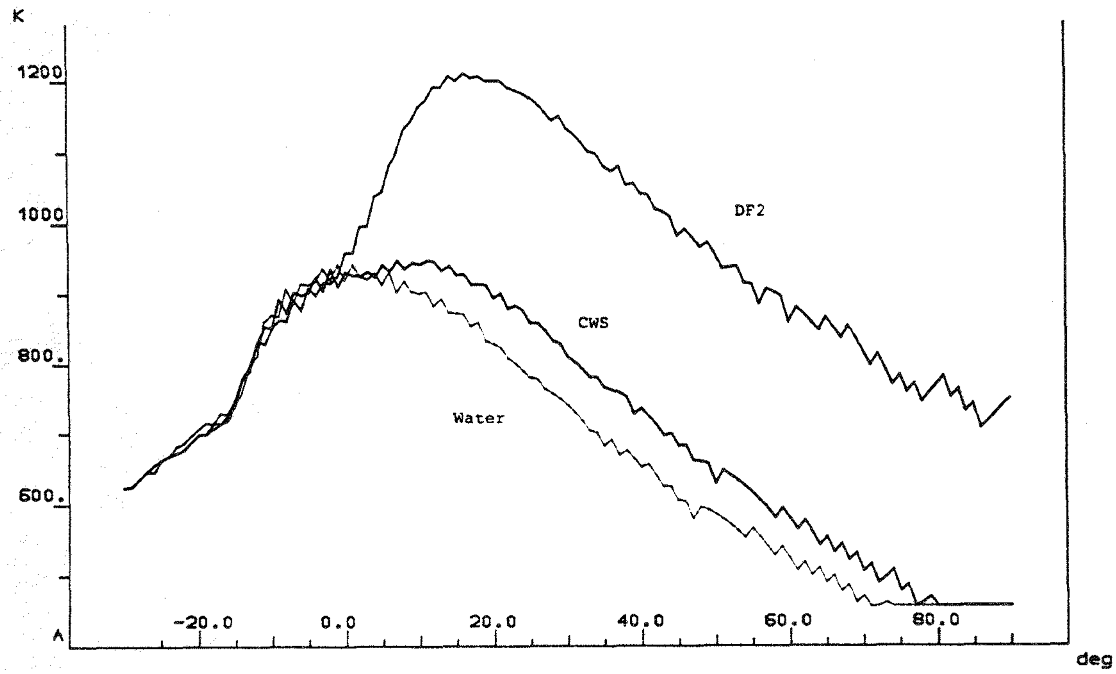

Figure 47. Cylinder gas temperatures with DF2, water, or slurry injection.

The instantaneous heat release rates for these cases are given in Fig. 45, indicating similar trends and the five different regimes to be discussed for Fig. 48 in detail. The combustion of DF2 fuel injected by the main injector has a much higher peak than those of the CWS case. The crank angle for the peak combustion rate moves to later crank angles with CWS injection which is attributable to the increase of ignition delay with slurry. The portions of the instantaneous heat release curves beyond about 55 degrees CA ATDC is oscillatory in nature and may not be as reliable. These heat release calculations are plotted by smoothing the calculated instantaneous heat release with a three-point smoothing algorithm.

The in-cylinder gas temperatures are shown in Fig. 46. As expected, the cylinder temperatures with DF2 is higher than those with CWS which itself is higher than those of water injection case. The peak temperature with DF2 is much higher then the other two. The peak temperatures with slurry is almost the same as with those of water injection. There can be two factors for this. First, the amount of slurry fuel injected is small, the total fuel burned is about 25 to 30% of the maximum load case. Second, the water evaporation and late and separate injection of the CWS fuel (a characteristic feature obtained by dual fuel injection) could cause combustion at almost constant gas temperatures. As discussed before, this constant temperature heat addition may be useful in increasing the engine efficiency, if the energy can be recovered by the piston appropriately.

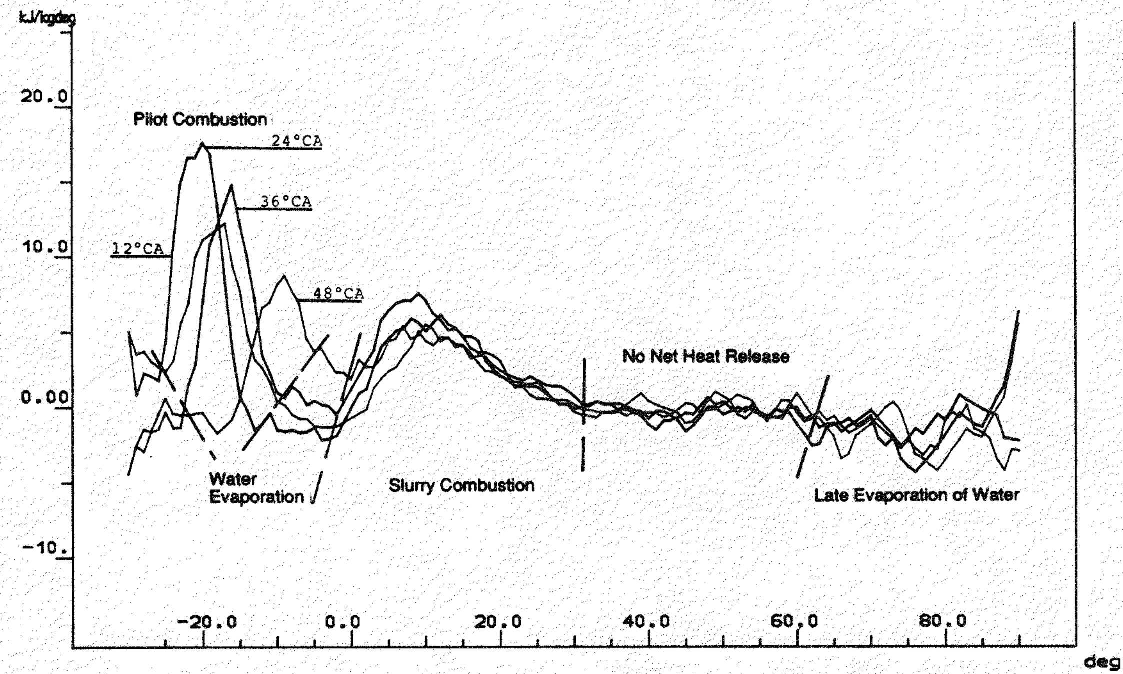

Figure 48. Instantaneous heat release rates for a different pilot timing.

D. EFFECT OF PILOT TIMING ON CWS COMBUSTION

The effects of the pilot timing on the combustion within the CWS cylinder is studied for several cases as it is one of the key parameters having extensive influence on the ignitability of the slurry. It was also relatively easy to change the pilot injection timing with the present test setup. In the following paragraphs, two different cases are presented: (a) High pilot/slurry ratio case, and (b) Low pilot/slurry ratio case.

High Pilot Slurry Ratio Case

In this case the relevant values of the test parameters were as follows:

Pilot Fuel Amount	:	100 mm3/injection
Main Fuel Amount	:	210 mm3/injection
Engine RPM	:	50%
Pilot Injection Start	:	834 RPM
Main Timing	:	40 Degrees BTDC

The results are plotted in Figures 48 through 51, for the cumulative and instantenous heat release, and the cylinder pressure and temperatures. The following trends are observable in these plots when the timing of the pilot injection is advanced:

a. There are two peaks on the instantenous heat release curves presented in Fig. 48. These peaks correspond to the rapid rise of the cumulative heat release curves in Fig. 49. The first peak (or rise) is due to the combustion of pilot fuel while the second is due to the combustion of coal water slurry.

b. With an advance in pilot timing, the heat release of the pilot fuel occurs at earlier crank angles. The peak of this heat release advances almost the same amount of the timing advance. The pilot heat release peaks covering a narrower range of crank angles. The valley between two peaks goes further down (in magnitude), indicating less interaction between the fuels injection. For 30 degrees advance case, the heat release between the peaks becomes negative (also the slope on Fig. 49 becomes negative) possibly an indication for the presence of slurry evaporation.

c. The maximum of the cumulative heat release curves changes with pilot advance. As the advance is increased, this maximum increases first, goes through a peak and then decreases a little (Fig. 49).

   In Fig. 49 there are five different regions identified such that each region is dominated by a particular event. These regions may be indicated as: (a) Pilot Combustion, (b) Water Evaporation, (c) Slurry Combustion, (d) Zero Net Combustion and (e) Late Water Evaporation regions. The limits of these regions are also indicated on Fig. 48. In the no combustion region, it is possible that the latent heat of water evaporation is just equal to the heat of combustion so that the instantaneous heat release is zero and cumulative heat release is constant. In the region called the late evaporation of water, it is most likely that the heat of fuel burned in that region is less than the latent heat of evaporation of the water and hence the instantaneous heat release is negative and the cumulative heat release curve is decreasing.

   Fig. 49 clearly indicates that the lines separating the slurry evaporation and slurry combustion regions and similar line between the slurry combustion and zero combustion regions are not affected by the advance of the pilot injection. On the contrary, the advancement of the pilot timing advances the start of pilot combustion and water evaporation. If we generalize the trends, the advancement of the pilot timing does not affect the start and the end of slurry combustion but it affects the timing of the pilot combustion, as well as the slurry evaporation under the condition of this test.

   If the crank angle at the end of the slurry combustion rise is taken as an indication of the completion of slurry combustion (or a major portion of it), then Fig. 49 shows that it is completed at about 30 degrees crank angle after TDC.

d. The cylinder peak pressure (Fig. 50) increases with an advance of the pilot timing. It also causes an increase in the cylinder pressure before TDC, still during the compression period, which is detrimental to the power recovery of the piston and the engine thermal efficiency.

e. The cylinder gas temperature increases with an advance of the pilot timing (Fig. 51). The five different regions identified on the heat release figures are also shown on Fig. 51. Due to the earlier combustion of the pilot fuel, the gas temperatures increases rapidly, starting at earlier crank angles. The evaporation of slurry water decreases the rapid increase

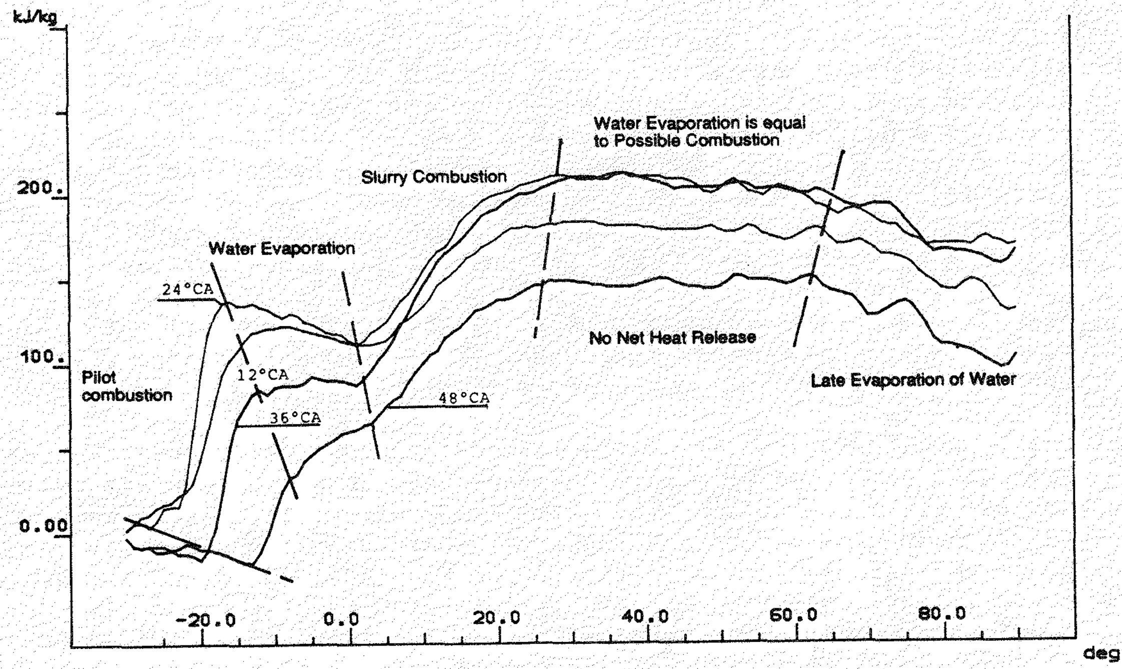

Figure 49. Cumulative heat release rates for different pilot timing.

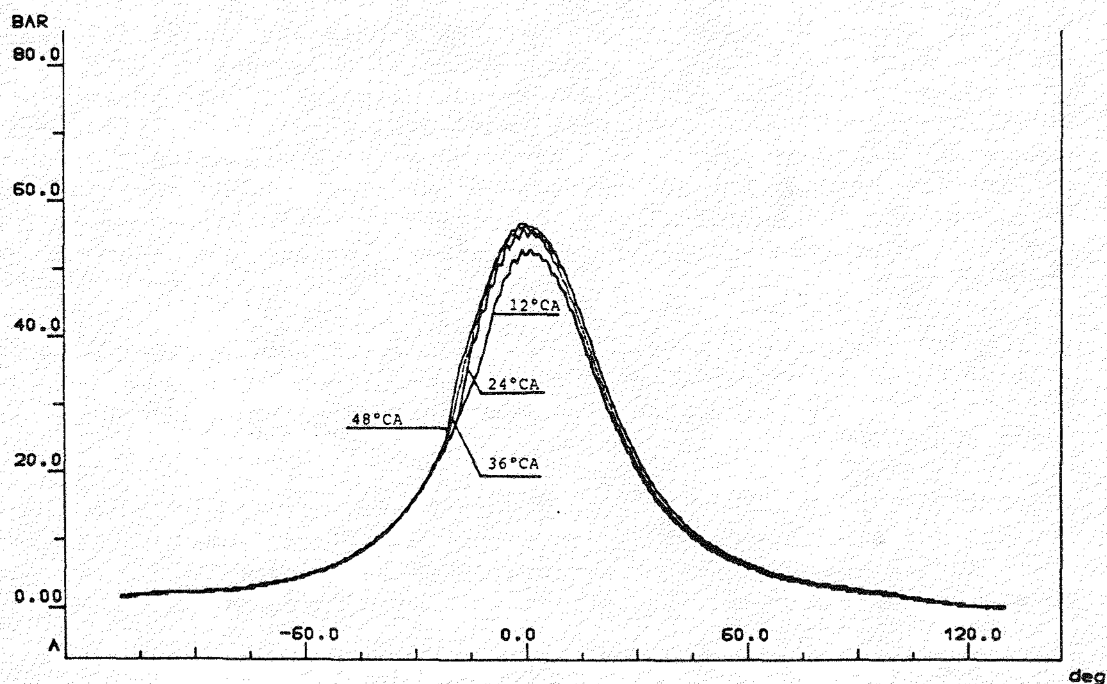

Figure 50. Cylinder pressures for different pilot timing.

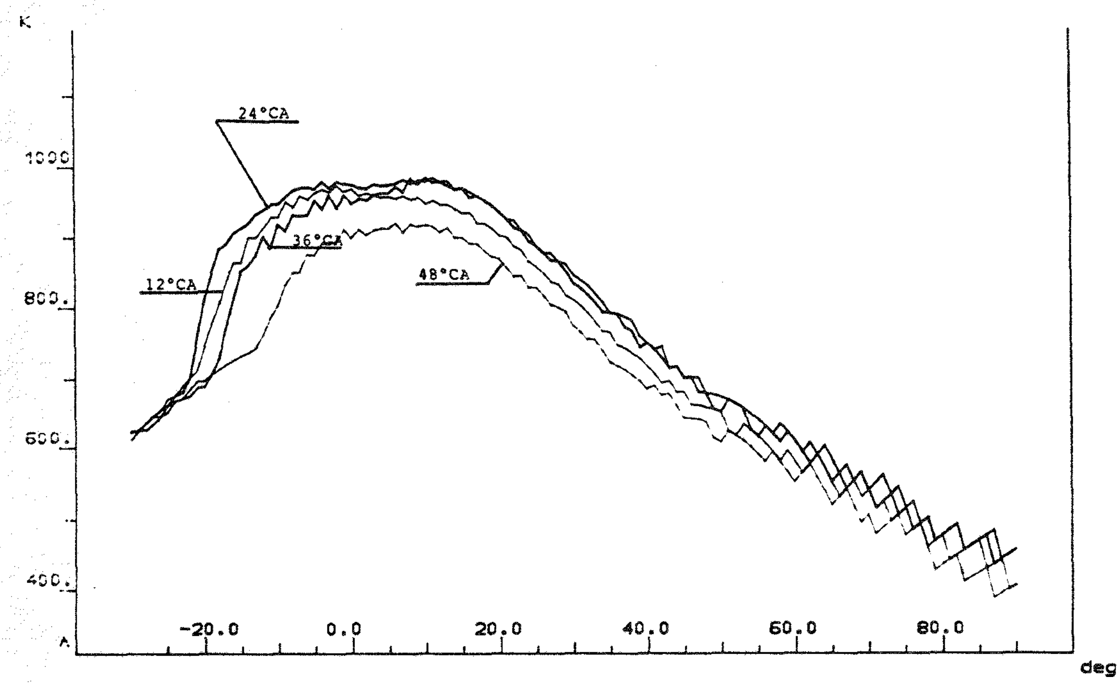

Figure 51. Cylinder temperatures for different pilot timing.

in the gas temperatures. At the beginning of the slurry combustion, the cylinder gas temperature is almost constant. The peak temperature region near TDC is almost constant for a small crank angle interval. Apparently, the rate of heat release of the slurry is just enough to compensate the piston power so that the temperature of the cylinder gases remain constant, for this particular data. It would be interesting to see if this trend can be generalized to a larger region of engine operation, because it is very important for the efficiency of the engine cycle. As is well known, for a given peak pressure limit, the addition of heat at constant temperature brings the engine operation to the ideal Carnot cycle efficiency, around that point. It may also be possible to obtain the same trend with pilot and main fuel injection with DF2. This feature needs further study at different engine speeds and loads.

Fig. 51 also shows that the constant crank angle period of cylinder gas temperature increases with an increase in the pilot advance. Further advance of the crank angle is expected to cause a decrease in the peak temperature near the end of vapor evaporation and slurry combustion. The pilot timing which may cause this may be an indication for the optimum timing for the pilot. In Fig. 51, the data with 36 degrees CA pilot timing advance indicates a reversal of trends after some optimum. The gas temperatures for 30 and 36 degrees crank angle advance does not differ much after about 8 degrees CA ATDC.

In Fig. 52 the exhaust temperature for the CWS burning cylinder is shown as a function of pilot timing relative to the timing of the reference run. The figure indicates a maximum around 6 to 18 degrees advance. It also indicates that the maximum is not very sensitive to the timing, being almost constant over a large time period. The range is about 6 and 18 degrees with reference to Test Run 26. A comparison of this figure with

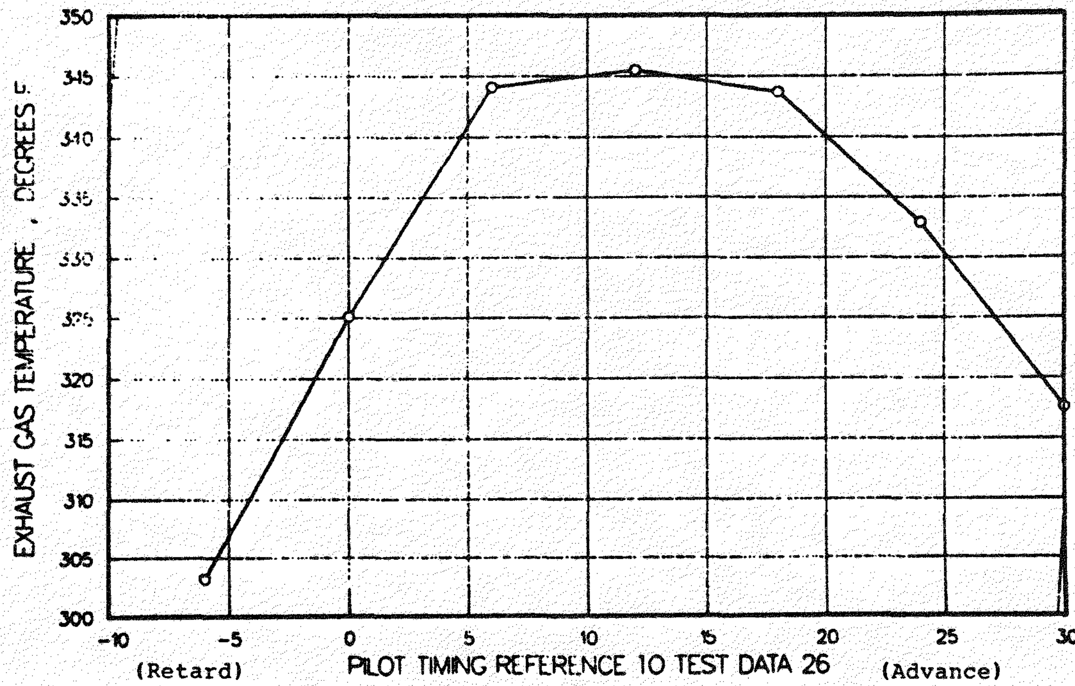

Figure 52. Engine exhaust temperature as a function of time.

Figs. 49 and 51 indicates that the pilot timing for the maximum exhaust temperature does not yield the maximum peak gas temperature or maximum cumulative heat release. Reasons for these trends needs further evaluation.

Low Pilot - Slurry Ratio Case:

In this case the relevant parameter values are as follows:

Engine Speed	:	837 RPM
Pilot Fuel Amount	:	80 mm3/injection
Main Fuel Amount	:	787.5 mm3/injection
Main Fuel Timing	:	1 Degree ATDC

The crank angle for the pilot injection start point is taken as the indicator for the pilot timing. It is changed from 46, 40, 34, 28, 24 and 18. The effects of the pilot timing on the instantaneous and cumulative heat release, and the in-cylinder gas temperature and pressure are shown on Figs. 53, 54, 55 and 56 respectively for the injection start timings at 40, 34, 28 and 24 crank angles BTDC. The following trends are observable from these figures, as the pilot timing is retarded toward TDC.

The Instantenous Heat Release Curves (Fig. 53)

a. The peak heat release due to the CWS combustion increases. It is highest at 24 degree case and lowest for 40 degree case.
b. The crank angle for the peak heat release point moves to later crank angles.
c. The start of slurry combustion, or the ignition point, is minimally affected by the pilot timing change.

