Coal-Fueled High-Speed Diesel Engine Development

**Annual Technical Progress Report
October 1990 - September 1991**

November 1991

Work Performed Under Contract No.: DE-AC21-90MC27222

**For
U.S. Department of Energy
Office of Fossil Energy
Morgantown Energy Technology Center
Morgantown, West Virginia**

**By
Detroit Diesel Corporation
Detroit, Michigan**

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Morgantown, West Virginia 26507-0880**

**By
Detroit Diesel Corporation
13400 Outer Drive, West
Detroit, Michigan 48239-4001**

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ACKNOWLEDGMENTS

Program Manager: R. M. Kakwani

Technical Contributors: Detroit Diesel Corporation
R. M. Kakwani
R. E. Winsor

Southwest Research Institute
T. W. Ryan, III
J. A. Schwalb

Ricardo North America
S. Wahiduzzaman

Arthur D. Little, Inc.
R. P. Wilson, Jr.

TABLE OF CONTENTS

<u>Section</u>		<u>Page</u>
	LIST OF TABLES.....	v
	LIST OF FIGURES.....	vi
1.0	EXECUTIVE SUMMARY.....	1
2.0	INTRODUCTION.....	2
3.0	PROGRAM OBJECTIVES AND DESCRIPTION.....	3
4.0	ECONOMIC ANALYSIS AND MARKET ASSESSMENT.....	5
4.1	OBJECTIVES AND APPROACH.....	5
4.2	SELECTION OF ENGINE APPLICATIONS.....	6
4.2.1	Candidate 300-2,000 hp Engine Applications.....	6
4.2.2	Selection Criteria for Coal Fuel Feasibility.....	6
4.2.3	Screening of Engine Applications.....	7
4.3	MINE HAUL TRUCK APPLICATION	
4.3.1	Background.....	8
4.3.2	Mine Locations and Coal Slurry Logistics.....	9
4.3.3	Schematic Design of Coal Engine System for Mine Haul Truck.....	10
4.4	INLAND WATERWAY TOWBOAT APPLICATION.....	10
4.4.1	Existing Towboat Engines - Population and Fuel Usage.....	11
4.4.2	Towboat Operating Routes and Coal Slurry Logistics.....	11
4.5	EMISSION CONTROL SYSTEM FOR MINE HAUL TRUCK AND TOWBOAT ENGINES.....	12
4.6	ENGINE SPREADSHEET MODEL FOR LIFECYCLE COSTS.....	13
4.6.1	Engine Efficiency and Fuel Cost.....	13
4.6.2	Engine Capital and Operating Costs.....	14
4.7	ANALYSIS OF ECONOMIC FEASIBILITY.....	15
4.7.1	Mine Haul Truck Application.....	16
4.7.2	Inland Waterway Towboat Application.....	17
5.0	ENGINE MODELING.....	18
5.1	EFFECT OF COMPRESSION RATIO.....	19
5.2	EFFECT OF INJECTION TIMING AND DROPLET SIZE.....	20
5.3	UNIFORM VS. MULTIPLE DROPLET SIZES.....	20
5.4	EFFECT OF IN-CYLINDER INSULATION.....	21

TABLE OF CONTENTS

<u>Section</u>	<u>Page</u>
5.5 EFFECT OF SWIRL LEVEL.....	21
5.6 EFFECT OF INJECTOR ORIFICE SPRAY ANGLE AND SIZE.....	22
5.7 EFFECT OF CWS FUEL TEMPERATURE.....	23
6.0 DESIGN OF INJECTION AND COMBUSTION SYSTEM	
6.1 SELECTION OF THE CONCEPT.....	23
6.2 CWS INJECTION SYSTEM.....	24
6.2.1 Design of the 149 CWS Injection System.....	25
6.3 SET-UP TEST FACILITIES.....	27
6.4 TEST ENGINE PREPARATION.....	28
7.0 CONCLUSIONS.....	28
8.0 FUTURE WORK.....	29
REFERENCES.....	30

LIST OF TABLES

<u>Table</u>		<u>Page</u>
4.2-1	Evaluation Criteria for DDC Coal Engine Application.....	31
4.4-1	Inland Waterway Towboat Engine Database Results.....	32
4.5-1	Emission Control System Design Assumptions.....	33
4.6-1	Slurry Cost Summary.....	14
4.6-2	Coal-Fueled Engine Capital Cost.....	34
4.6-3	Oil-Fired Diesel Engine Capital Cost.....	35
4.6-4	Coal Engine Maintenance Cost.....	36
4.6-5	Oil-Fired Engine Maintenance Cost.....	37
4.6-6	Coal Engine Capital & Operating Cost.....	38
4.6-7	Oil-Fired Engine Capital & Operating Cost.....	39
6.1-1	DDC 149 Coal-Fueled Engine Injection & Combustion Concepts..	40

LIST OF FIGURES

<u>Figure</u>		<u>Page</u>
3-1	Program Schedule.....	41
4.3-2	Emission Control Device for Coal-Fired Diesel Mine-Haul Truck (2000 HP).....	43
4.4-1	1800 HP Viking Class Towboat.....	44
4.7-1	Mine Haul Truck Coal Engine Cost Savings: Coal Cleaning Cost/Load Factor Effects.....	45
4.7-2	Mine Haul Truck Coal Engine Cost Savings: Nozzle Life & Load Factor Effects.....	46
4.7-3	Mine Haul Truck Coal Engine Cost Savings: Interest Rate/Load Factor Effects.....	47
4.7-4	Mine Haul Truck Coal Engine Cost Savings: Diesel Fuel Use Effects.....	48
4.7-5	Tow Boat Coal Engine Cost Savings: Coal Cleaning Cost/Load Factor Effects.....	49
4.7-6	Tow Boat Coal Engine Cost Savings: Nozzle Life & Load Factor Effects.....	50
4.7-7	Tow Boat Coal Engine Cost Savings: Interest Rate/Load Factor Effects.....	51
4.7-8	Tow Boat Coal Engine Cost Savings: Diesel Fuel Use Effects.....	52
5-1	Comparison of Heat Release Rate from Predicted and Measured Cylinder Pressure Data.....	53
5.1-1	Effect of Compression Ratio: Autoignition @ 1900 RPM, 100% Load.....	54
5.2-1	Effect of CWS Injection Timing & Droplet Size: Autoignition @ 1900 RPM, 100% Load.....	55
5.2-2	Effect of CWS Injection Timing & Droplet Size: Autoignition @ 1900 RPM, 50% Load.....	56
5.3-1	Uniform & Multiple Droplet Size Distribution: Autoignition @ 1900 RPM, 100% Load.....	57

LIST OF FIGURES (continued)

<u>Figure</u>		<u>Page</u>
5.5-1	Effect of Swirl Level on CWS Burnout and Power Output: Autoignition @