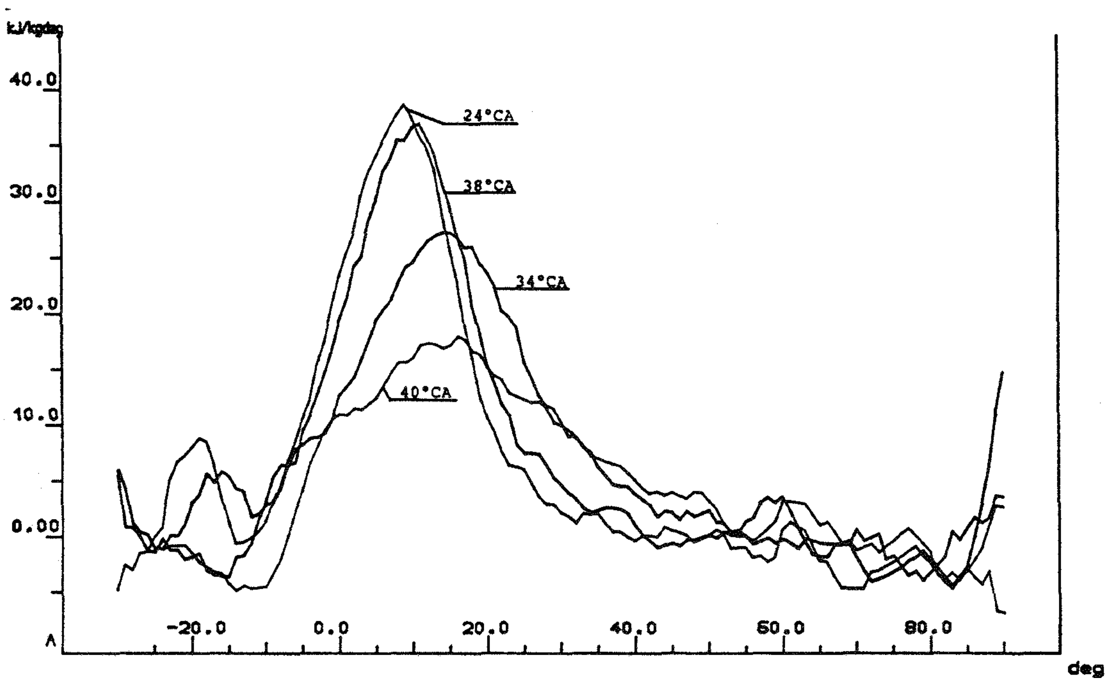

Figure 53. Effects of pilot injection timing on the instantaneous heat release. Low pilot/slurry ratio.

d. The magnitude of the negative heat release due to the slurry evaporation decreases.
e. The late heat release during the expansion period decreases.
f. From the observation that the heat release is more concentrated near TDC, one would infer that the engine efficiency would increase by the retarded pilot injection, within the range of the parametric values investigated.

The Cumulative Heat Release Curves (Fig. 54)

a. The cumulative heat release increases with retardation of the pilot timing up to 28 Degrees CA BTDC, then decreases,
b. The negative heat release portion during the CWS evaporation decreases and then it is completely eliminated. The heat release due to the DF2 combustion becomes larger than the heat of evaporation of slurry water and DF2 combustion becomes more evident.

The Cylinder Gas Temperature Curves (Fig. 55)

a. The gas temperatures (and the peak cylinder temperature) decreases,
b. The crank angle for the peak gas temperature moves away from TDC to later crank angles during expansion period,
c. At later crank angles, of the expansion period, the gas temperatures become larger, suggesting an increase in the combustion of slurry at these crank angles.

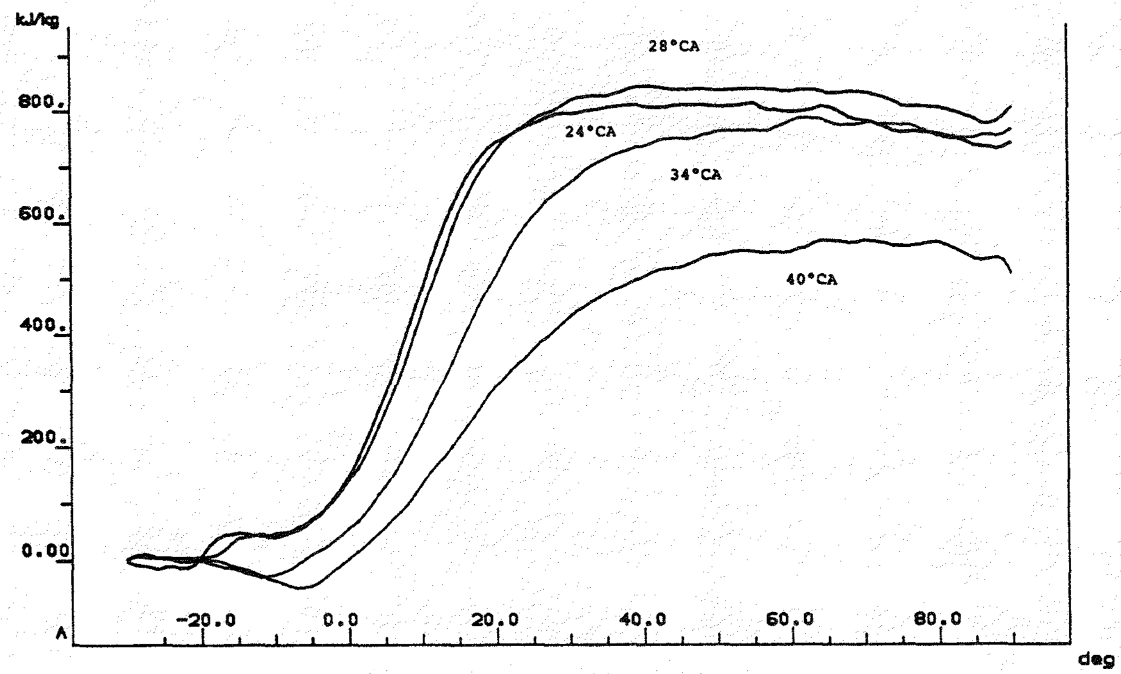

Figure 54. Effects of pilot injection timing on the cumulative heat release. Low pilot/slurry ratio.

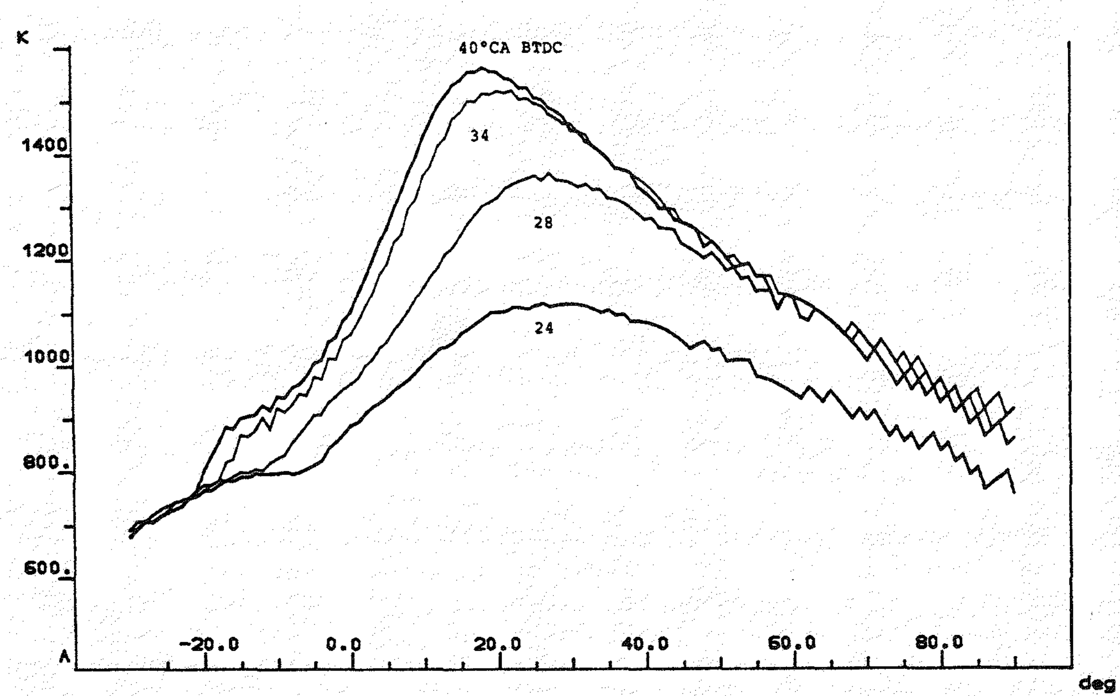

Figure 55. Effects of pilot injection timing on the cylinder gas temperatures. Low pilot/slurry ratio.

The Cylinder Pressure Curves (Fig. 56)

a. The peak pressure decreases and the crank angle for peak pressure advances toward TDC with later pilot injection.

E. EFFECTS OF PILOT FUEL AMOUNT ON SLURRY COMBUSTION

The effects of the pilot fuel amount on the slurry combustion is also investigated under several conditions of the engine operation. In general, as the pilot amount is decreased to a lower level, the exhaust smoke increases and becomes more visible. The quantitative trends are presented on Figs. 57, 58, 59 and 60 in terms of the cylinder gas pressure and temperature, as well as the total and instantaneous heat release curves. In these tests, the pilot fuel amount is decreased from 80 to 57, 36, 30, 39, 22.5 and 20 mm3/injection but the data is presented only for the 80, 36, 22.5 and 20 mm3/injection data points. The values of relevant parameters to this operation point are as follows:

Engine Speed	:	837 RPM
Pilot Timing (Start of Injection)	:	40 Degrees ATDC
Main Fuel Amount	:	902
Main Fuel Timing	:	1 Degrees ATDC

It is interesting to note that for the points plotted on these figures, the energy ratio of the fuels supplied by the pilot and the CWS are about 17.8, 8.9, 5.72 and 5.12 percent. In other words, the engine was operated with a pilot injection having a little over 5% of the total energy content of combustion, repeatedly, very near the full capacity of the unit injector. The operation of the engine at the lowest pilot amount point was such that there was

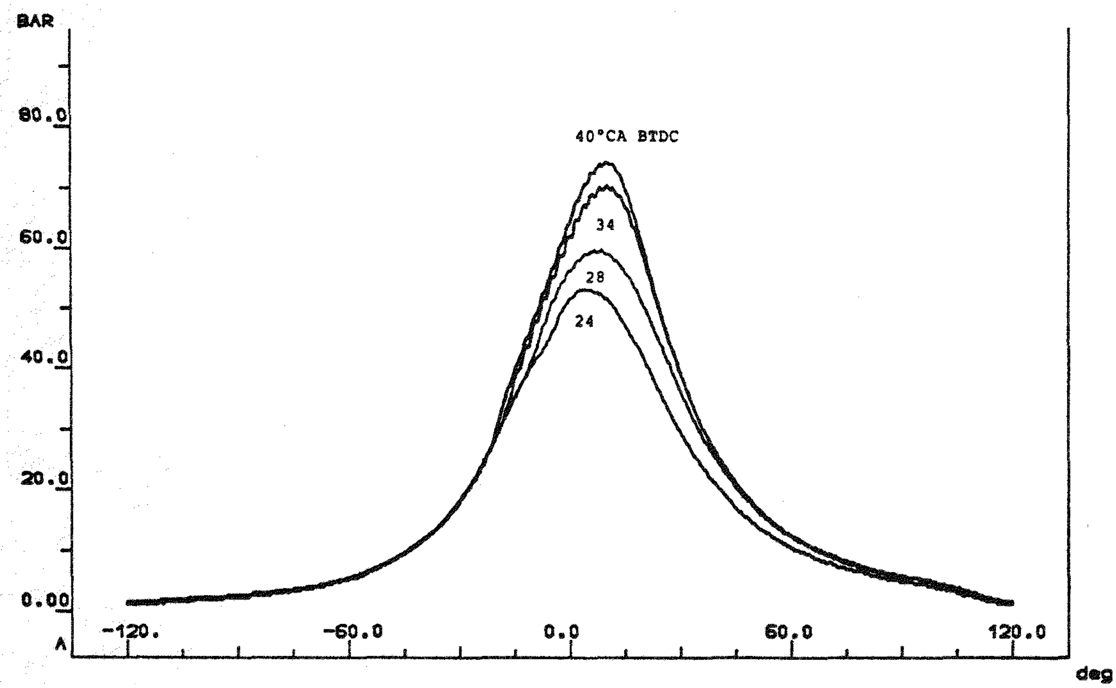

Figure 56. Effects of pilot injection timing on the cylinder gas pressure. Low pilot/slurry ratio.

a large cycle by cycle variation, as observed on the scope. The exhaust gas smoke level was also high. However, exhaust emissions were not measured.

A close examination of Figs. 57, 58, 59 and 60 would indicate the following trends with an increase in the pilot amount.

Cylinder Pressure Curves (Fig. 57)

a. Starting from the point of combustion of pilot fuel, the cylinder pressure increases,
b. Looking at the pressure traces, one would conclude that the start of combustion, or the ignition delay, does not seem to be changing. However, it is well known that the pressure trace is not a good indicator for the start of combustion. The cylinder temperature and/or heat release curves would be a better indicator.
c. Peak pressure location is retarded a little toward later crank angles.

Cylinder Temperature Curves (Fig. 58)

a. The cylinder temperature increases uniformly, starting from the initiation of combustion. Peak cylinder temperature also rises.
b. The crank angle for peak temperature moves toward TDC.
c. The rise in gas temperatures due to the CWS combustion moves toward earlier crank angles, indicating an earlier start of slurry combustion.

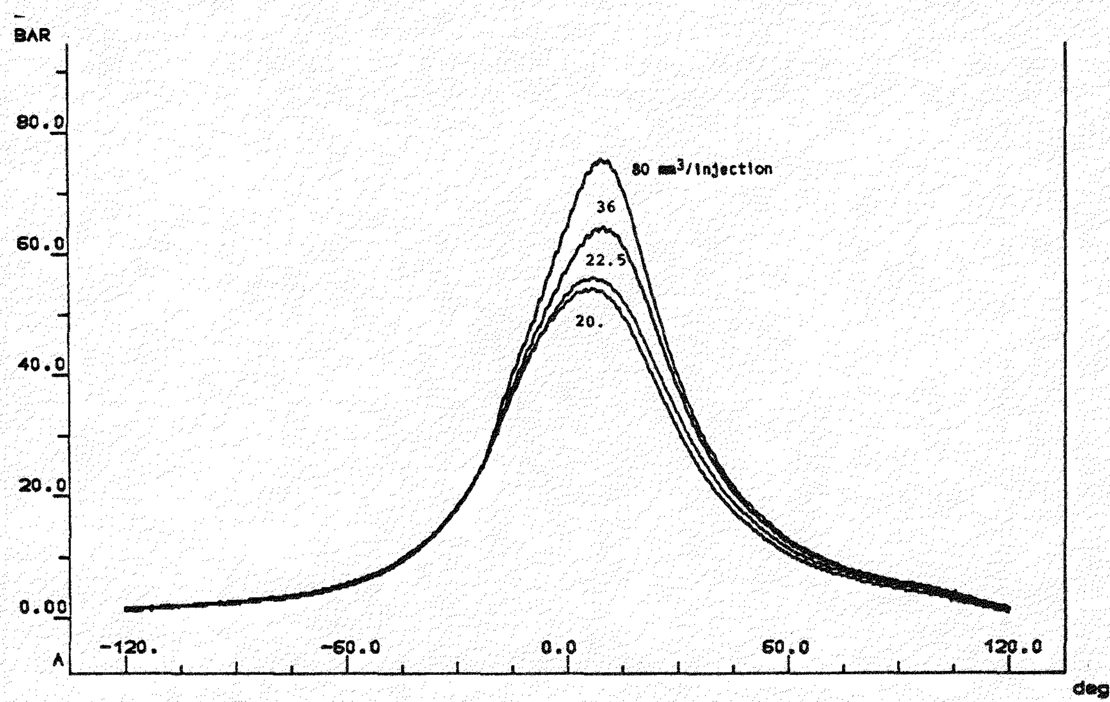

Figure 57. Effects of pilot fuel amount on cylinder pressure.

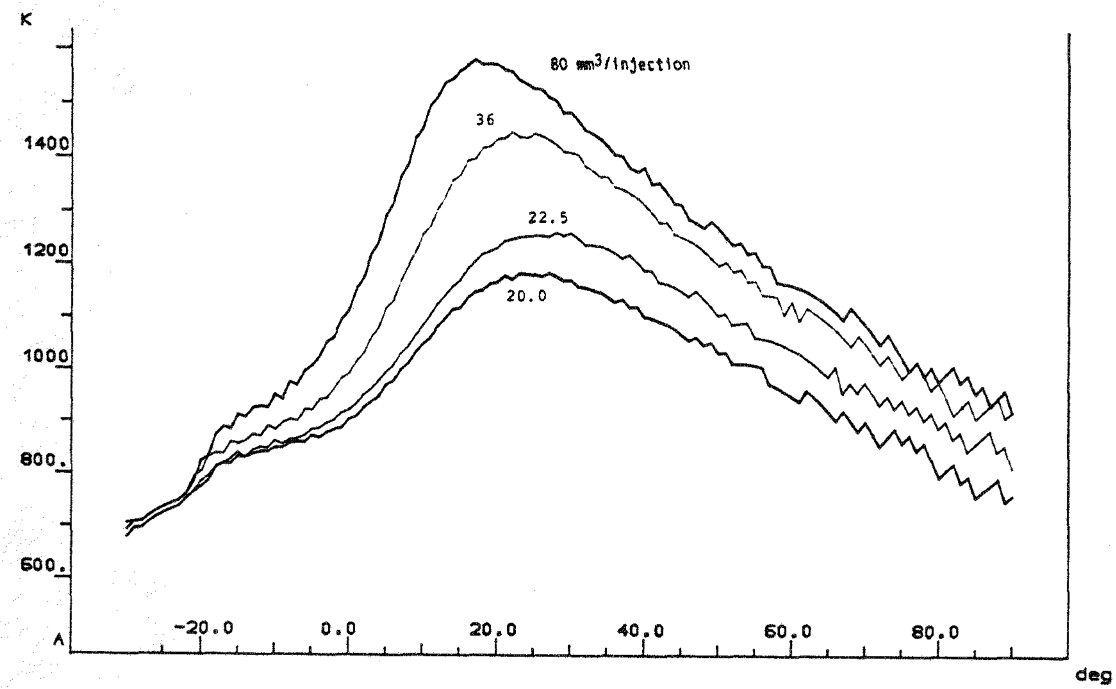

Figure 58. Effects of pilot fuel amount on cylinder gas temperature.

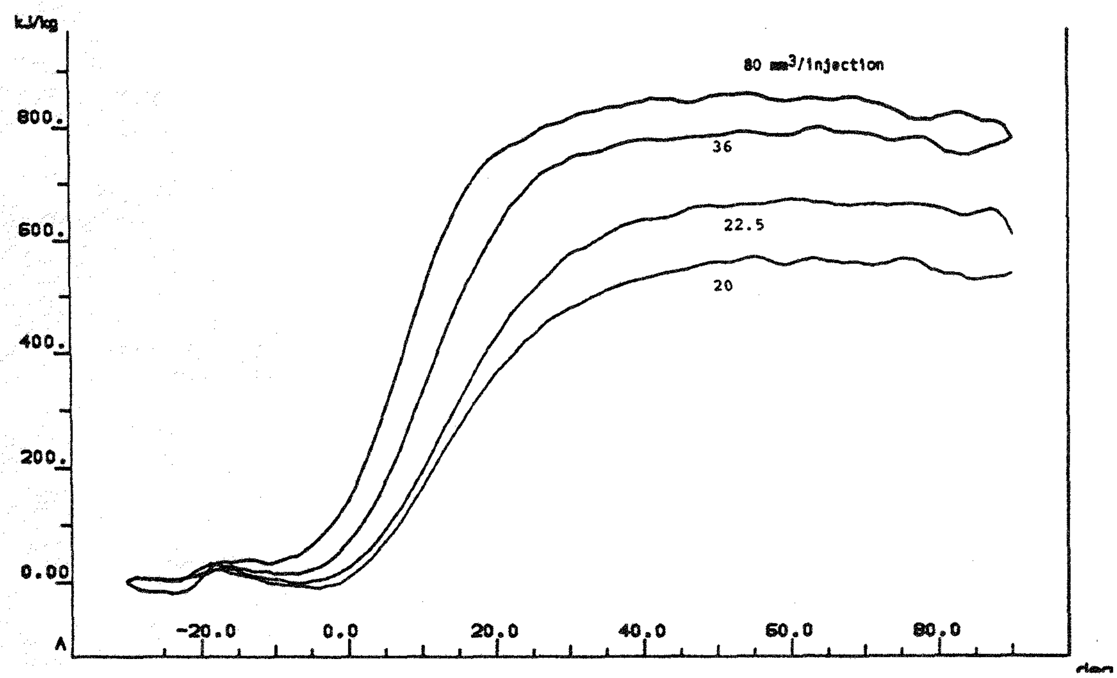

Figure 59. Effects of pilot fuel amount on cumulative heat release.

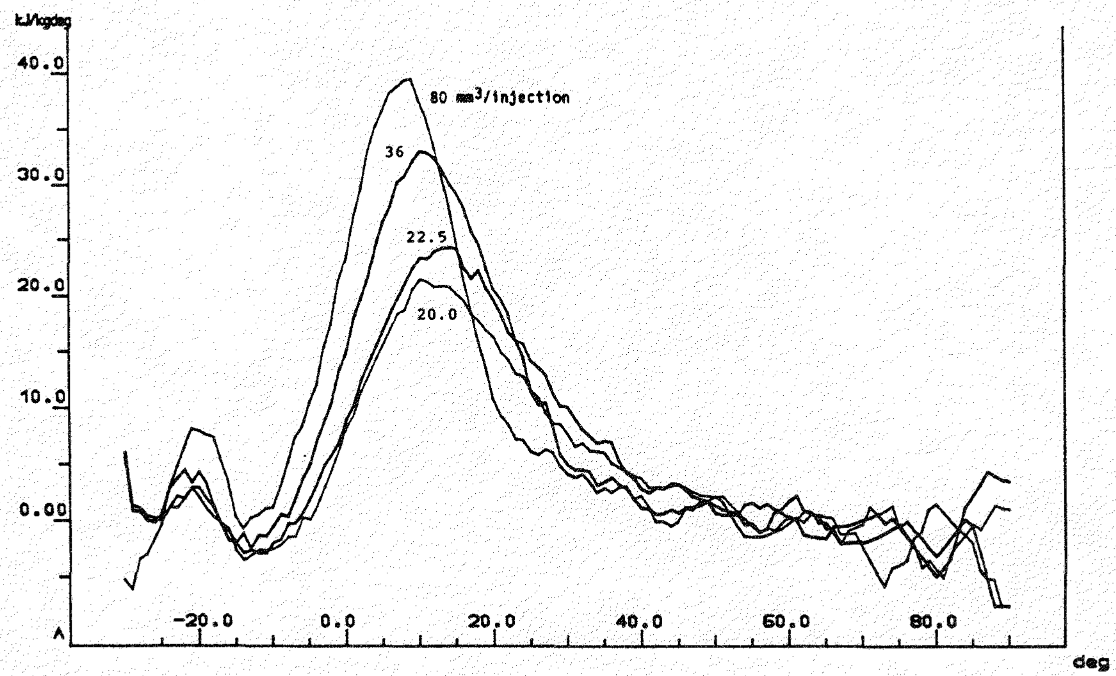

Figure 60. Effects of pilot fuel amount on instantaneous heat release.

Total Heat Release Curves (Fig. 59)

a. Total heat release increases
b. The crank angle for the completion of combustion (as indicated by the crank angle for the horizontal portion of the cumulation heat release curve) moves to earlier crank angles.
c. The pilot combustion portion of the curve becomes more visible, but it is rather small.
d. At low pilot amounts, there seems to be a negative total heat release portion of the curve. This means that the energy released by the pilot combustion was less than the latent heat of the slurry water evaporation at that crank angle. However the gas temperature still above the self-ignition limit of the coal and the ignition occurs. This is due to the high gas temperatures of the cylinder gases during compression.

Instantenous Heat Release Curves (Fig. 60)

a. The magnitude of the peak heat release due to CWS combustion increases.
b. The peak for the CWS heat release moves to earlier crank angles, toward TDC.
c. The heat release, occurred during the late expansion period, decreases,
d. Negative heat release due to slurry water evaporation decreases,
e. Pilot heat release increases.

F. EFFECTS OF SLURRY AMOUNT ON CWS COMBUSTION

The effects of increasing the slurry amount on the slurry combustion process is also investigated by increasing only the slurry amount, while keeping all other parameters held constant. Representative results are presented on figures 61, 62, 63 and 64 for the slurry amount of 377.5, 688.5 and 824.0 mm3/ injection. Other relevant parameters for these tests are:

Engine Speed	:	835 RPM
Pilot Amount	:	100 mm3/injection
CWS Injection Timing	:	11 Degrees ATDC
Pilot Injection Start	:	40 Degrees CA BTDC

On these figures the following trends are observable with an increase in the CWS amount injected.

Cylinder Pressure Curves (Fig. 61)

a. The peak pressure increased
b. The pressure rise rate is almost constant, implying that the CWS combustion rate is not affected by the increase in the slurry, controlled by other (probably evaporation and air mixing) phenomena. The combustion of the extra CWS is delayed to later times.
c. Location of the peak pressure moves slightly toward later crank angles.

Cylinder Temperature Curves (Fig. 62)

a. The cylinder gas temperature increases uniformly and the peak temperature also increases.
b. The peak gas temperatures also increases and moves toward later crank angles,
c. The peak gas temperature is later than the peak pressure.

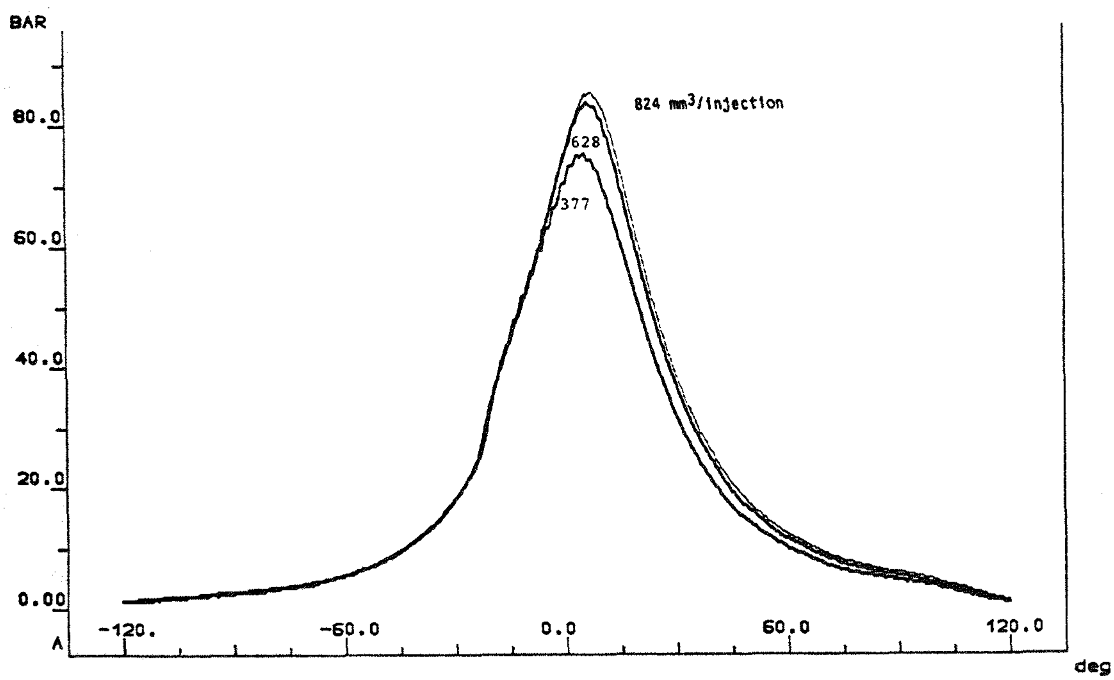

Figure 61. Effects of CWS fuel amount on cylinder pressure.

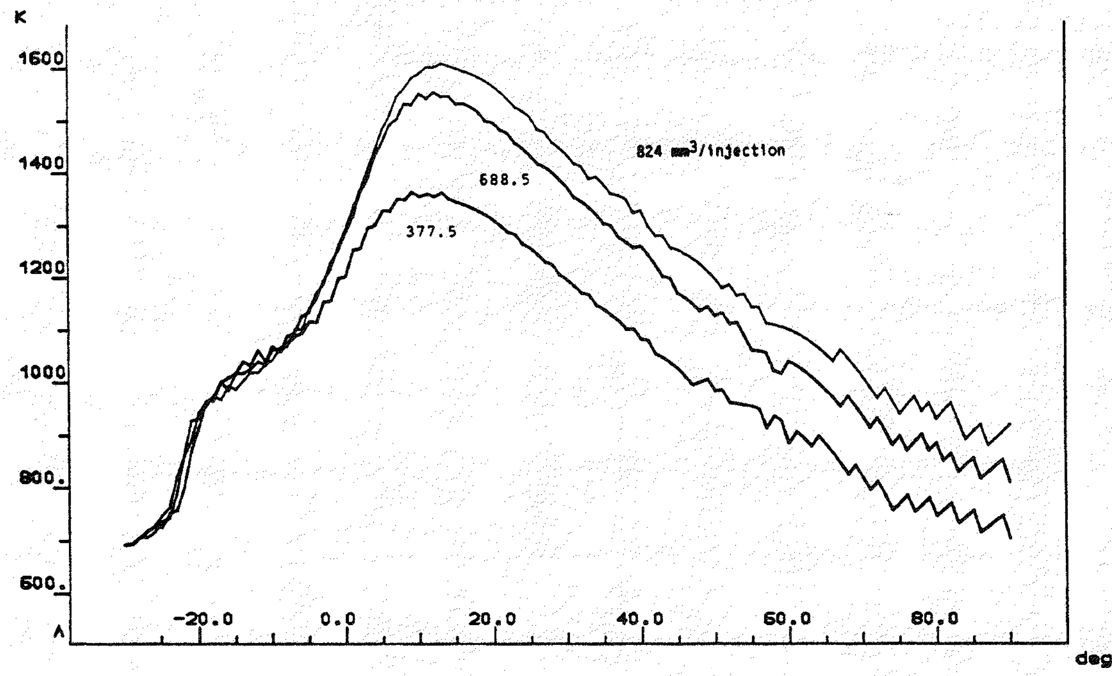

Figure 62. Effects of CWS fuel amount on cylinder gas temperature.

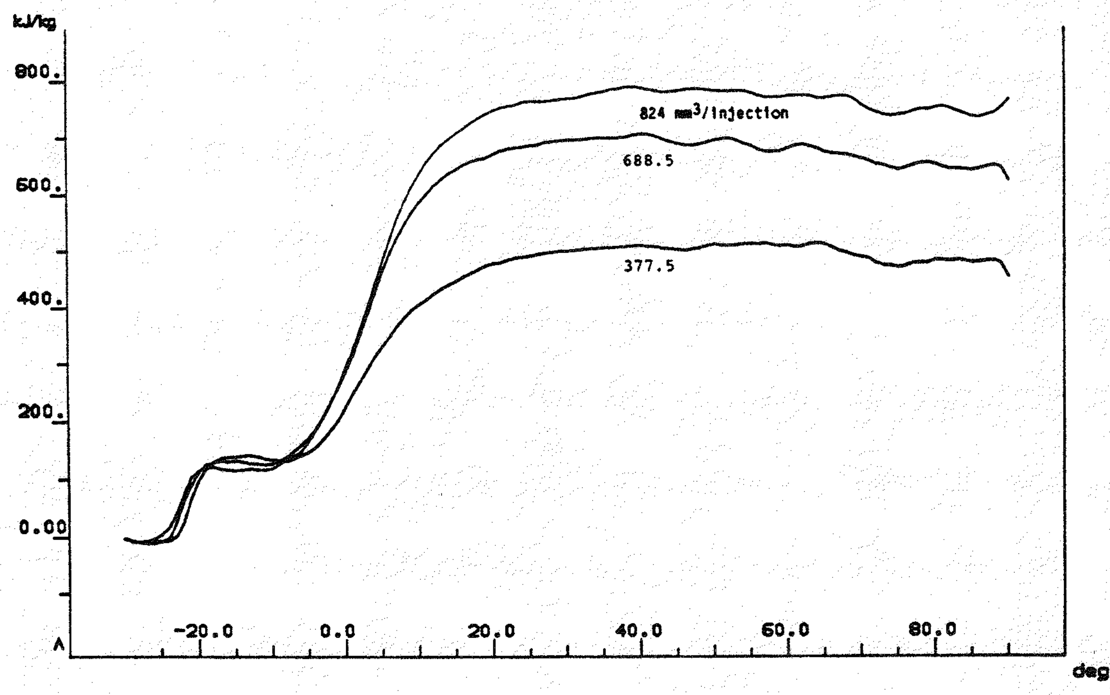

Figure 63. Effects of CWS fuel amount on cumulative heat release.

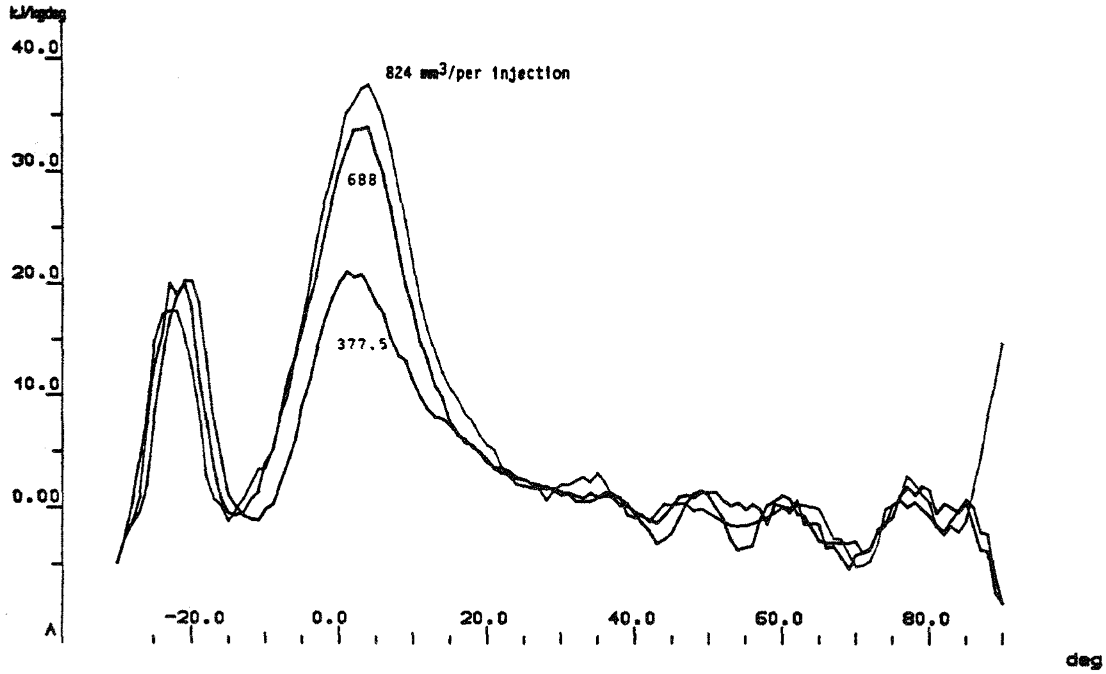

Figure 64. Effects of CWS fuel amount on instantaneous heat release.

Cumulative Heat Release Curves (Fig. 63)

a. The cumulative heat release increases. The increase does not seem to be proportional with the increase in the CWS amount injected, implying an increase in late combustion, leading to a reduction of combustion efficiency.
b. The curves do not have negative slopes at late expansion crank angles implying the absence of late slurry water evaporation.
c. The combustion start for the pilot advances a little with an increase in the slurry fuel amount. This is attributed to the increase in the cylinder gas temperatures at the beginning of the compression period and is a feature of the two-cycle engine operation. As there are about 10 to 20 percent exhaust gases in the cylinder, at the beginning of the compression, due to the scavenging characteristics of the engine, and as the cylinder gas temperatures increase with an increase in the CWS amount, the initial gas temperature in the cylinder would also increase with CWS amount. This would lead to an increase in the gas temperatures at the rank angle for pilot fuel injection and decrease its ignition delay. Increasing the slurry amount would therefore advance both the pilot ignition and slurry ignition for two-cycle engines.

Instantenous Heat Release Curves (Fig. 64)

a. Increase the peak heat release due to CWS combustion.
b. Advance the combustion of both pilot and CWS. The mechanism responsible for this phenomena is described above. This is a very clear indication of the basic feature that increasing the CWS amount would advance ignition of pilot and the slurry combustion.
c. The instantenous heat release in the late expansion period, about after 20 degrees CA ATDC is not affected by an increase in the slurry amount.

G. EFFECTS OF ENGINE RPM ON SLURRY COMBUSTION

The effects of the engine speed on the combustion of the CWS is investigated by keeping the pilot and CWS fuel the same (both timings and amounts) but changing the engine speed through the engine speed control. This would shift the load from the DF2 burning cylinder to the CWS cylinder (when the engine RPM is reduced) as the fueling rate of the CWS cylinder is kept constant. A large speed variation is covered including the maximum speed (900 RPM) and idle speed (365 RPM). Only representative data points are plotted on Figs. 65, 66, 67 and 68 for 900, 700 and 465 RPM operation points. In these tests, the values of relevant parameters were as follows:

CWS Fuel Amount	:	792.2 mm3/injection
Pilot Fuel Amount	:	80 mm3/injection
CWS Fuel Timing	:	1 Degrees CA ATDC
Pilot Injection Start	:	40 Degrees CA BTDC

With a decrease in the engine speed the following trends are observable from the figures:

Cylinder Pressure Curves (Fig. 65)

a. The cylinder pressure decreases uniformly, including the peak pressure,
b. The crank angle for the peak pressure moves slightly toward TDC.
c. The maximum pressure rise decreases.
d. Pressure rise starts at an earlier crank angle.

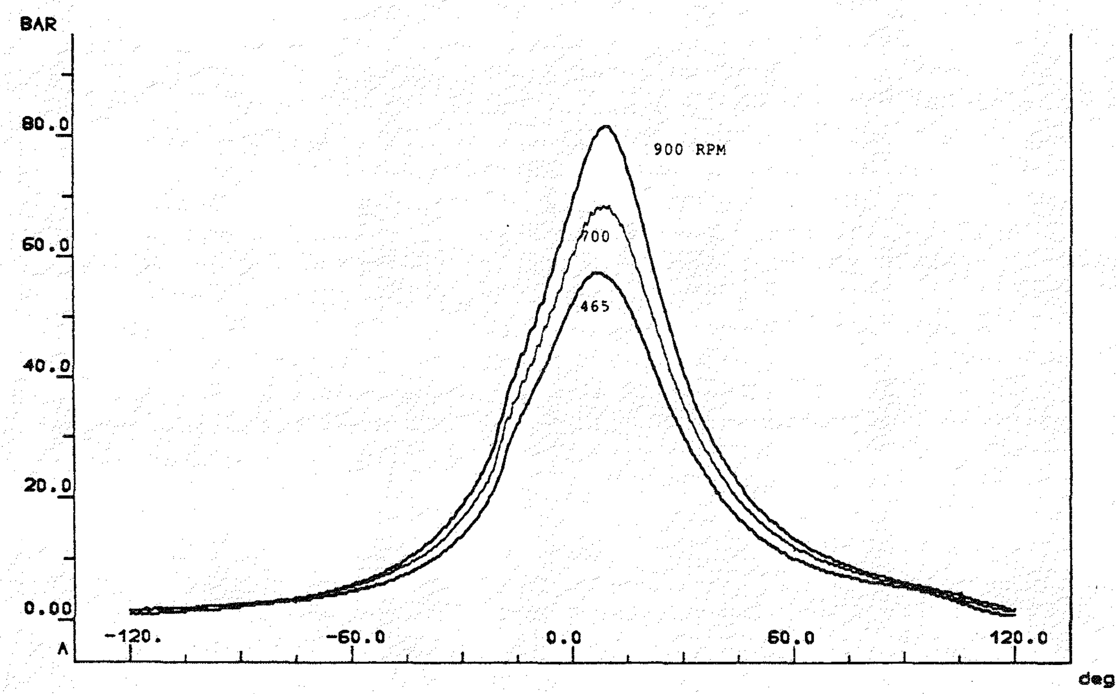

Figure 65. Effects of engine speed on cylinder pressure.

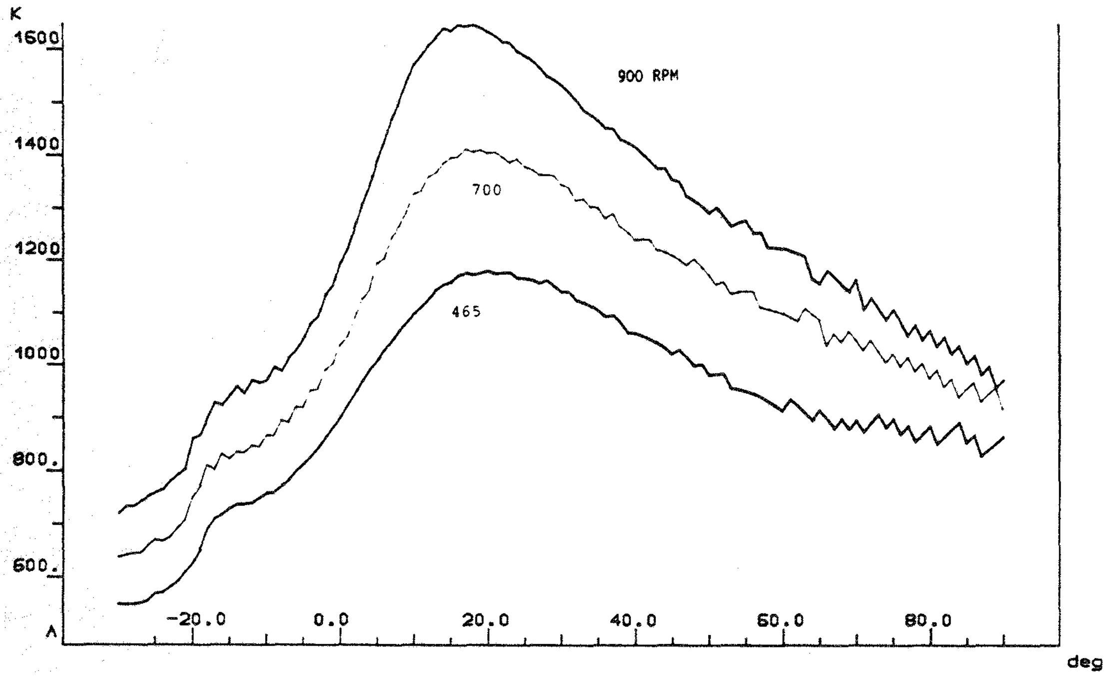

Figure 66. Effects of engine speed on cylinder gas temperature.

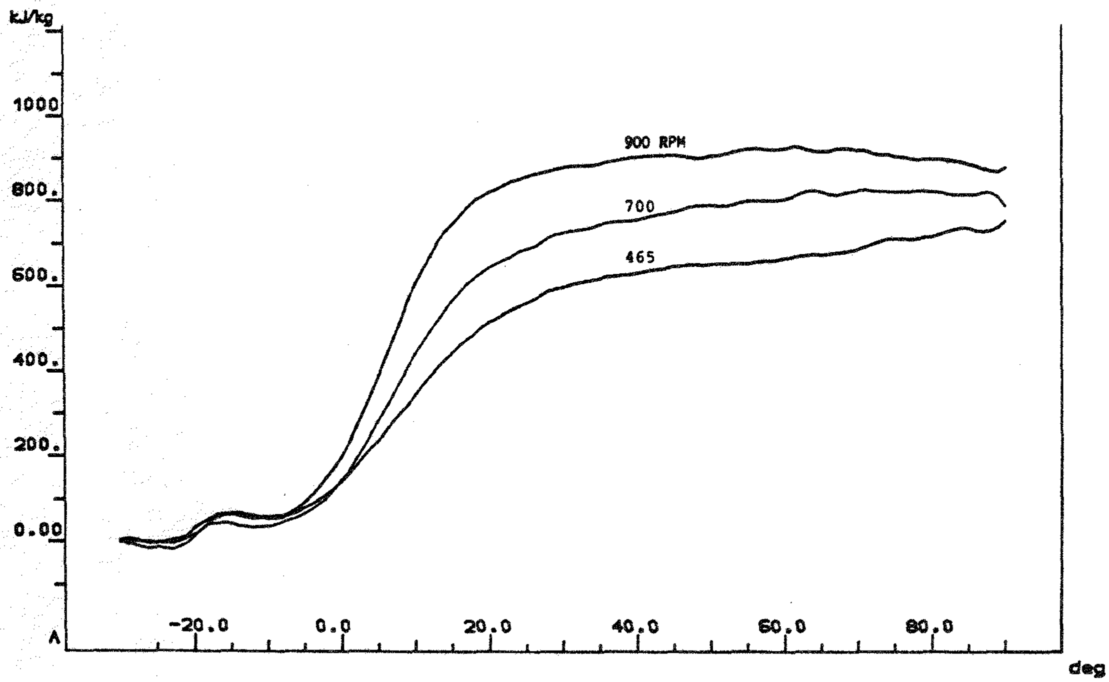

Figure 67. Effects of engine speed on cumulative heat release.

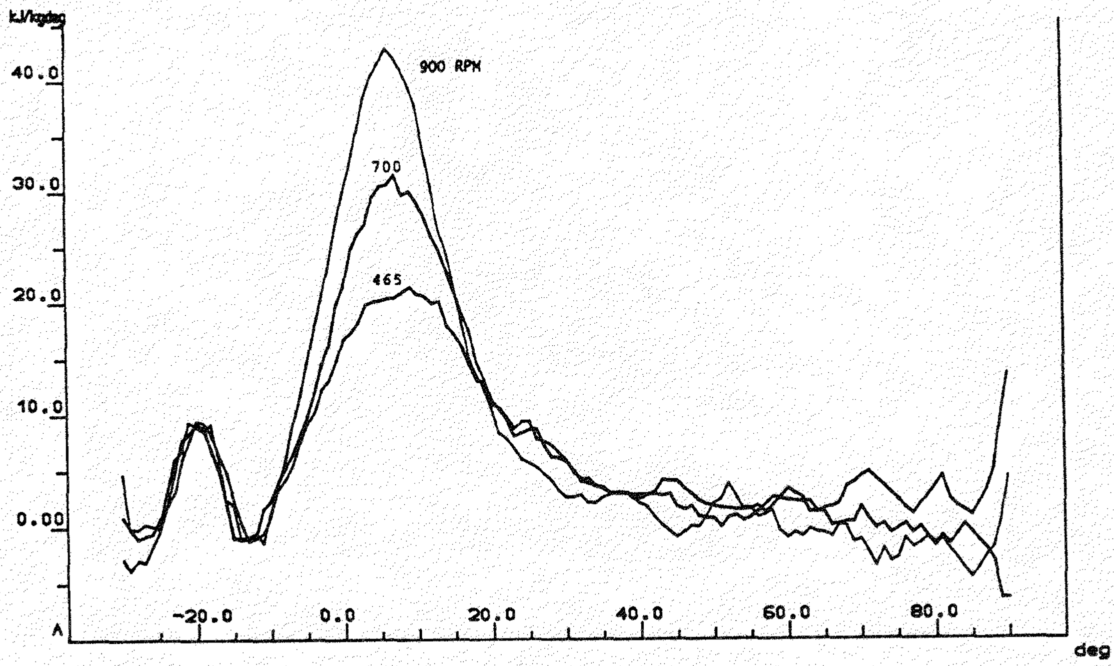

Figure 68. Effects of engine speed on instantaneous heat release.

Cylinder Gas Temperature Curves (Fig. 66)

a. The cylinder gas temperature decreases uniformly and the peak also decreases.
b. The crank angle for peak temperature moves toward TDC.
c. Pilot combustion starts a little later.

Cumulative Heat Release Curves (Fig. 67)

a. The cumulative heat release decreases.
b. The combustion of the CWS continues to later crank angles.
c. The slope of the combustion curve (combustion rate) decreases.

Instantaneous Heat Release Rate (Fig. 68)

a. The peak heat release rate decreases.
b. The late heat release increases. The implication of (a) and (b) is such that the combustion efficiency may be decreasing.
c. CWS combustion starts almost at the same crank angle. It is not affected by the engine speed.
d. Pilot combustion starts a little later.

H. EFFECTS OF INLET AIR HEATING

Inlet air heating is recognized as an alternative method of enhancing the ignition of slurry and a special steam heater is installed on the inlet fresh air line of the test setup. The effects of the inlet air heating on the ignitability and combustion of the CWS was studied by running tests at the same condition and by increasing the inlet air temperature in steps. The effects of the inlet air heating was expected to be different for low and high amount of pilot injection. For very large amount of pilot, the effects of the inlet air heating would be minimal, as there would be enough heat to evaporate the slurry. Hence the tests were run with medium and low amount of pilot fuel.

Inlet Air Heating Effects With Large Amount of Slurry

In this test the slurry amount was large and the pilot amount was medium. The values of the relevant properties were as follows:

Engine Speed	:	838 RPM
CWS Amount	:	824 mm3/injection
CWS timing	:	11 Degrees ATDC
Pilot Amount	:	58 mm3/injection
Pilot injection Start	:	40 Degrees CA BTDC
Percent of Pilot Energy	:	13.0%

Three cases covered in the figures 69, 70, 71 and 72 are for the inlet air temperatures 92.8, 142.1 and 160.6 F. A detailed study of these curves indicate the following trends with increasing inlet air temperature:

a. There is a little change in the cylinder pressure curves but the amount is small and the trend is not clear. There is a small increase in the peak pressure.
b. The cylinder gas temperatures increase considerably and uniformly at all angles.
c. The peak heat release rate of the CWS combustion increases. The CWS combustion advanced to earlier crank angles a little but the combustion of pilot fuel is not affected much.
d. The total heat release increases.

It should be noticed that the gas pressures did not change much while the calculated gas temperatures increased considerably. This is due to the decrease in the cylinder gas mass with increasing inlet air temperatures. The calculated cylinder gas masses for these cases (including the fuels) are 13.7, 12.58 and 12.17 grams respectively. Even if the fuel combustion efficiency would remain the same, this decrease in the cylinder mass would result in the higher temperatures without increasing the pressure (as the fuel amounts kept constant). Additionally the increase in the cylinder gas temperature is also increasing the combustion efficiency and thus lead to higher cumulative heat release.

4.6 PROBLEMS ENCOUNTERED

In this section, the problems encountered with the engine test hardware, particularly related to the use of CWS, is covered in more detail. Essentially, there were three major problem areas; namely, (a) the slurry storage system, (b) the CWS injection system, and (c) the slurry changeover system. The problems of the slurry storage system were described in detail in Section 2.3 and will not be repeated here.

As mentioned before, there were two slurry injection systems used in these tests, namely the main system and the backup system. The details of the main system are available in Section 3.5 E. In review, it has a single injection pump and an injector nozzle. Essentially the problem with this system was the

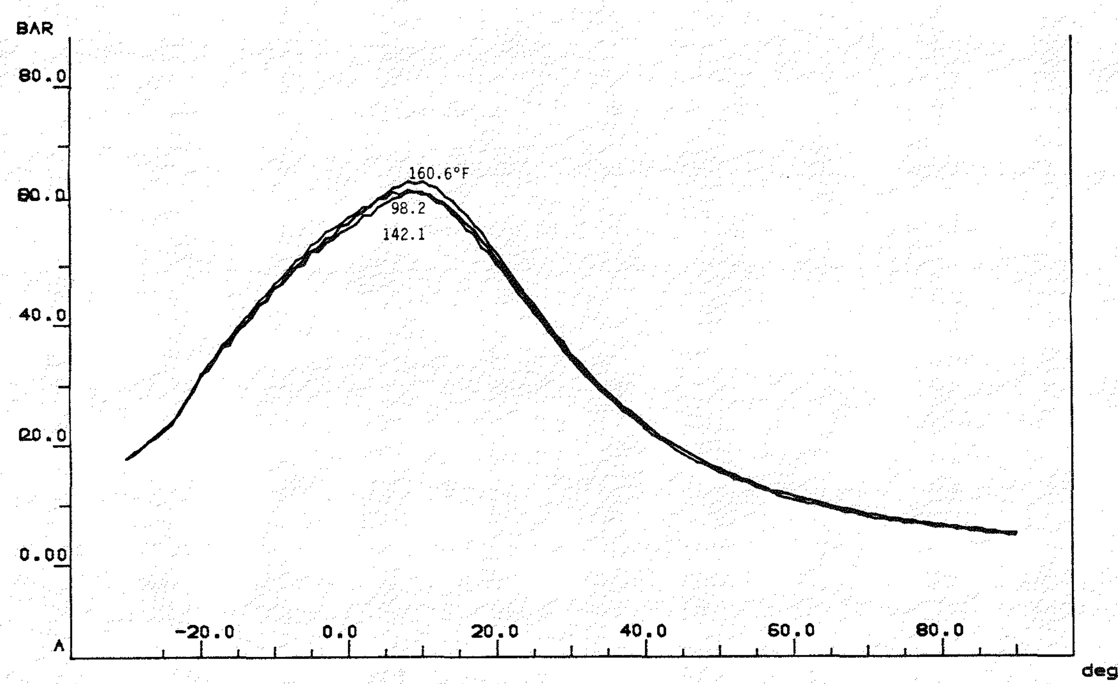

Figure 69. Effect of inlet air heating on cylinder pressure.

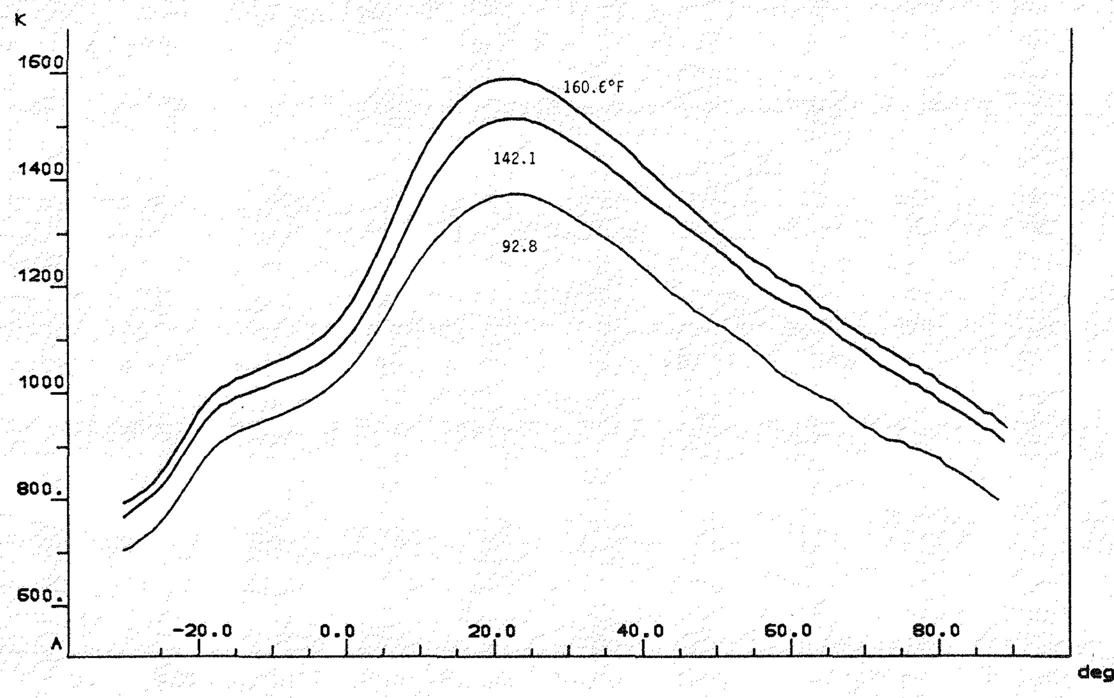

Figure 70. Effect of inlet air heating on cylinder gas temperature.

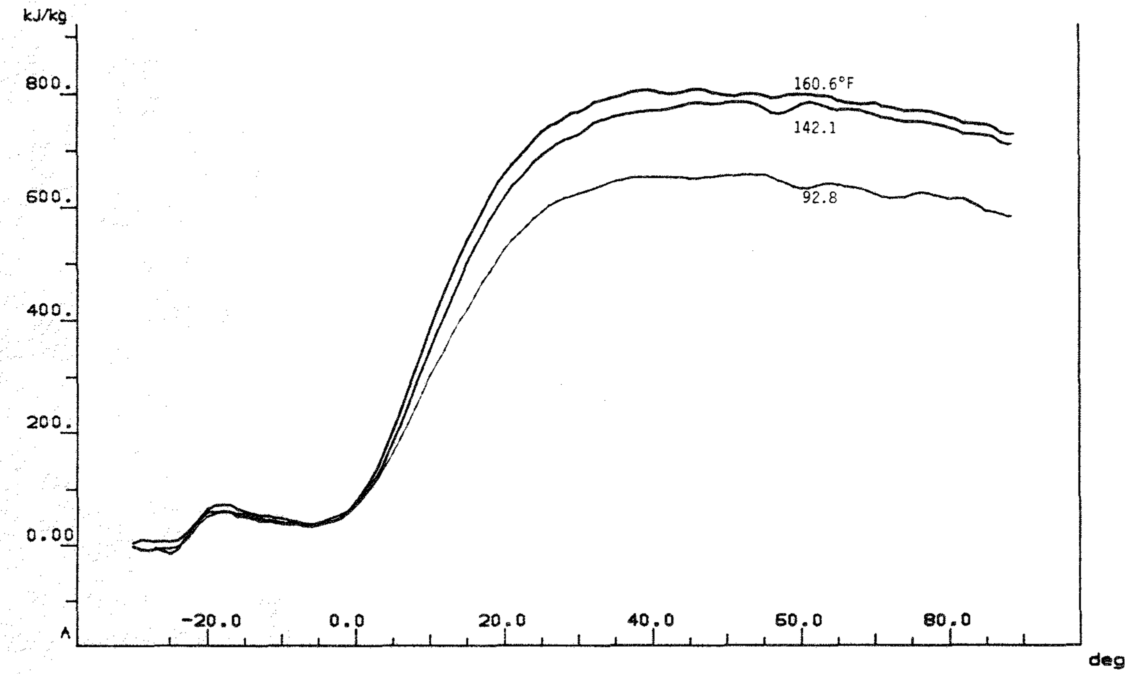

Figure 71. Effect of inlet air heating on cumulative heat release.

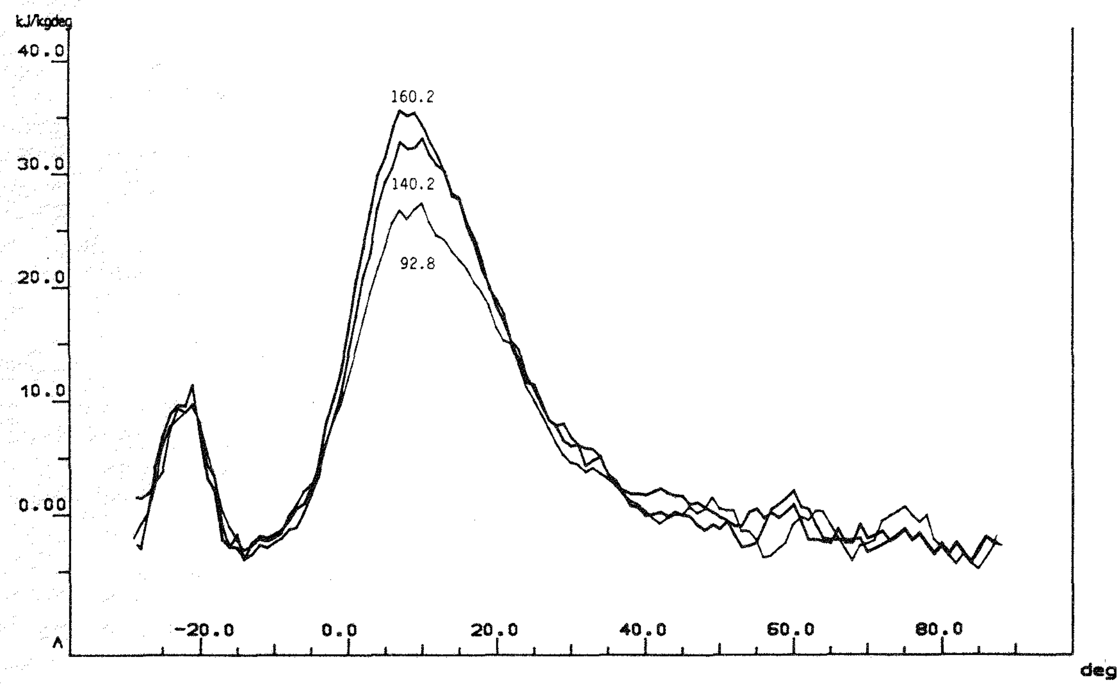

Figure 72. Effect of inlet air heating on instantaneous heat release.

suitability of injector clearances for the three different fluids injected at different times. The system is operating with DF2 and water satisfactorily but when the water is switched to CWS, the exhaust temperature begins to rise; but before it was stabilized, the injector plunger was stuck. The same phenomena was repeated at the same point of the operation several times. Two different methods were tried to alleviate the problem. Either the plunger bushing clearance was changed or a different type lubricant was tried between the plunger and the bushing. A lubricant was sought in which none of the fluids (namely the DF2, water and coal particles) would be soluble in and wetted by this lubricant. All reasonable alternatives tried first in laboratory tests then on the engine did not eliminate the plunger sticking problem. Even a dry lubricant with even backing procedure was attempted. Finally, due to the delays in the project, the main injection system was replaced with the backup system. The backup system worked in the first trial for about 23 min. The tests were terminated not because of problems but for other reasons. Evaluation of the test results with the backup system indicated a trend worth mentioning. When the water is injected, the fluid temperature is about 65°F. When the system switched to slurry, the fluid line temperature begins to rise up to about 90 to 100°F range, and problems with the injection system begins in the form of oscillations of the exhaust gas temperature or as an increase in the cycle-to-cycle variations for the cylinder pressure trace on the scope. The increase in the slurry temperature is attributed to the heating of the slurry due to the circulation within the loop. When the slurry temperature is reduced, either by increasing the cooling capacity or by switching to water and then switching back, then those indications would disappear. It seems that keeping the slurry temperature low, below 80°F, would be beneficial in eliminating injector plugging problems.

The first unit injector used was finally stuck at a test point with a slurry temperature of about 98°F at the end of about 3 hours of actual slurry injection. Up to that point, there was no deterioration of its performance, as observed from engine test data recorded. After its failure, it was taken apart, cleaned, visually inspected and sent for flow tests. Visual inspection did not show any scratches or observable wear marks on the plunger. Flow test results were not available at this writing. They will be compared against initial values taken before starting using that injector.

There were some switchover system problems. The pressure on the circulation loop was controlled by pressure relief valves to reduce and eliminate possible cavitation. Changeover system transient time, particularly when switching from water to CWS, was too long. This is measured by taking engine data at smaller time intervals and reduced by increasing the outlets flushing the water out of the system.

The operation of a single injector pump system for a V-2 engine also created some vibration problems. These were eliminated by using conventional techniques.

5.0 ENGINE COMPONENT DURABILITY

5.1 BACKGROUND STATEMENT

A major obstacle to development and ultimate commercialization of a coal-fueled diesel engine is the wear and corrosion of the combustion chamber parts exposed to the fuel and the products of its combustion during operation. Problems encountered or to be expected include wear of the piston rings and cylinder liner, exhaust valve seat and valve guide wear, and corrosive attack of the hot valve head. EMD has experience in similar environments gained during development programs with high-sulfur and high-ash fuels. The coal-fueled engine is likely to combine the worst of the high-sulfur and high-ash environments. Early attention to the resulting problems will shorten the coal-fueled engine development program.

The main goals of this project are determination of the areas of primary wear and corrosion attack during operation of a two-stroke cycle diesel engine fueled by coal and to select, by experience and by rig testing, piston ring face and cylinder bore coating materials to mitigate wear in those areas. Information gained during this program will be used as basis for future coal-fueled diesel engine projects.

5.2 OBJECTIVES

The objectives of this program were:

- o To choose best current technology ring face and cylinder liner bore coatings for wear reduction in coal-fueled diesel engines.
- o To procure 10 cylinder kits of rings, liners, and pistons coated as appropriate with the above listed materials.
- o To determine the effects of coal fuels on engine component durability including the effects on the coatings chosen and procured above.
- o To qualify additional ring and liner material combinations for future coal-fueled diesel applications using a friction and wear test rig.

5.3 PROJECT DESCRIPTION

- o Best current technology choice of piston ring and cylinder liner bore materials based on accumulated experience of the investigators.
- o Tribology study using a special test rig for wear-resistant materials.
- o EMD 8-645 1000-hour DF2 fueled engine operation wear study.
- o EMD 2-645 engine operation with CWS (10-hour) engine component wear test program.

ENGINE COMPONENT WEAR TEST PROGRAM STRATEGY

Baseline Wear Information

Piston ring face and cylinder liner bore wear-resistant coatings have been chosen based on the accumulated experience of the investigators in like tests. (See Results/Accomplishments for coating details.) Four coated power assemblies and four standard production power assemblies will be run in an 8-cylinder model 645 turbocharged diesel engine with a goal of 1000 hours on DF2 fuel. Detailed component measurements were taken prior to the commencement of the run; the same measurements on worn components will be taken after the run conclusion for comparison. Additionally, one assembly of each type will be removed and replaced with new at 300 and 600 hours into the test; the wear comparison will thus consist of assemblies with 300, 400, 600, 700, and two with 1000 hours, all of which will have unworn dimensions available for wear amount determination.

Wear Information Based on CWS Operation

Two coated assemblies were scheduled to be run for a total of 10 hours in the 2-cylinder 645 engine on CWS. One cylinder was to be fueled with DF2 only; the other cylinder with the highest ratio of CWS to DF2 we can attain for continuously good operation with coal fuel. Comparative wear information would be reported on the two cylinders. While the wear reduction effort in the current program is directed toward ring faces and cylinder bores, wear comparisons of all combustion chamber parts affected by the coal fuel will be made for direction for future programs. Unfortunately, time and funding constraints prevented the completion of the 1000 hour tests and any of the CWS testing.

5.4 RESULTS/ACCOMPLISHMENTS

Initial Choices of Ring and Cylinder Bore Materials

Three recommendations of piston ring face and cylinder liner bore material combinations were made by Adiabatics, Inc. based on their accumulated experience in like investigations. The recommended combinations were:

1. Liner: 0.0015 inch NiCrAlY bondcoat
   0.0070 inch plasma-sprayed yttria-stabilized zirconia
   0.0085 inch total coating thickness

   Coating to be post-densified with chrome oxide.

   Rings: Option 1: Plasma-sprayed Cr_2O_3
   Option 2: Plasma-sprayed alumina-titania
   Option 3: Plasma-sprayed chrome carbide

2. Liner: 0.005 inch (maximum) Kaman Sciences Corporation SCA-1000 wear coating, post-densified with chrome oxide

   Rings: Option 1: Plasma-sprayed alumina-titania
   Option 2: Plasma-sprayed CR_2O_3
   Option 3: Plasma-sprayed chrome carbide

3. Liner: 0.0015 inch NiCrAlY bondcoat
   0.0065 inch plasma-sprayed chrome oxide
   0.0080 inch total coating thickness

   Rings: Option 1: Plasma-sprayed chrome carbide
   Option 2: Plasma-sprayed alumina-titania

Adiabatics, Inc. recommended the second coating combination most highly; however, some potential incompatibility of the cylinder bore coating process with the production cylinder liner configuration was identified. The EMD cylinder liner has steel water jackets attached to the gray cast iron cylinder liner body by silver brazing. The repeated temperature cycling to 1050°F required in the plasma spraying and chrome oxide densification process of the coatings in the first and second recommendations would possibly compromise the integrity of the braze joints. To test this, a cylinder liner was cycled six times between room temperature and 1050°F, the temperature of coating and densification. When the joints held, the SCA-1000 coating was chosen as the liner material.

To select among the optional ring face coatings, their properties were examined. Critical properties of the optional coatings as extracted from Union Carbide literature are shown in the following table.

Table 12.
Piston ring-face-coating properties.

Coating	UCAR No.	Bond strength --psi	Coating density --g/cc	Modulus of rupture --psi	Thermal expansion coefficient --in./in.°F
Cr_2O_3	LC-4	5,000	5.0	39,000	3.7 x 10-6
Al_2O_3 + TiO_2	LA-7	9,000	3.6	21,000	NA
Chromium carbide	LC-1C	10,000+	NA	77,000	5.6 x 10-6

Chromium carbide (Union Carbide LC-1C) was chosen based on its higher bond strength and modulus of rupture, a coefficient of thermal expansion close to that of the ductile iron ring base material, and an earlier experience of coating delamination with alumina-titania ring face coatings. The chemical make-up of the Union Carbide LC-1C coating is 80% chrome carbide (92 Cr, 8 C) and 20% nickel chrome (80 Ni, 20 Cr).

EMD procured ten cylinder sets of liners and rings coated with the chosen materials. The rings were obtained from Engine Products Division of Dana Corporation who used Union Carbide coating services to apply the face coating. The cylinder liners are being fabricated through a cooperative effort of EMD and Kaman Sciences Corporation. Twenty liners were honed oversize to accommodate the coating thickness and sent to Kaman for plasma spraying with the bore material. The liners were then returned to EMD for honing to size and sent back to Kaman for densification with chrome oxide. A final light honing at EMD completed the liner. Considerable logistical difficulty was encountered in this process. The oven in which the liners are heated during the densification process will hold only two parts of the size of EMD cylinder liners. The parts, once coated with the ceramic, must be passed through the honing and densification process expeditiously or they are ruined. The oven capacity limitation thus means that liners can be coated only in small lots; the resulting stop-start coating process is devoid of any economy of scale. This has strung out the coating process of the liners. Added to this has been honing through of the thin ceramic coating due to distortion of the liners that has occurred during spraying and densification and the lack of precision honing machinery in the EMD engineering shop. Additionally, the coating has come off some of the liners during honing due to improper cleaning of the liners prior to plasma spraying.

To date, the 8-cylinder 645 wear test engine has accumulated 262 total hours of running. Two visual inspections were conducted and No. 1 power assembly has been removed for dimensional evaluation. This power assembly was pulled 28 hours earlier than specified due to coated ring chipping problems with the compression rings and to coincide with the replacement of a failed turbocharger.

Previous lab testing had indicated that the braze joint integrity of the cylinder liners would not be adversely affected by the 1000°F heating process for ceramic coating and densification. All ceramic coated liners exhibited extensive braze joint leaks requiring impregnation with an anerobic adhesive at Casting Impregnators located in Chicago. Three applications of the adhesive were necessary to stop all the leaks.

Cylinder bore profiles for the ceramic liners were a major departure from EMD standard specifications. Due to time constraints for this test, past problems with honing through the ceramic coating on previous liners, and the belief that these deviations would have minimum impact on the test, it was decided to use these liners. The upper bore of the ceramic liners has 0.0025 in. taper per side with the bore area near the gasket face having the smallest diameter. The port relief was 0.0025 in. deep compared to a 0.004 to 0.009 in. EMD requirement. The liner profiles were taken on all four ceramic liners and only two standard liners and are shown in Figures 73 through 78. A summary of the bore diameter measurements is itemized in Table 13.

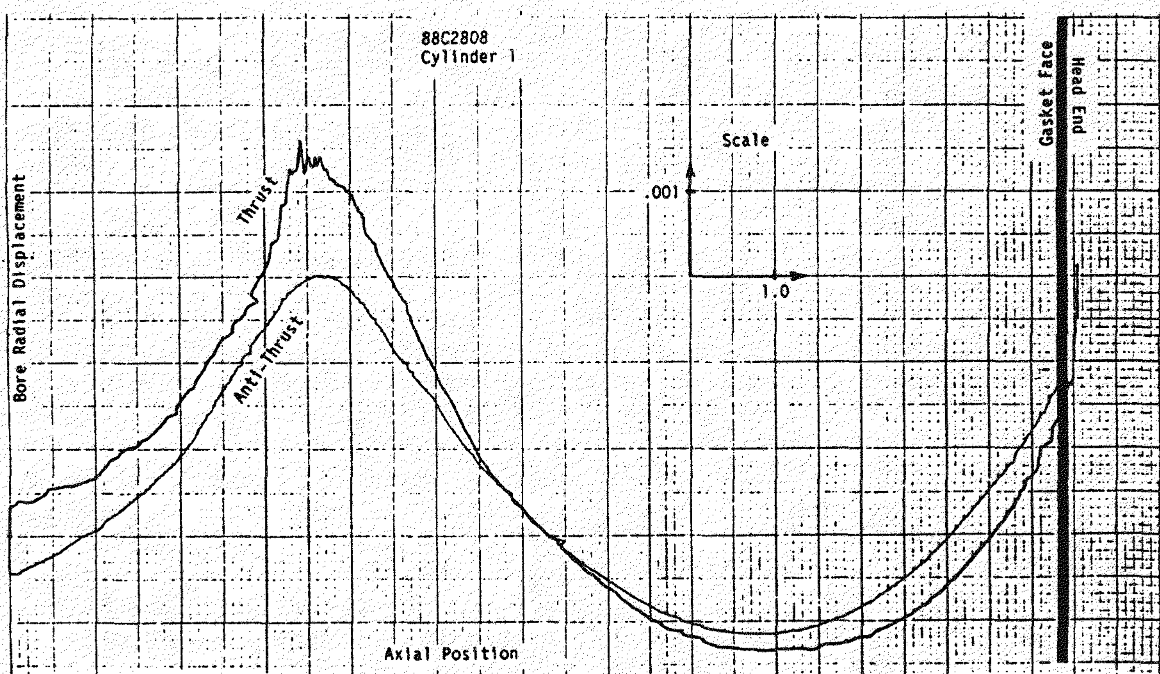

Figure 73. Cylinder bore profile - cylinder 1.

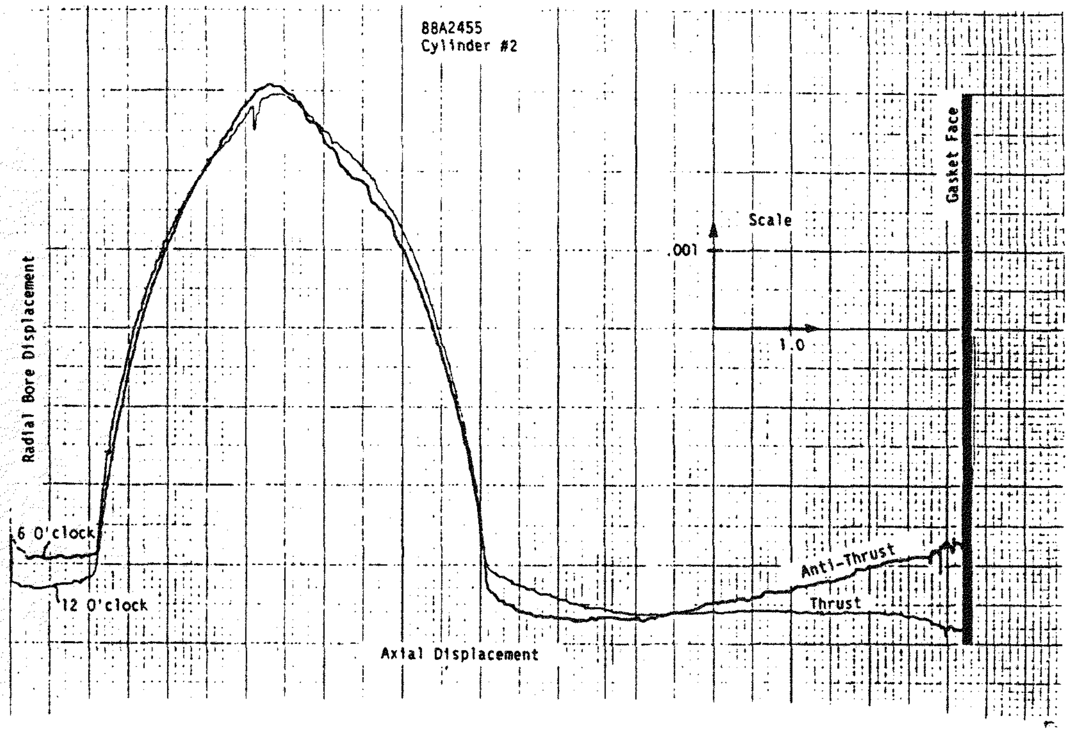

Figure 74. Cylinder bore profile - cylinder 2.

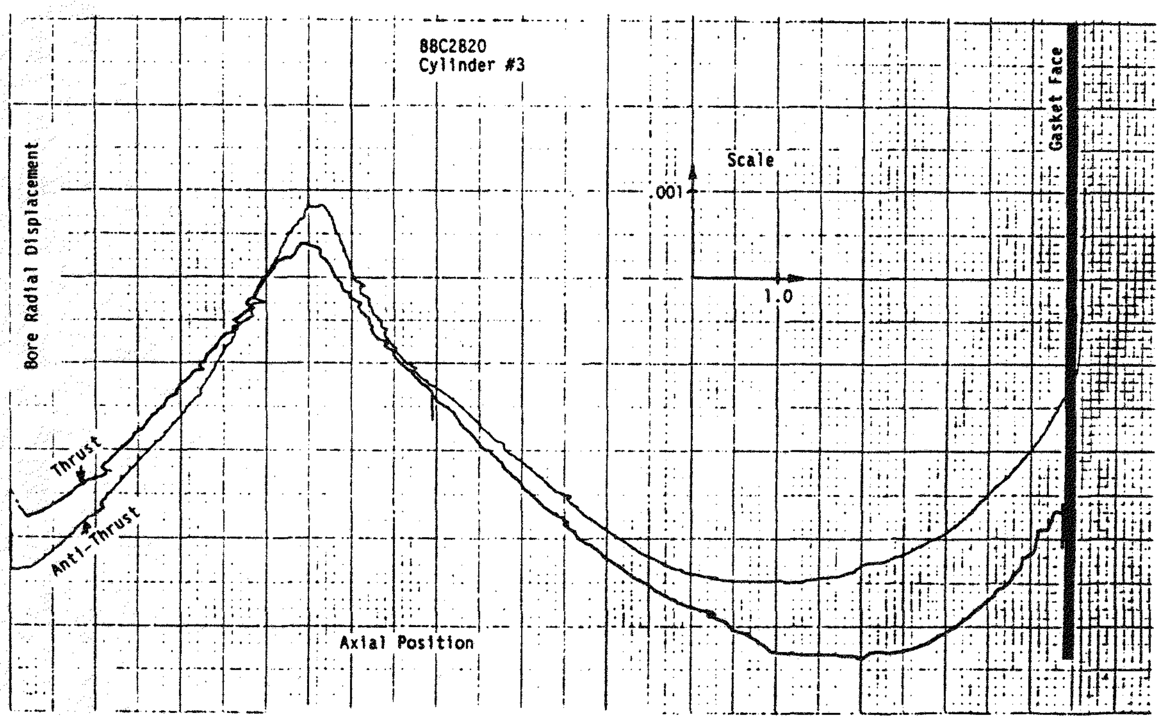

Figure 75. Cylinder bore profile - cylinder 3.

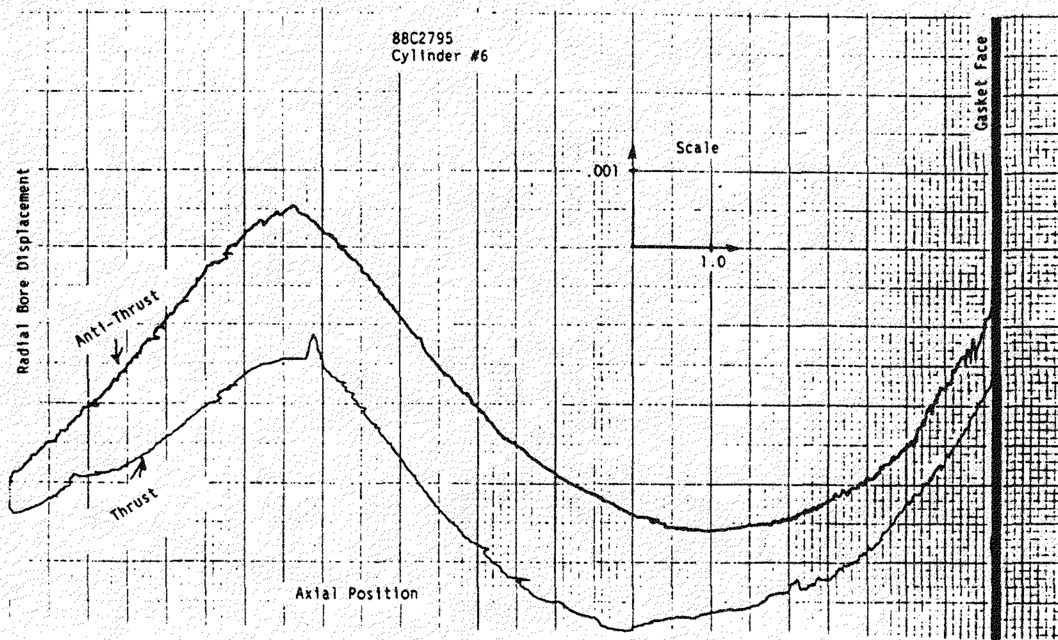

Figure 76. Cylinder bore profile - cylinder 4.

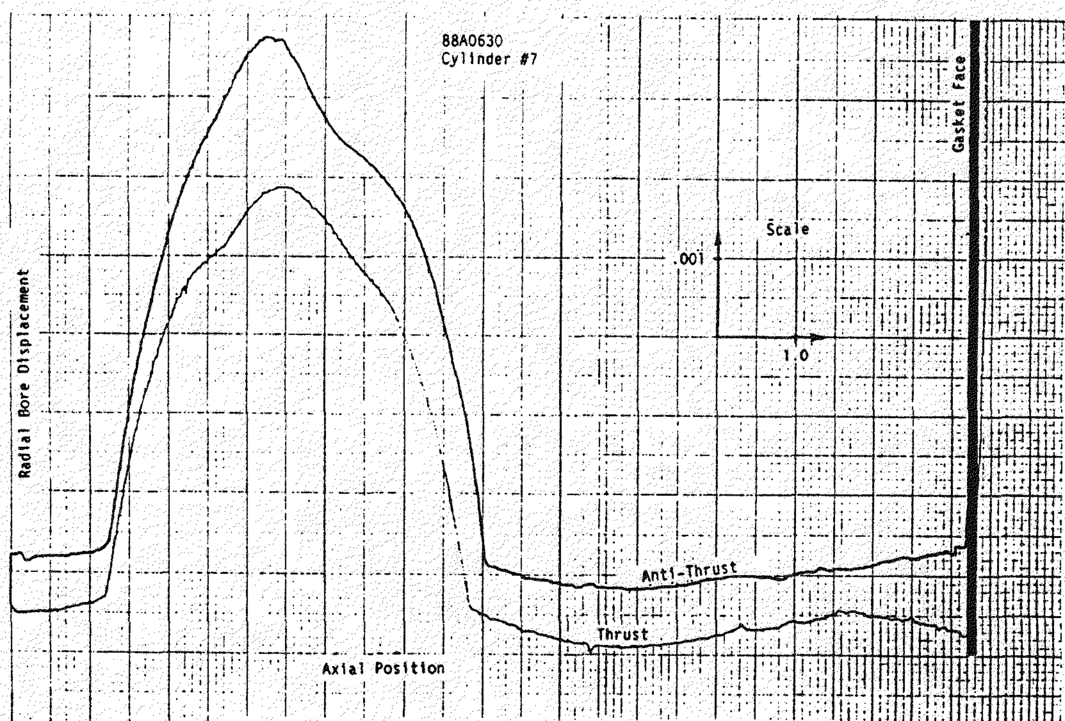

Figure 77. Cylinder bore profile - cylinder 5.

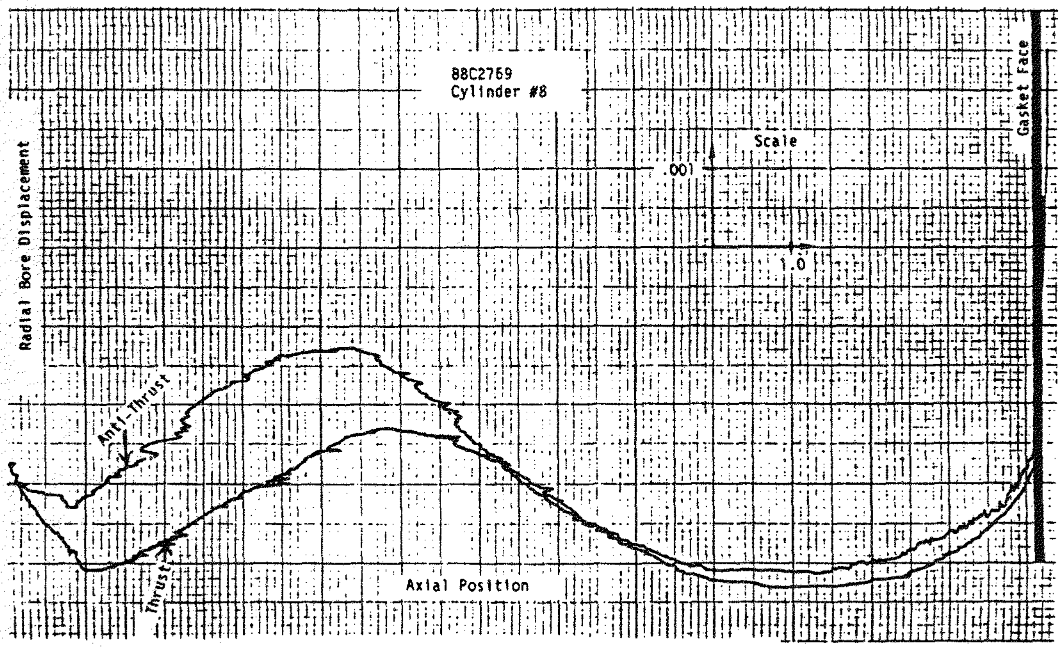

Figure 78. Cylinder bore profile - cylinder 6.

100-Hour Visual Inspection

Visual inspection of the power assemblies after the first 100 hours revealed problems with the chromium carbide coated compression rings. Nos. 1 and 2 compression rings in No. 1 cylinder had the coating chipped off the ring face. Predominant loss of material occurred near the ring tips. Evidence of vertical scratches on the other coated rings in the other cylinder locations suggested that this problem was going to occur in other locations. It is hypothesized that the ring chipping and vertical scratches on the ring faces are created in the port relief area. This theory is substantiated by the visual distress of the ceramic coating around the liner ports with no evidence of bore distress in the other areas of the liners.

The port relief area of the coated liners had evidence of distress. On all four edges of the port openings there was ceramic missing with parent metal showing through. The port struts (vertical sections between port openings approximately 3/8 in. wide) had ceramic retained in the center of the strut approximately 5/16 in. wide. The port struts also had fine, closely spaced horizontal scratches, which are believed caused by the rings jumping through the port relief area. The upper bores of the ceramic coated liners appeared normal with no evidence of peeling or breakthrough. The standard EMD liners and ring sets appeared normal with no evidence of distress.

200-Hour Visual Inspection

A 200-hour visual inspection showed continuing deterioration of the Nos. 1 and 2 compression rings. Cylinder No. 1 rings continue to lose the chromium carbide coating. No. 2 ring has a section of missing plating that extends from

Table 13.
Test cylinder bore profile values.

Liner S/N	88C2868		98A2455		88C2820		89L2697		88L1000		88C2795		88A0630		88C2769	
	Cylinder 1		Cylinder 2		Cylinder 3		Cylinder 4		Cylinder 5		Cylinder 6		Cylinder 7		Cylinder 8	
Axial Location	Perpen	Parral	Perpen	Parral	Perpen	Parral	Perpen	Parral	Perpen	Parral	Perpen	Parrel	Perpen	Parral	Perpen	Parrel
1	9.0650	9.0648	9.0608	9.0607	9.0618	9.0644	9.0612	9.0613	9.0620	9.0616	9.0641	9.0643	9.0614	9.0613	9.0612	9.0624
2	9.0625	9.0619	9.0605	9.0606	9.0618	9.0624	9.0611	9.0613	9.0617	9.0614	9.0622	9.0618	9.0611	9.0608	9.6000	9.0610
3	9.0611	9.0604	9.0601	9.0605	9.0610	9.0616	9.0614	9.0612	9.0614	9.0611	9.0607	9.0604	9.0606	9.0607	9.5960	9.5990
4	9.0606	9.0601	9.0598	9.0603	9.0610	9.0615	9.0608	9.0610	9.0612	9.0606	9.0603	9.0599	9.0603	9.0604	9.5960	9.5950
5	9.0610	9.0605	9.0598	9.0599	9.0615	9.0620	9.0609	9.0612	9.0612	9.0609	9.0604	9.0603	9.0604	9.0605	9.6010	9.5980
6	9.0624	9.0616	9.0605	9.0602	9.0628	9.0631	9.0613	9.0615	9.0614	9.0611	9.0614	9.0614	9.0607	9.0609	9.6110	9.6060
7	9.0645	9.0637	9.0652	9.0603	9.0647	9.0645	9.0624	9.0626	9.0628	9.0624	9.0632	9.0632	9.0644	9.0649	9.6280	9.6180
8	9.0673	9.0661	9.0708	9.0709	9.0669	9.0666	9.0688	9.0685	9.0687	9.0687	9.0659	9.0659	9.0707	9.0714		
9			9.0686	9.0704	9.0666	9.0672	9.0676	9.0677	9.0681	9.0684	9.0666	9.0648	9.0685	9.0697		
10	9.0649	9.0645	9.0611	9.0613	9.0642	9.0652	9.0616	9.0619	9.0617	9.0616	9.0644	9.0650	9.0616	9.0618	9.6090	9.6190
11	9.0634	9.0629	9.0607	9.0613	9.0623	9.0636	9.0610	9.0614	9.0612	9.0612	9.0626	9.0631	9.0614	9.0615	9.6200	9.6250
12	9.0620	9.0616	9.0608	9.0612	9.0613	9.0626	9.0609	9.0613	9.0611	9.0611	9.0612	9.0615	9.0613	9.0616	9.6240	9.6260
13	9.0613	9.0608	9.0609	9.0611	9.0611	9.0622	9.0610	9.0614	9.0613	9.0612	9.0604	9.0607	9.0615	9.0617	9.6340	9.6290
14	9.0611	9.0607	9.0609	9.0614	9.0607	9.0616	9.0610	9.0613	9.0614	9.0613	9.0599	9.0602	9.0615	9.0616	9.6380	9.6350
15	9.0604	9.0604	9.0608	9.0615	9.0601	9.0607	9.0609	9.0612	9.0612	9.0612	9.0598	9.0598	9.0614	9.0615	9.6360	9.6300
16	9.0596	9.0604	9.0607	9.0611	9.0590	9.0598	9.0607	9.0610	9.0611	9.0610	9.0595	9.0595	9.0613	9.0614	9.6000	9.6030
17	9.0596	9.0609	9.0605	9.0611	9.0584	9.0590	9.0605	9.0609	9.0608	9.0609	9.0596	9.0597	9.0614	9.0612	9.5980	9.6080
18	9.0605	9.0622	9.0604	9.0606	9.0578	9.0589	9.0605	9.0609	9.0606	9.0607	9.0609	9.0608	9.0609	9.0612	9.5850	9.6050
19	9.0618	9.0641	9.0603	9.0606	9.0582	9.0600	9.0605	9.0609	9.0606	9.0608	9.0628	9.0628	9.0607	9.0614	9.6000	9.6350

the ring gap for a circumferential length of 1 in. Compression rings in cylinder No. 3 are also beginning to chip. Ring plating failure occurs both from the edge of the ring and from the center of the ring face.

Cylinder No. 1 is showing honing scratches in the upper bore of the liner. These scratches were not observed at the 100-hour inspection. The ceramic liner in cylinder No. 3 is beginning to form wear marks in the upper bore of the liner beginning about 1.5 in. down from the gasket face. These wear marks are 1 to 1.5 in. wide and extend for 2 in. in the axial direction. Scratches in these wear areas are in the axial direction. These wear marks are located over the Nos. 5 and 6 stud boss ribs. These wear patches could be caused by bolt-up bore distortion (maximum at this area) or by the chipped compression rings.

Cylinder No. 6 has vertical cuts across the compression ring faces. A wear patch was observed in the port relief area 90 deg to the left from the thrust axis of the liner.

All the standard EMD liners and ring sets show excellent wear characteristics with no evidence of distress.

At 262 engine hours the turbocharger on this engine experienced an oil leak failure. This turbocharger failure had no adverse effect on this wear test. During the turbocharger replacement, No. 1 power assembly was removed for the scheduled 300-hour dimensional inspection. Fig. 79 shows the chipping that occurred on the ring faces of this power assembly. Fig. 80 shows the coating removal that occurred on the edges of the port struts. Currently, these components are undergoing dimensional inspection with liner wear measurements enumerated in Table 14. Measured wear on this liner is within measurement error so no discernable wear has occurred in this liner. This liner and piston will be reused in the construction of the replacement power assembly for cylinder No. 1.

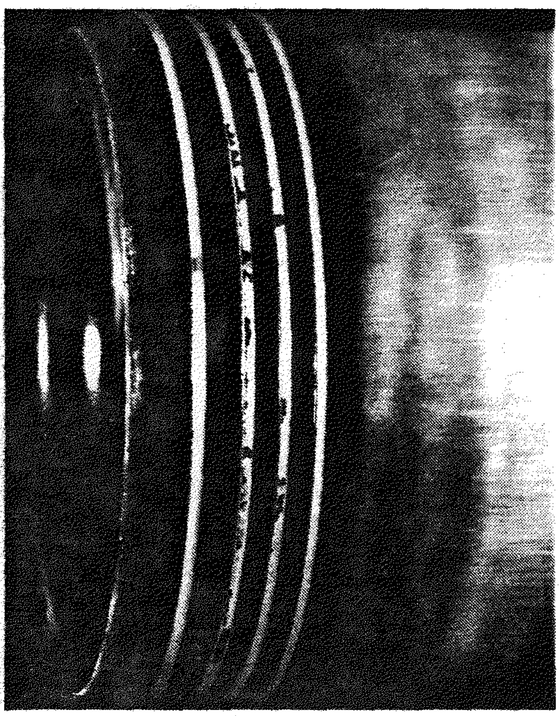

Figure 79. Ringface coating condition - 262 hours - power assembly 1.

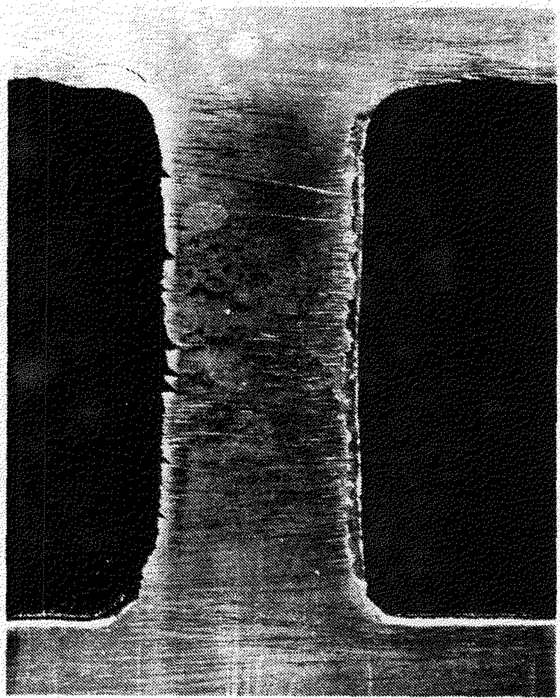

Figure 80. Liner port strut coating condition - 262 hours - power assembly 2.

Table 14.
Test cylinder #1 profile values after 262 hours.

Axial Location	Liner S/N 88C2808 Cylinder		Liner S/N 88C2808 Measured Wear	
	Perpend	Parallel	Perpend	Parallel
1	9.0648	9.0645	0.0002	0.0003
2	9.0624	9.0618	0.0001	0.0001
3	9.0610	9.0602	0.0001	0.0002
4	9.0605	9.0597	0.0001	0.0004
5	9.0609	9.0602	0.0001	0.0003
6	9.0622	9.0612	0.0002	0.0004
7	9.0644	9.0634	0.0001	0.0003
8				
9				
10				
11	9.0634	9.0626	0.0000	0.0003
12	9.0621	9.0612	-0.0001	0.0004
13	9.0612	9.0605	0.0001	0.0003
14	9.0611	9.0603	0.0000	0.0004
15	9.0605	9.0600	-0.0001	0.0004
16	9.0596	9.0599	0.0000	0.0005
17	9.0596	9.0604	0.0000	0.0005
18	9.0595	9.0617	0.0010	0.0005
19	9.0615	9.0637	0.0003	0.0004

Tribological Investigation of Materials Potentially Suitable for Coal-Fueled Diesel Combustion Chamber Components.

An experimental investigation was conducted by Adiabatics, Inc. using a high temperature friction/wear machine developed by them for evaluating the tribology of ring and liner materials in the products of coal combustion environment. This machine, described at the 1987 METC Heat Engines Conference (Koch, 1987), is designed to simulate the normal forces and relative motion at the ring and liner contact surfaces while exposed to the coal combustion environment. The objective of the program was to assist EMD in the selection of the best available wear resistant materials for a future coal-fueled diesel engine development program. Appendix A contains details of the Adiabatics, Inc. program.

The list of candidate materials was reduced to a final test matrix of combinations of the most promising materials. Adiabatics was not able to obtain all of the desired specimen coating materials for testing. Considerable effort was expended locating a source for these material combinations, and the supplier ultimately selected (Plasma Technics, Inc.) had limited success and was ultimately unable to deliver acceptable coatings. On each available combination, an 18-hour test was run. Combinations run and friction and wear results are tabulated in Appendix A. A summary of the results follows.

- The four tests with the lowest friction and lowest total wear in the products of coal combustion environment all ran with M2 tool steel specimens without coatings; therefore, this material was recommended for the piston rings.

- The four best liner coatings giving the lowest friction and wear results are:

  - experimental molybdenum
  - chrome oxide
  - tungsten carbide/cobalt matrix
  - tribaloy 800

- Interestingly, the "best current technology" combination chosen at the start of the investigation fell short in friction and wear and is not included in the "best" list.

- Adiabatics further concluded that running friction and wear tests in a coal combustion environment is difficult because the combustion deposits act as lubricants. Measured torque data showed a decrease in torque with the addition of coal and a torque increase as the coal burned out. A particular test showing the coal lubrication effect used the best current technology recommended cylinder liner and ring materials on roller and specimen respectively. This gave a coefficient of friction range of 0.36 to 0.40 without coal. The addition of a burning coal environment brought the coefficient down as low as 0.08 with a maximum value of 0.36 and an average of 0.164. The wear also decreased from a total wear rate of 9.4 mg/hr to 1.972 mg/hr, a more than fourfold reduction with the addition of burning coal.

In order to better understand the observed lubrication when running with coal, an additional test was run to determine whether the lubrication is due to the products of combustion or to the deposition of unburned coal powder on the friction surfaces. The test consisted of running the friction/wear machine at room temperature until the friction level stabilized and then slowly adding dry micronized coal powder to the test atmosphere. When the coal powder was added, the friction coefficient dropped immediately from 0.40 to 0.25. The addition of coal powder was then stopped and the machine was operated for an additional hour. The friction coefficient remained very constant at the 0.25 level. Examination of the specimen and roller at the end of this test revealed a thin layer of coal deposits on the wear surfaces of both parts. Adiabatics' conclusion from this portion of the testing was that at least some of the lubrication noted during the friction and wear testing could be due to deposition of unburned coal (soot), which acts as a lubricant on the wear surfaces.

These findings indicate that there mav be an advantage in designing coal-fueled diesel engine systems which exclude abrasive ash from the sliding surfaces while allowing lubricating carbon particles in the interface regions. Caution is advised in relying too heavily on the observed lubrication effect for wear protection; a well-developed coal-fueled engine with high combustion efficiency might make so few carbon particles that any coal lubrication effect might disappear.

Baseline Wear Study

Baseline DF-2 wear information on the best current technology piston ring and cylinder liner bore coatings is being gained by a 1000-hour test in an 8-645EB turbocharged engine located in a test cell at Electro-Motive Division's LaGrange, Illinois, facility. The engine is built with a split set of coated and standard assemblies. Cylinder positions 1, 3, 6, and 8 have Kaman Sciences SCA-1000 coated liners. All four compression rings in these cylinders are ductile iron with chromium carbide face coatings. Cylinders 2, 4, 5, and 7 are standard with hardened bore liners, the top three compression rings hard chrome faced, and the fourth taper faced filled groove ductile iron. The standard oil ring combination is used in all cylinders.

The following full initial ring and liner measurements have been taken:

- o Liner diameters parallel and perpendicular to the crankshaft centerline at 1-in. increments for the length of the liner bore
- o Liner profiles
- o Ring radial walls at five locations on each ring
- o Ring widths
- o Ring gaps
- o Piston ring grove widths
- o Piston skirt diameters

The same measurements will be taken after the conclusion of the test for comparison to the initial ones for wear assessment.

The 1000-hour test will be run on a simulated locomotive duty cycle, as follows:

- o 5 minutes, 200 rpm, idle, maximum cooling

- o 15 minutes, 650 rpm, 830 bhp, maximum heating rate to temperatures in next step
- o 30 minutes, 900 rpm, 1550-1650 bhp, 195 $\pm$5°F water inlet temperature
- o 30 minutes, 650 rpm, no load, maximum water cooling
- o Repeat cycle for a total of 750 times (1000 hours)

At all steps the air inlet temperature is being maintained at 90 +2°F. In the EDM cell control systems, the air inlet and water inlet temperatures are the major temperature independent variables. Water outlet temperature and oil temperatures are functions of the engine and the cell cooling systems, which are designed to perform as much like locomotive systems as practicable.

Cylinder liner bores were measured using two techniques before assembly in the test engine. All liner bore diameters were measured in 1 in. increments for the length of the bore perpendicular and parallel to the crankshaft axis using a dial bore gage. The liner bore profiles were also measured using an electronic transducer system. This transducer system graphically plots the radial displacement of the bore wall at a ratio of 1000 to 1 on the Y-axis with the axial location plotted at a ratio of 1 to 1 on the X-axis.

Wear evaluation of these liners and ring combinations consist of visual inspections of the power assemblies in the engine every 100 hours of running time and dimensional inspections during and at the conclusion of this test. One coated power assembly will be removed after 300 hours of running for dimensional evaluation. This power assembly will be replaced with new coated components, as necessary, and the test will resume. At 600 hours running time, a second coated power assembly will be removed from the engine for dimensional inspection. This liner will be replaced with a new coated assembly and the test will continue until the 1000 hours have been achieved. At the conclusion of the 1000 hours, all power assembly components will be disassembled and dimensionally inspected.

CWS Fueled Engine Wear Information

The wear information relating to an engine operating on a coal-water slurry will be gained in an EMD 2-645 engine during coal combustion development testing. One cylinder of this engine is to be fueled with CWS and only sufficient DF2 pilot charge to assure smooth combustion, while the other will be fueled with DF2 only. This engine will therefore provide an excellent back-to-back comparison of wear rates with CWS and with DF2.

The liners and rings of both cylinders will be coated components as described in the above section on the baseline wear test. The same piston, ring, and liner dimensions will be observed. In addition, other areas of the combustion chamber and its immediate environs will be monitored, in particular, valve face and seat and valve steam and guide wear. In this way we will gain direction for future wear reduction efforts in coal-fueled engines.

At this writing this test has not yet begun. The primary reason for the parts not yet being installed in the coal-fueled engine is the difficulty in getting liners as described above. Additionally, reliability and clogging problems that have held up testing have been experienced with the CWS injection equipment and with the apparatus which controls relative timing of the CWS and pilot charge injections on each stroke.

6.0 ECONOMIC STUDIES

There were 2 significant economic studies included in the work for this program. The first study, accomplished early in the program, was an assessment of the coal-fueled locomotive in comparison with a current diesel fueled locomotive with respect to the requirements such as fuel costs, maintenance costs, railroad acceptability etc. This study was published as a Topical Report titled "Coal-Fueled Diesel Locomotive Economic Assessment," dated October 1, 1986 and identified as DOE MC 22123-2281.

The second economic study addressed factors similar to the first study for six coal-fueled locomotive concepts other than coal-fueled diesel. The six coal-fueled locomotive concepts evaluated are:

Steam Turbine - Electric
Coal Gasifier - Spark Ignition - Electric
Coal Gasifier - Diesel - Electric
Fluidized Bed Steam Reciprocating - Electric
Coal Burning - Diesel Electric
Coal Gas Turbine - Electric

Summaries and conclusions from the two reports are presented in this report.

6.1 COAL FUELED DIESEL LOCOMOTIVE ECONOMIC ASSESSMENT

INTRODUCTION

Traditionally, the American railroad industry as a whole has been slow to embrace new and significantly different forms of motive power. In the industry's 150-year history only three distinct forms of locomotives have made an impact on regular service on mainline U.S. railroads.

- o steam--the mainstay between the early 1830s to 1957
- o straight electric--played a minor role from the turn of the century to the present time
- o diesel-electric--dominant motive power from the early 1950s to the present time

The reason for this stability is not an overly conservative reaction to change, but rather an everyday experience that demands rugged and reliable power to perform with little attention for up to a month at a time with almost no monitoring, inspection or maintenance. Present-day railroads operate trains, with up to six locomotives in a consist, that travel anywhere from a few hundred to a few thousand miles with only one engineer, whose time is occupied operating (driving) the locomotive. There is no one in the engine room keeping constant watch over the temperatures of every cylinder or other conditions that could give warning of impending equipment failure.

Electric locomotives, when compared to steam, were simple, rugged, and reliable. They could also pull more for a given horsepower rating than a steam locomotive of the same horsepower rating. This added pulling capability is explained by the following:

A reciprocating steam locomotive has a pulsating tractive effort that has periodic peaks and valleys. When maximum tractive effort exceeds the adhesion limit required to maintain the average tractive effort, the steam locomotive will slip. Once a slip occurs, the train will usually stall.

By comparison, electric drive series DC traction motors have nonlinear speed torque characteristics that give them a tractive effort proportional to the square of inverse speed (in theory). Thus, maximum and average torque are the same.

In early twentieth century America, the future of the electric locomotive seemed bright. However, the Great Depression and World War II prevented large scale electrification. After these two periods the diesel-electric was perfected. Unlike the electric system, the diesel did not require an expensive power distribution system and the locomotive was not restricted to operate only where that distribution system had been installed. Counter to the United States locomotive development, in Europe the electric locomotive gained wide acceptance because shorter distances and resulting higher traffic densities could justify the investment in the power distribution system.

Until the last decade, diesel fuel costs were a relatively small part of the operating cost of a locomotive. However, since the Arab oil embargo of the early 1970s, fuel costs have become the predominant operating cost. As a result, cost of the fuel as well as availability have become considerations for the railroads. The bulk of known oil reserves are outside United States borders. A great deal of the external known reserves are in the control of unstable political entities in the Middle East and South America. Any interruption of the oil supply from these external sources could be catastrophic to the economy and safety of the country.

To counter the questionable availability and potentially volatile cost of petroleum fuels, alternate locomotive fuels are being investigated. Within the boundaries of the United States, huge deposits of coal are being considered as a source of fuel. Until recently in this country, this coal was primarily used for stationary power plant applications and, as noted earlier, for reciprocating coal-fueled steam locomotives. However, research and development within the last five years has demonstrated that coal fuels may be a viable locomotive fuel.

Dr. Rudolph Diesel's original intent was to burn coal in his engine. As is well known, initially it did not work. Subsequent work from the early 1900s through the end of WW II in Europe did in fact demonstrate the feasibility of low speed, low power, coal-fueled diesels. The return of cheap petroleum fuel after WW II removed any incentive for further development of coal-fueled engines; a fact that persisted until the oil crisis of the 1970s.

Now that energy independence, coupled with possible balance of payments considerations, are deemed important, the potential for a coal-fueled diesel engine again should be considered. There have been significant advances in removing ash (probably the biggest cause of early coal-fueled diesel failures) and, to a lesser degree, sulfur from coal. This clean coal should enhance the possibility of success for a new round of coal-fueled diesel engines. The coal industry is continuing to improve the coal cleaning process as well as reduce the cost; newly developed and evolving wear and erosion/corrosion resistant material and coatings will likewise enhance the chances for success for a coal-fueled diesel engine. Thus, with the economic and political pressures and the technology developments, it is desirable to again attempt to develop the coal-fueled diesel for locomotive applications.

Other things being equal, the economic viability of a new product or concept will be the supreme judge of its success. Therefore, both the required technology and the economic viability must be shown. This report includes a first-order exercise in determining the economic merit of a coal-fueled diesel locomotive. The economic viability will be based on many assumptions as to the availability of suitable technology. For the purposes of this exercise, only economic factors are considered; politically instituted fuel shortages, etc, are not considered.

SUMMARY

This report provides evaluation of the economic considerations that must be made in determining the commercial viability of a coal-fueled diesel-electric locomotive. Of necessity, a great many assumptions must be made when locomotive hardware, fuel processes, and fuel specifications do not yet exist. However, assumptions made have been based on information already existing or on educated guesses about how experiences with other fuels (such as residual) have impacted the costs and operation of the Electro-Motive Division (EMD) diesel engine. Wherever possible, customer inputs were solicited and used when given. Assumptions about whether fuel will be bought preprocessed, processed in house, or what economic return on investment would be required to assume the perceived risk associated with a new technology, were all direct inputs from major U.S. rail carriers.

An engine that has been designed to successfully burn coal will probably be significantly different from today's diesel engines. New technologies created to cope with the challenges presented by coal fuels should trickle down to petroleum-fueled engines also.

Fuel refining in the coal industry is in its infancy at this time. Break throughs in processes or technologies that could make this fuel compatible with yet-to-be-designed engine hardware systems will have a direct bearing on the economic analysis of diesel engines burning coal for fuel.

Of necessity, the economic analysis of coal-fueled diesels will be an iterative process. The report presented here is an attempt to start the process with costs that are known or reasonably predictable at this time.

CONCLUSIONS

Based on the assumption noted in the body of this report and the results of the economic analysis, the following conclusions are presented:

- The difference between coal fuel prices and diesel fuel prices is the single biggest factor that will determine the economic acceptance of a coal-fueled diesel locomotive. As the spread between coal fuel and diesel fuel increased, the coal-fueled diesel locomotive becomes increasingly attractive.
- The coal-fueled diesel locomotive engine must be engineered to have a minimum two year life before overhaul.

The following conclusions are based on studies where diesel fuel was priced at \$5.86/million Btu (MMBtu) and the coal fuel price varied with its description. For the 50/50 coal-water slurry, fuel was priced at \$4.00/million Btu. It is this differential that must support increased maintenance, acquisition and operating costs, as well as an increase in the expected internal rate of return (IRR). Using these values for diesel and coal-water fuels, some conclusions are the following:

- o Using 50/50 coal-water slurry, a sensitivity to wear rates shows that cylinder life must be two years to attain a 21.9% IRR.
- o If acquisition cost can be held to no more than 15% above that of the conventional diesel-electric, then the IRR would be 30.2%.
- o Efficiency with coal is assumed to be as good as, if not better than, that with DF2. However, if efficiency were to drop by 10%, the IRR would be 15.3%.
- o Using coal slurried with methanol or DF2 produces a low or negative IRR. The worst case is a 50/50 coal-methanol slurry, which produces an IRR of -37.5%.
- o Dry powdered coal is expected to be more difficult to handle than coal slurry fuel. However, the price advantage should make further study of this fuel attractive. The IRR for coal powder fuel is projected to be 39.3%.

It can be seen from the analysis presented that coal-fueled diesel engines can be a viable alternative to oil-fueled engines. The problems to solve are primarily combustion efficiency, engine wear, fuel processing/pricing, and fuel handling.

If the technical problems can be solved, the IRR on investment will make the coal-fueled engine attractive from an economic viewpoint. The availability of coal versus petroleum will moderate the price of the oil fuel. The uncertainties of such a situation make the spread in price between coal and oil difficult, if not impossible, to predict.

6.2 A LIFE CYCLE COST STUDY OF SIX ADVANCED COAL FIRED LOCOMOTIVE CONCEPTS

Economic Summary

Economic utilization of coal in an environmentally acceptable manner has increasing importance as a replacement for petroleum derived fuels. The railroad industry will have to choose alternatives for their petroleum fueled diesel-electric locomotives as supplies of diesel fuel become more expensive and difficult to procure. This study is an economic comparison of several coal-fueled locomotive concepts based upon both present knowledge and assumptions of their technical feasibility. These concepts are analyzed using a life cycle cost method which compares costs and savings of two alternatives which are based upon the following:

1. Acquisition Cost
2. Fuel Cost
3. Maintenance Cost

From these costs, the relative IRR and present value of cost (PVC) for the two alternatives can be evaluated. The conceptual coal burning locomotives are compared against a standard, present production, Electro-Motive 8D60 model during the period 1995 to 2010. A description of the SD60 locomotive is given near the front of this report.

The basis for this cost study comes from companies which have proposed the concepts being analyzed. Assumptions were made whenever necessary information was not provided. This is particularly important for those locomotives which are only conceptual and have no data derived from prototype testing. Sensitivity studies around several of these assumption are given.

Figure 81 summarizes the important comparative, information for the six locomotive concepts studied. Figure 81 gives a description of the locomotives and the comparative economic and financial performance.

It should be pointed out that fuel price predictions are only estimated based on a stable economic environment. The sensitivity studies are intended to make comparisons should fuel prices vary radically from stable conditions.

COMPARISON OF CONCEPTS

The self-contained coal burning locomotives in this report are:

A. STEAM TURBINE-ELECTRIC
B. COAL GASIFIER-SPARK IGNITION-ELECTRIC
C. COAL GASIFIER-DIESEL-ELECTRIC
D. FLUIDIZED BED STEAM RECIP-ELECTRIC
E. COAL BURNING DIESEL-ELECTRIC
F. COAL GAS TURBINE-ELECTRIC

There are six concepts. The first four burn relatively inexpensive steam coal at $1.52/MBTU with relatively low efficiency. The coal fueled diesel and gas turbine both burn relatively expensive micronized slurry at $4.40/MBTU with higher efficiency engines.

Concepts II, III, IV and V are all based on a 3800 horsepower locomotive equivalent to an SD60 using high performance Super=Series wheel creep control. The coal burning diesel is used in a consist of three units which share one tender. The steam turbine is a 9000 horsepower unit on 18 powered axles using no advantage for high adhesion. The gas turbine is a 5800 HP unit with AC traction giving very high adhesion capability and 33% more powered axles. Together, this combination allows each gas turbine unit to replace 1.5 SD60's in tractive effort capability. The gas turbine operates in a consist of two units with one tender to replace three SD60's. The steam turbine is cost based on a 20 year life while all the other five concepts are assumed to have a 15 year life. This is a compromise between the twenty five year life claimed by Shoemaker and Associates and the fifteen year life assumed to be a standard. It is not unreasonable to assume twenty years based on the lower amount of untried components in this design.

CONCLUSIONS

Based on the assumptions made, none of the concepts has an IRR that looks attractive. The gasifier locomotive with a diesel engine has the highest IRR at 8.6% but this is short of its 30% hurdle rate goal.

	I	II	III	IV	V	VI
	Steam Turbine/ Electric	Gasifier/ Spark Ignition/ Electric	Gasifier/ Diesel/ Electric	Fluidized Bed/ Steam Recip. Electric	Coal Fueled Diesel/Electric	Gas Turbine/ Electric
	Shoemaker & Assoc.	Brobeck Corp.	Brobeck Modified	NSPC	GM/EMD	EMD/Allison Gas Turbine Division
Type of Coal	Crushed mine-mouth	Crushed mine-mth	Crushed mine-mth	Crushed mine-mth	Micronized Slurry	Micronized Slurry
Aux. Fuel & Supplies	Limestone, treated water	Sorbent, treated water	Sorbent, treated water	Limestone, treated water	None	Diesel Fuel
Combustion Process	Indirect (fluidized bed)	Indirect (gas producer)	Indirect (gas producer)	Indirect (fluidized bed) Wormser	Direct	Direct
Prime Mover	Steam turbine (BBC)	Spark ignition gas engine	Diesel engine w/high pressure gas injection	Reciprocating steam engine (EMD case & shaft)	Two cycle Diesel engine	Gas turbine - Open cycle
Auxiliary Power Source	None	None	None	None	None	Engine-generator set
Transmission	DC Motors No Super Series	DC Motors W/Super Series	DC Motors W/Super Series	DC Motors W/Super Series	DC Motors W/Super Series	AC Motors W/Super Series
Total Weight -	1,200,000	720,000	720,000	740,000	544,000 (X3)	674,000 (X2)
% Weight on Drivers	100	50	50	55	71	77
Number of Axles (driven/idle)	18/0	6/6	6/6	6/6	6/2 (X3)	8/2 (X2)
Power for Traction - HP	9000	3800	3800	3800	3800 (X3)	5800 (X2)
Number of Units in Consist	2	2	2	2	1-1/3 (X3)	1-1/2 (X2)
Length of Consist - Feet	210	130	130	140	88 (X3)	97 (X2)
Estimated Efficiency - % (Medium Duty Cycle)	12.5	24	17	18	36.4	27.6

Figure 81. Economic analysis of coal-burning locomotive concepts.

	I	II	III	IV	V	VI
	Steam Turbine/ Electric	Gasifier/ Spark Ignition/ Electric	Gasifier/ Diesel/ Electric	Fluidized Bed/ Steam Recip. Electric	Coal Fueled** Diesel/Electric	Gas Turbine** Electric
	Shoemaker & Assoc.	Brobeck Corp.	Brobeck Modified	NSPC	GM/EMD	EMD/Allison Turbine Division
IRR if DF2 = $1.50/gal.	35.8%	14.9%	46.2%	16.3%	41.2%	37.8%
IRR if coal = $1.00/MBTU	11.9%	-.4%	17.0%	19.1%	--	--
Acquisition Cost	**$5,973,500**	**$3,590,800**	**$1,878,100**	**$3,139,175**	**$1,790,400**	**$2,900,100**
Electricals	1,480,800	--	--	--	--	461,000
Engine Adder	1,200,000	1,862,400	95,800	-37,000***	95,800	1,215,000
Carbody/Trucks	815,900	--	--	--	--	665,000
Auxiliary Eqpt.	--	--	--	41,000	--	456,500
HP Compressor	--	--	53,900	--	--	--
Gasifier	--	200,000	200,000	--	--	--
Aux. Car	--	99,400	99,400	99,000	53,000	53,000
Fuel System	--	41,800	41,800	--	127,000	--
Emissions	--	--	--	--	100,000	--
Combustor	2,098,000	--	--	1,515,000	--	--
Condenser	350,000	--	--	134,000	--	--
Baseline Loco		1,375,000	1,375,000	1,375,000	1,375,000	--
Fuel Cost $/MBTU	1.52	1.52	1.52	1.52	4.40	4.40
Annual Fuel Cost $/Year	**$ 470,000**	**$ 162,000**	**$ 228,000**	**$ 218,000**	**$ 291,000**	**$ 450,000**
Av. Maintenance Cost $/Year	**$ 277,000**	**$ 75,000**	**$ 196,000**	**$ 183,000**	**$ 116,000**	**$ 188,000**
Replacement Ratio (XSD60)	2.79	1	1	1	1	1.5
Availability %	90	90	85	90	80	80
Highest Risk Technology	**Combustor**	**Gasifier**	**Gasifier & Engine**	**Combustor**	**Engine**	**Engine**
Design Difficulty of Critical System	**Medium**	**Medium**	**High**	**Medium**	**Very High**	**Very High**

*Based on the following SD60 costs: Acquisition = $1.375M, Maint. = $46,300/yr. av., Puel = $5.56/MBTU ($.72/gal.), Annual Fuel = $327,800
Availability = 95%

JOS - 11/1/88

**These 2 concepts will show significantly improved IRR's if CWS costs are reduced as discussed in Section 6.2 Conclusions.

***The additional engine expense for this concept is shown as a deduction. The steam reciprocating engine has no ignition requirements and the peak operating pressure is much lower than that of the diesel.

Figure 81. Economic analysis of coal-burning locomotive concepts.

If the price of diesel fuel were to rise from $.72/gallon to $1.50/gallon or conversely, CWS prices drop significantly, some of the concepts would look more attractive. In this situation:

- o The Steam Turbine IRR of 36% surpasses its hurdle rate of 25%
- o The Gasifier/Diesel IRR of 46.2% surpasses its hurdle rate of 30%
- o The Coal Fueled Diesel IRR of 41% surpasses its hurdle rate of 40%
- o The Coal Fired Gas Turbine IRR of 38% approaches its hurdle rate of.40%

The gasifier spark ignition locomotive has an extremely large, high-displacement engine which is responsible for a locomotive acquisition cost almost twice that of the gasifier diesel combination. This high acquisition cost is responsible for this option's poor economic performance. The fluidized bed steam recip-electric is an expensive locomotive partially due to the cost of the fluidized bed combustion (FBC) and also electric transmission is retained. This locomotive could be built without the electric transmission at a perhaps lower cost and show a better economic performance.

REDUCED COAL PRICE

Four of the concepts burn relatively unrefined steam coal. The economic studies are based on a coal fuel price of 1.52/MBTU. It is conceivable that some operators could obtain this fuel at a much lower price. IRR of the four concepts which burn steam coal was therefore shown for a coal fuel price of $1.00/MBTU. In this situation IRR improves over the baseline case; however, it is nowhere near as attractive as the situation when the price of the petroleum based fuel is almost doubled.

It should be noted that while the Coal Fueled Diesel and Gas Turbine show very good IRR at high prices for DF2, they also have, perhaps, the most difficult technological problems to solve with regard to wear, deposition and corrosion. These problems are all being addressed at this time; however, programs are not far enough advanced to know their outcomes.

The direct fired diesel and gas turbine have the highest efficiencies, especially the diesel. Their IRR is hampered at this time by the high price of the coal fuel after processing. Breakthroughs in fuel processes may yet bring the price down for the coal fuels which these two engines could burn.

1989 Update:

Particular attention should be given to the estimated fuel price predictions used in this analysis. Recent technological progress in coal water slurry processes suggest that the $4.40/MBTU cost used in this study may be considered either too high or viewed more as a worst case situation. Concepts V and VI in Fig. 81 (respectively the diesel and gas turbine prime movers fired directly with coal water slurry) are particularly sensitive to these cost reductions and become more economically attractive than what is presently indicated in the summary table.

7.0 CONCLUSIONS

This is a multifaced project involving several different subjects and activities. As a result, the conclusions are also multifaced in nature and will be presented in different groups and will be limited to the technical subject matter covered in this report.

There are several conclusions in technical details described when the test results are presented in Section 4. Even there, the major trends were just mentioned as observations from the data plots. Those conclusions which are in detailed technical nature will not be repeated here. The major accomplishments of the activities will be covered first, and then the conclusions about the major unexpected results will be emphasized.

7.1 MAJOR ACCOMPLISHMENTS

One of the most important accomplishments of the test activities at EMD is the development of information about the combustion process of the coal water slurry. Although there is a large volume of information available in the literature, they were like example demonstrations of what one can expect, rather than a detailed systematic trend analysis. With this report, the effects of the pilot timing, pilot fuel amount, CWS slurry injection timing, engine speed, inlet air heating and the water injection are investigated and reported in detail. The effects of the CWS amount on the combustion of the slurry was also investigated but not reported as the data reduction would require considerable amount of additional time.

Special emphasis was placed on understanding the effects of the water injection through the main injector on the combustion of the fuel injected by the pilot injection. This study gave one further insight on what to do and what not to do for burning the slurry in an engine environment. Detailed data was presented on the subject.

A slurry storage system designed in the earlier phase of this project was utilized over an 11-month period and information about the storability of the slurry with this system was developed, through monitoring the essential characteristics of the slurry. It was proven that this particular storage system could preserve the basic characteristics of the slurry for about six months.

A slurry transfer and changeover system was designed, built, maintained and utilized as part of the actual engine tests. After debugging the systems problems, it was used successfully for several months. This is a system closely simulating the perceived operation within a CWS burning locomotive.

The engine allocated to the project was operated using coal water slurry for over 3.5 hours, with the backup injection system, without any major problems. Several important engine operation conditions are worth mention. The engine is operated at top speed (900 RPM), at maximum load possible with the present injector and CWS, with a minimum amount of pilot injection, about 5.12% of the energy content of the total fuel. After the elimination of the system's mechanical problems and understanding the basics of the slurry combustion, there was no real problem in achieving these important milestones within the first or second trial. The EMD unit injector, after modifications for injector cooling and its clearances, performed better than expected in the engine tests. With the first injector, more than three hours of actual CWS injection and combustion time was achieved without any observable deterioration of the injector performance. More than 50% of this time, the injector was injecting slurry near full capacity. Detailed study of the injector wear was not completed at this writing, but the trend raises cautious hopes for a reasonable success.

Against these major accomplishments, the project was not immune of some very serious problems. The most important one was the attempt to develop a new injection system, satisfying several stringent requirements for coal water slurry combustion in diesel engine. The CWS injection capacity of this sytem was closer to those of the perceived locomotive application. The system is essentially a fuel pump and an injection nozzle system. When it was built, several mechanical problems were surfaced. After debugging most of these problems, the system was discontinued mainly due to the plunger sticking problem occurring immediately when the system is switched to slurry. The objective of the project was fulfilled by the backup system, planned at the beginning of the project for such an emergency.

7.2 MAJOR UNEXPECTED RESULTS

As mentioned before, there are numerous conclusions about the technical details of the CWS combustion and the effects of several parameters on it. They are not repeated here. However, there are some major trends observed in these tests which need to be brought out clearly. These are highlighted in this section.

One of the important results is about the possibility of having combustion in the cylinder near TDC, and as a constant temperature heat addition process for some reasonable and non-negligible range of crank angles. The conditions for this possibility is not well established for this present engine, for a wide range of engine speed and load. However, using the evaporation characteristics of the slurry and adjusting the pilot fuel and CWS injection timings (and amounts), a wide range of constant temperature combustion may be possible. This will improve the efficiency of the engine, as is well known. How much improvement can be expected is not known. Further study and investigation of this concept is needed.

Another important point centers around the following technical question, "Is there any inherent thermodynamic advantage of 2-cycle or 4-cycle engine operation for burning the coal water slurry more efficiency and more easily?". This question is very important because if there are any inherent advantage that cannot be overcome by detailed design considerations, the R&D resources allocated on the wrong alternative may not be so fruit bearing. On thermodynamic considerations, mainly high thermal loading of the two-cycle engines and the heating requirements for the evaporation of the water content of the slurry, one would expect some preference for 2-cycle engine operation. The test data has shown another, very important evidence that is useful for engine operation with CWS. It shows that when the slurry amount is increased, the combustion of the pilot is advanced by itself, which in turn advances the start of combustion of the slurry. This is attributed to the increase of the chamber gas temperatures due to the leftover residue gases form the previous cycle within the cylinder. In a 4-cycle engine operation, for the same pilot amount and timing (and the main fuel timing), increasing the slurry amount would usually cause a retardation of the slurry combustion and is not desirable. Of course, this concern may be alleviated through innovative design modifications, but realization of this particular feature may be an important characteristic for burning coal slurry. Of course, this observation needs further checks and detailed, focused study and confirmation.

It is also important that in these tests the effects of the inlet air heating was not found to be a big factor influencing the combustion of the slurry. Usually, it was expected that air heating would be critical when the conditions in the cylinder is very close to the ignitability limit of the slurry. One way to achieve this is to have a small amount of pilot. In the present test, the combustion energy supplied by the pilot was about 13% of the total. Although it was not very low (or high), the inlet air heating was not changing the combustion characteristics of the slurry much. Due to high thermal loading of the 2-cycle engines, the effects of inlet air heating may not be as influential.

Another important concept developed within this work is the study of the ignitability of the slurry by injecting water into the engine cylinder, calculating the cylinder gas temperatures from cylinder pressure traces and then comparing them to the ignitability limit of the slurry.

8.0 RECOMMENDATIONS

The following recommendations are being presented from the standpoint that the test program is being terminated by the Department of Energy (METC) before completing the stated objective. The status of the CWS combustion research effort and the ceramic coated engine component 1000 hour baseline test are as follows:

> The 2-cylinder 645 research engine has operated for a cumulative time of 3 hours and 32 min (23 + 46 + 86 + 57 minutes without problem) before encountering any mechanical problems. Further, sustained CWS combustion has been achieved with very near full CWS injector capacity with only a little over 5% of the total energy from the pilot injection.
>
> The baseline test for ceramic coated piston rings and liners operating on DF2 has completed 262 hr of the 1000 hr of planned operation. Problems are being encountered regarding the integrity of the ceramic coating/base metal interface bonding.

Regarding these two above listed tests, the following is generally recommended:

1. The CWS combustion work has progressed to the point that the approach being investigated is worthwhile, productive, and 10 cumulative hours (or more) of operation with significantly longer periods of uninterrupted operation appears reasonably attainable; therefore, it is reasonable to recommend that further combustion work be done.

2. Problems encountered in both the fabrication and baseline testing indicate that the present ceramic coated components are not good candidates for further wear resistant testing. A productive 10-hr wear test within the CWS combustion environment might be achieved if a set of rings and liner survive the baseline test (or are used in their present condition after 262 hours); however, it is quite evident that much work would have to be done regarding the metallurgy, materials sciences, and manufacturing techniques for these components.

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Appendix A

This appendix contains the pertinent tribological information generated by Adiabatics, Inc., Columbus, Indiana, in their work to identify candidate wear and corrosion resistant materials for evaluation in a coal-fueled EMD diesel engine. The information is excerpted from Adiabatics, Inc. Report No. FR85155-288 "Coal-Fueled Diesel Systems Research." Dated 26 February 1988 and submitted to Allison Gas Turbine Division.

The objective of this program was to assist the Electro-Motive Division of General Motors in the selection and implementation of the best available wear-resistant materials for piston rings and cylinder liners based on existing technology and then implement a bench test screening program to select advanced technology wear-resistant materials and to assist EMD in designing an advanced technology single-cylinder engine.

PROGRAM

Adiabatics, Inc., in consultation with Electro-Motive Division, examined existing EMD piston, ring, and cylinder liner designs and expected operating conditions. Based on these examinations, Adiabatics recommended the current technology wear materials reported in section 5 of the report. A second task for Adiabatics, Inc. was to select 30 tests for the high temperature friction wear test rig (shown schematically in Figure A-1) involving: materials, lubricants, temperature, stress, and coal combustion environment to yield the highest probability of identifying advanced technology, wear-resistant components that should be incorporated into EMD's single-cylinder test engine to be run on coal fuel. Adiabatics, Inc. was to use its current tribological data base to make those selections which best reflect the latest available technology. A portion of the selected tests were to be used to establish a baseline for the test rig including repeatability and degradation.

A test matrix was generated (Table A-1) which specified 30 recommended test combinations from a matrix of 9 piston ring materials and 12 liner materials. This matrix was modified several times during the program to reflect current thinking and material availability.

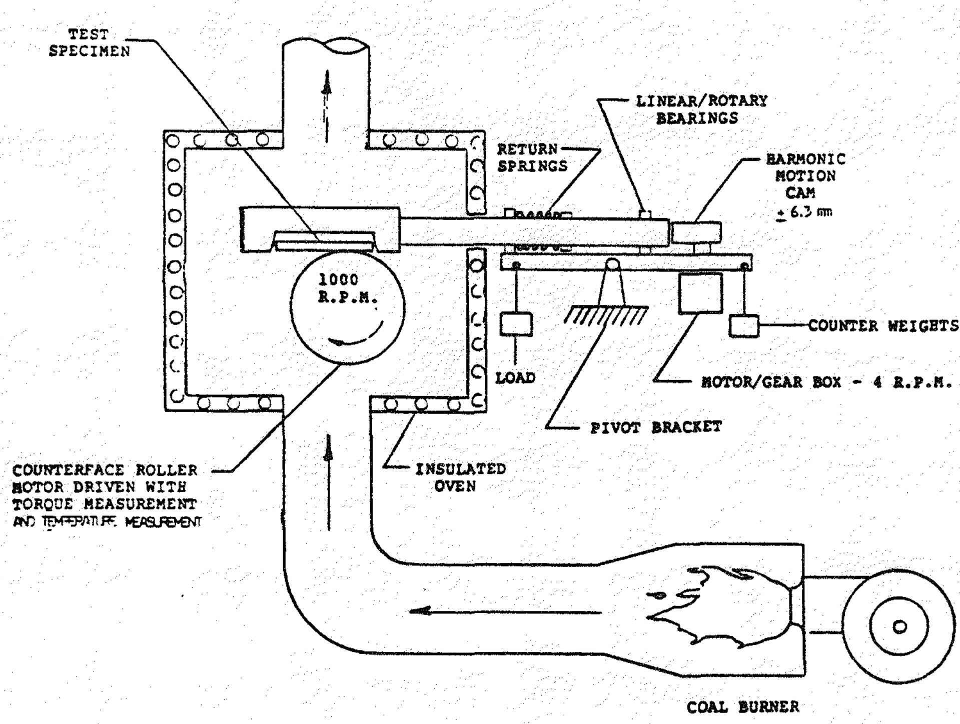


Figure A-1. Arrangement of friction and wear test apparatus using coal combustion products and ash for wear environment.

Table A-1.
Initial test matrix of the thirsty material combinations to be tested.

Steel + Lubricant Piston Ring (Specimens) / (Roller) Cylinder Liner	(Martensitic) + Iron	(Cu + PbO) + M2 Steel	(Cu + MoS_2) + M2	(MoO_3 + LiF) + M2	(T 700 + LiF) + M2	M2 Steel	(SDL-1) + Iron	(Cu + LiF)+M2 Steel	Carpenter 709	
Cast Iron (CI)	□ Base-line	○	○	○	○	○	□	□ Base-line	○	
CI + METCO 106 Plasma Sprayed Cr_2O_3	○ OR	□ OR	□ OR	□ OR	□ OR	□ OR	○	○	○	
CI + METCO 136 Plasma Sprayed Cr_2O_3	○	□	□	□	□	□	○	○	○	
CI + Tribaloy 800	○ OR	□ OR	□ OR	□ OR	□ OR	□ OR	○	○	□	
CI + T 700	○	□	□	□	□	□	○	○	□	
CI + Mo or Alloy	○	□	□	□	□	□	○	○	○	
CI I Ti 318	○	□	□	□	□	□	○	○	○	
CI + CerMets	○	○	○	○	○	□	○	○	○	
CI + Stellite 1	○ OR	□ OR	□ OR	□ OR	□ OR	□	○	○	○	
CI + Stellite 6	○	□	□	□	□	□	○	○	○	
CI + Aluminum Titanate	○	○	○	○	○	○	○	○	○	
Haynes 25	○	○	○	○	○	○	○	○	○	

□ Strongly Recommended

○ If Time and Money Permit

Total □-30 Tests

This task consisted of procuring the rollers and specimens per Task 2.1 and conducting the testing and analyzing the results. Figure A-1, A-2, and A-3 illustrate the principle of the test machine and include photographs of the test machine and a typical test setup. Table A-2 is an update listing of the test matrix showing the tests actually performed with the numbers in the listing representing the test number. Table A-3 is a listing of the test results showing the wear rates for the rollers and specimens and the coefficient of friction data. Figure A-4 is a graphical presentation of the data. As is obvious from comparing Tables A-1 and A-2, we were not able to contain all of the desired specimen coating materials for testing. Considerable effort was expended in locating a source for plasma spraying these material combinations and the supplier finally selected (Plasma Technics, lnc.) had limited success and was unable to deliver acceptable coatings. This placed severe schedule and monetary restrictions on the program causing two program schedule extensions without additional funding.

CONCLUSIONS

Based upon the test results we have concluded that running a friction and wear test in a coal combustion environment is difficult because the deposits act as lubricants. Figure A-5 is a typical chart recording of the temperatures of the specimen and roller and the drive torque required to turn the roller. The chart is annotated to show when additional coal is added to the burner and to note any other disturbances of the system. As is obvious from the torque trace, the addition of coal causes the torque to decrease and that the torque increases as the coal burns out. Visual observation of the burner shows that the smoke level increases when coal is added and decreases to a very low level when the burning coal bed becomes uniformly hot. Our hypothesis is that the smoke is indicative of the amount of airborne soot which deposits on the wear specimens and acts as a lubricant. In order to test this hypothesis the last series of tests was run with the same material combinations with and without coal combustion. Tests 25 and 27 were run with the Task 1.0 recommended ring and cylinder liner materials (rings coated with chrome carbide and the roller coated with the Kaman chrome-oxide (SCA 1000) process). The coefficient of friction running without coal ranged from 0.36 to 0.40, and the addition of the burning coal environment lowered the coefficient of friction to as low as 0.08 with a maximum of 0.36 and an average of 0.164. The wear also went down appreciably when the coal was burned, from a total wear rate of 9.4 mg/hr to 1.972 mg/hr which is more than a fourfold reduction.

In order to better understand the observed lubrication when running with coal, an additional test has been run to determine whether the lubrication is due to products of the combustion or from the deposition of unburned coal powder on the test surfaces. The test consisted of running the friction/wear machine at room temperature until the friction level stabilized and then slowly adding dry micronized coal powder to the test atmosphere. The dry coal powder used was Otisca bituminous from the Kentucky Blue Gem seam and has been used for coal-powder-fueled diesel engine tests at Adiabatics, Inc. The average particle size is 5 microns with 96% of the particles below 20 microns. The ash content is 0.8%. Immediately after adding the coal powder, the friction level dropped dramatically from 0.4 to 0.25. The addition of coal powder was then stopped and the machined operated for an additional hour with no additional coal added. Surprisingly, the friction level remained very constant at the

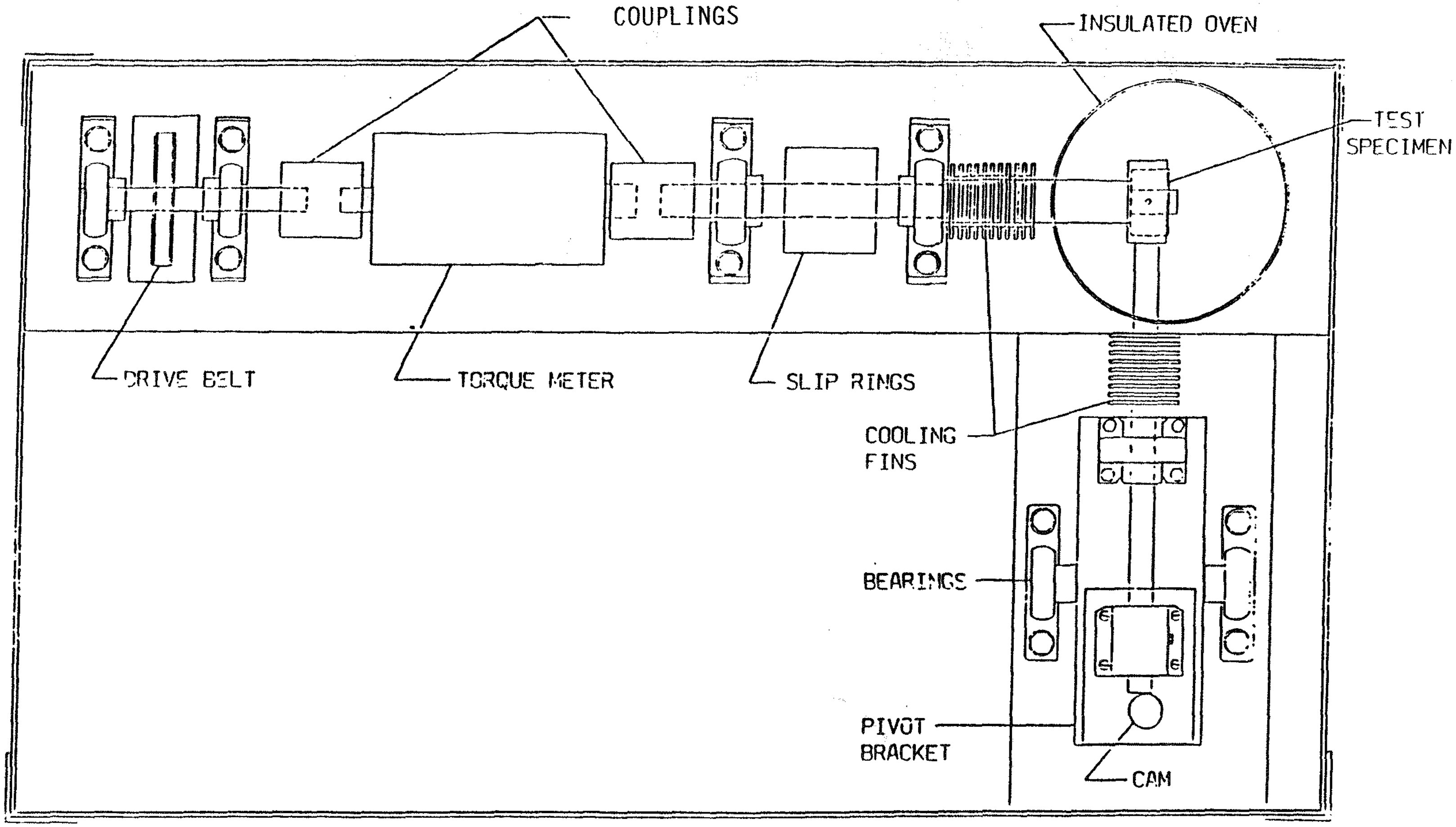
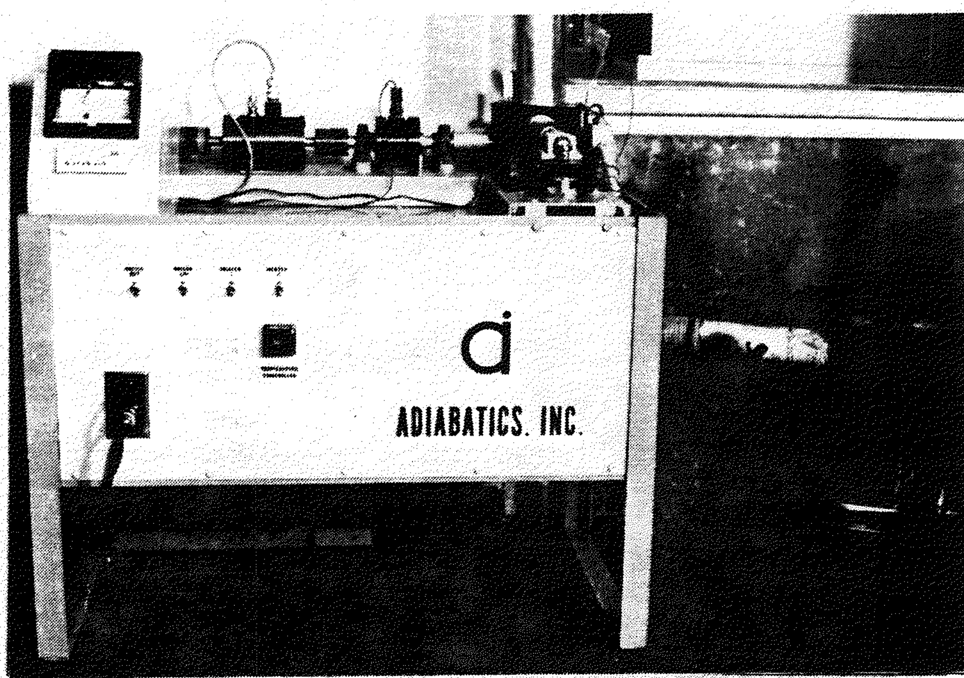


Figure A-2. Typical test setup of friction/wear test rig.



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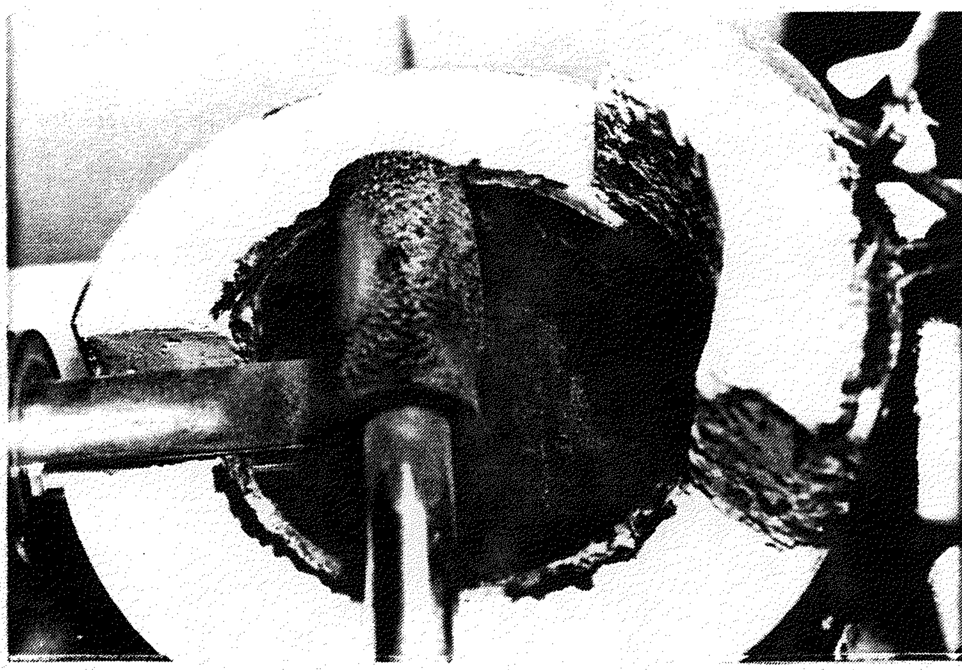

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Figure A-3. Photographs of the friction/wear test rig.

Table A-2.
Table A-2. Test Schedule.

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Specimens Steel + Lubricant (Piston ring) / Rollers (Cylinder liner)	CI (Martensitic)	(CuPbO)+M2 Steel	(Cu+MoS) + M2	(MoO +LiF) + M2	(T800+LiF) + M2	M2 Steel	CI(M)+SDL-1(Oil)	(Cu+LiF) + M2	Carpenter 709	Cr_2O_3	Cr_3C_2
Cast Iron (CI)	T-2 / T-3	○	○	○	○	○	□	□	○	○	○
CI + Cr_2O_3	○	□	□	□	□	T-12	○	○	○	○	○
CI + T800	T-7	T-17	□	□	□	T-4	○	□	T-16 / T-5	○	○
CI + Exp Mo	○	T-18	□	□	□	T-8	○	○	○	○	○
CI + Cr_3C_2/25 NiCr	○	○	○	○	○	T-9	○	○	○	○	○
CI + WC - 9 Co	○	○	○	○	○	T-10	○	○	○	○	○
CI + Stellite 6	○	T-19	□	□	□	T-6	○	○	○	○	○
CI + Al Titanate	○	○	○	○	○	○	○	○	○	○	○
CI + Ti 318	○	○	○	○	○	○	○	○	○	○	○
CI + Cr_2O_3 (W.R. Grace Co.)	○	○	○	○	○	T-1	○	○	○	○	○
CI + Cr_2O_3(SCA)	○	○	○	○	○	T-29	○	○	○	T-24	T-25 / T-27

□ **Recommended**

T-# Tests Completed, Test #

Table A-3.
Table A-3. Friction/wear test results.

TEST NO.	ROLLER MATERIAL	SPECIMEN MATERIAL	WEAR RATE (g/hr)		COEFFICIENT OF FRICTION		TEST CONDITIONS
			ROLLER	SPECIMEN	RANGE	AVG.	
1	Cr_2O_3 (W.R.G)	M2 Steel	.3253	.0005	.32	.32	air 900°F
2	Cast Iron	CI (Mart)	23.7	.104	.54	.54	air 120°F
3	Cast Iron	CI (Mart)	13.72	.071	.46	.46	coal comb. 350-400°F
4	Trib. 800	M2 Steel	.0008	.00016	.06-.44	.18	coal comb. 350-400°F.
5	Trib. 800	Carp. 709	.0013	.0006	.08-.48	.21	coal comb. 350-400°F
6	Stellite 6	M2 Steel	.0061	.0002	.10-.32	.204	coal comb. 350-400°F
7	Trib. 800	CI (Mart)	.0013	.0016	.06-.48	.20	coal comb. 350-400°F
8	Exp. Mo.	M2 Steel	+.00026	.00036	.08-.32	.15	coal comb. 350-400°F
9	Cr_3C_2/25NiCr	M2 Steel	.0036	.0006	.07-.42	.18	coal comb. 350-400°F
10	Wc-9Co	M2 Steel	.00037	.00055	.07-.52	.18	coal comb. 350-400°F
12	Cr_2O_3	M2 Steel	.000017	.00066	.08-.36	.18	coal comb. 350-400°F
16	Trib. 800	Carp. 709	.0006	.00113	.07-.47	.20	coal comb. 350-400°F
17	Trib. 800	M2+CuPbO	.00011	.00175	.10-.40	.22	coal comb. 350-400°F
18	Exp. Moly	M2+CuPbO	.00013	.0014	.08-.34	.196	coal comb. 350-400°F
19	Stellite 6	M2+CuPbO	.0054	.0010	.08-30	.202	coal comb. 350-400°F
24	Cr_2O_3 (SCA)	Duct. iron + Cr_2O_3	.00425	.00525	.32-.58	.44	air 450°F
25	Cr_2O_3 (SCA)	Duct. iron + Cr_3C_2	.0016	.0078	.36-.44	.39	air 450°F
27	Cr_2O_3 (SCA)	Duct. iron + Cr_3C_2	.000072	.0019	.08-.36	.164	coal comb. 350-400°F
29	Cr_2O_3 (SCA)	M2 Steel	No wear	measurem't	Reduction from .40-.24		coal powder ≈ 100°F

All tests were run at a roller speed of 1000 rpm and with a contact force between the specimen and roller of five pounds.

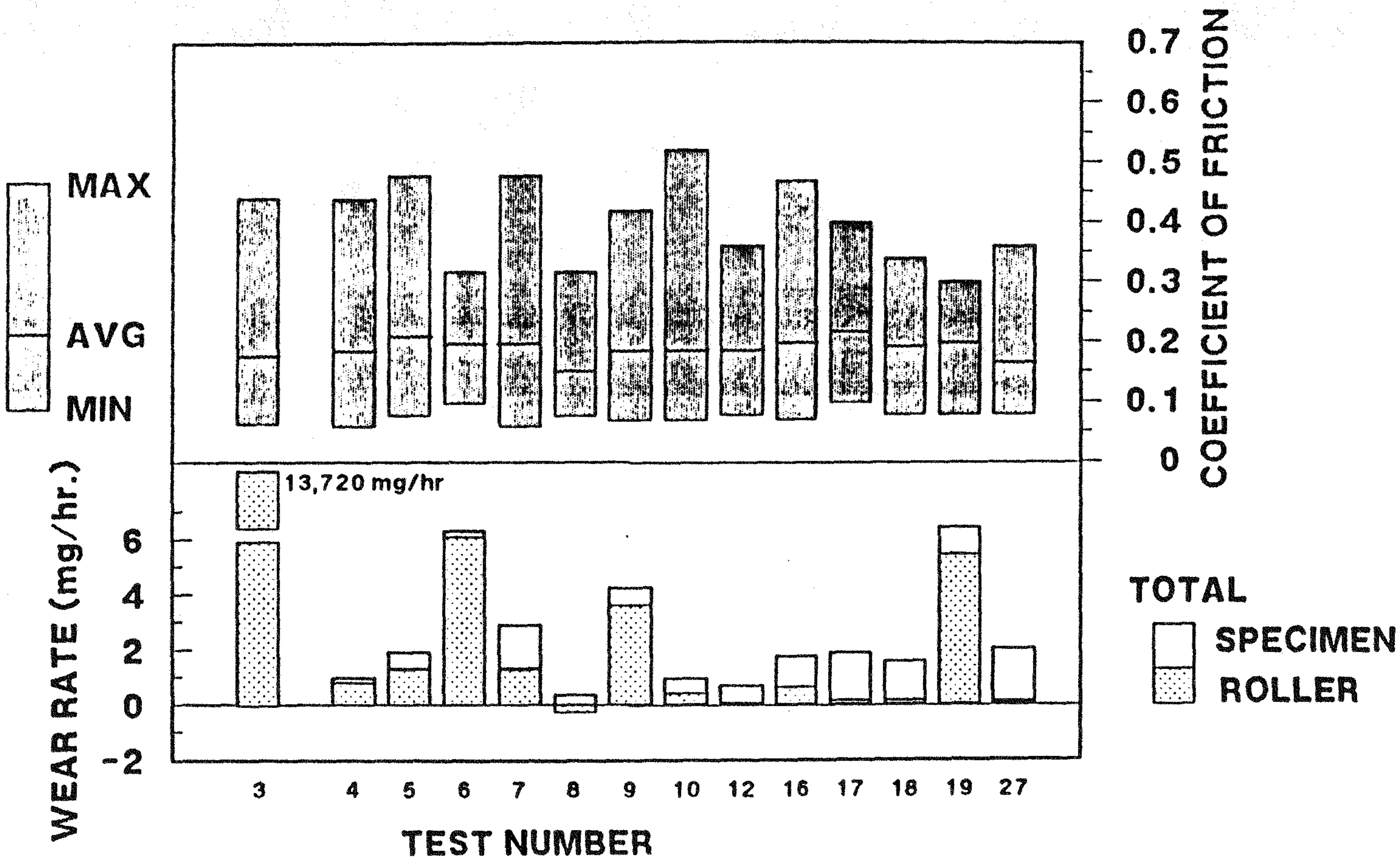

Figure A-4. Friction and wear test data.

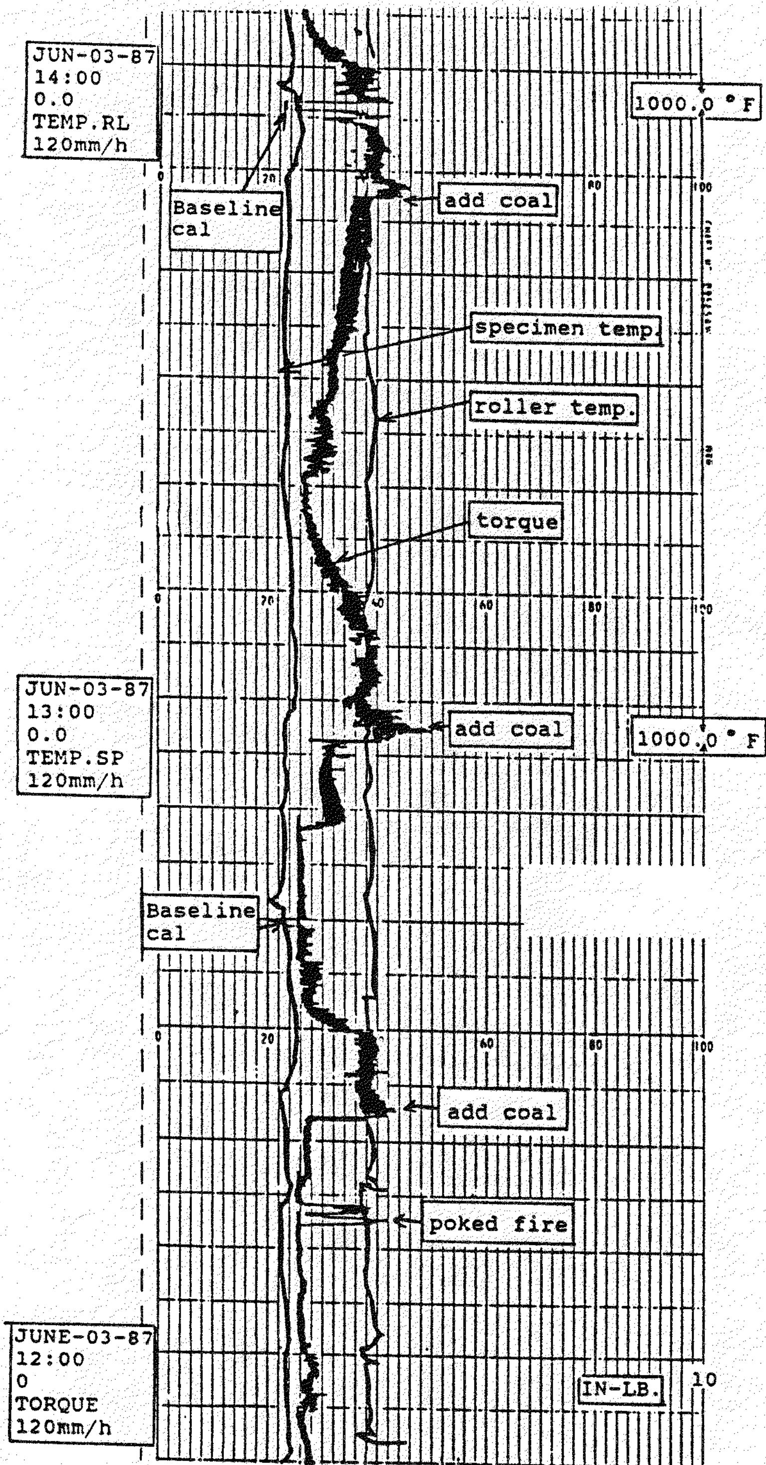

Figure A-5. Friction/wear rig coal combustion.

0.25 level. Inspection of the test specimen and roller at the end of this test revealed a thin layer of deposits on the wear surfaces of both parts. Our conclusion from this test is that at least some of the lubrication noted during the friction wear testing could be due to deposition of unburned coal (soot) on the wear surfaces which acts as a lubricant.

Based upon the tests performed in the coal environment, the following are the recommended coatings for the piston rings and cylinder liners for the coal engine:

Piston ring — The four tests with the lowest friction and the lowest total wear were all run with M2 tool steel specimens without coatings. Therefore this material is recommended for the piston rings.

Liner coating
1. Experimental molybdenum
2. Chrome oxide
3. Tungsten carbide/cobalt matrix
4. Tribaloy 800

The above coatings, with the first being the best, gave the lowest friction and wear results. The coatings were all applied by Stoody Deloro Stellite, Inc. of Goshen, Indiana.

RECOMMENDATIONS FOR FUTURE WORK

As is generally true with research projects into unexplored areas, more questions were generated than were answered. The question with the most profound consequences is whether the lubrication from the burning coal exists in the engine, and whether the results from the friction wear test machine correlate with engine tests. Basic questions also have been raised about the method of running the tests, especially in the area of how the coal is introduced and burned. The questions raised about the mechanism of coal lubrication raised by the short dry coal powder test are intriguing, particularly since a fairly long lasting dry lubricant film was apparently formed. Without additional data the question as to whether the best materials were good because they were compatible or because they allowed the coal lubricant to form a better lubricating surface will not be known. If, for instance, the lubrication film does not form in the engine, then perhaps some other materials should be recommended.

In order to answer these questions and others the following activities are recommended for future investigation:

1. Obtain the remaining, untried, test specimens and complete the original recommended test series.

2. Rerun each of the test combinations without the coal combustion (holding all other variables constant) to determine the friction and wear characteristics.

3. Run back-to-back or parallel tests of different material pairs in a coal combustion engine to determine whether the wear in the engine correlates to the wear testing either with or without the coal combustion.

4. Perform analytical studies of the surface chemistry of the specimens and rollers from the friction and wear testing and from the engine testing to determine the constituency of the deposits and whether they vary with different substrate materials.

5. Prepare test specimens and engine components with chemically active surfaces (catalytic coatings such as platinum would be used) to determine if the deposition and lubricating characteristics can be improved upon.

6. Modify the coal introduction and burning system to improve test repeatability and eliminate the need for constant monitoring. The scheme recommended would be to use a closed loop coal burner with constant dry coal powder feed rate and automatic control of the combustion airflow rate to hold the test specimen temperature constant.

APPENDIX B.

Definitions of Abbreviations/Terms Used in this Volume

CWS	Coal Water Slurry
DF2	Diesel Fuel No. 2
TDC	Top Dead Center
BDC	Botton Dead Center
ATDC	After Top Dead Center
BTDC	Before Top Dead Center
SwRI	Southwest Research Institute
AMAX	AMAX Coal Company
AR	As Required
BTU	British Theraml Unit
lb.	Pound (weight)
wt.	weight
cP	Centipoise
ppm	Parts Per Million
EMD	Electro-Motive Division, General Motors Corp.
GM	General Motors Corporation
CA	Crank Angle
cm	Centimeter
Inc.	Incorporated
KIVA	Name of computer program referenced in (18)
FLUENT	Name of computer program referenced in (2)
KIVA-COAL	Name of computer program referenced in (10)
EGR	Exhaust Gas Recirculation (into the cylinder)
P	Pressure
psia	Pounds per Square Inch Absolute

R	Rankline
Cyl	Cylinder
min	Minute
mm	Millimetre
in.	inch(es)
cu.in.	Cubic Inch
rpm	Revolutions per Minute
F	Farenheit
KJ	Kilojoule
KG	Kilogram
deg	Degrees
AVL	AVL List GMBH Kleistrasse 48, A-8020 Graz, Austria
exh	Exhaust
Temp	Temperature
SCFM	Standard Cubic Feet per Minute
psig	Pounds per Square Inch, Gauge
HR	Heat Release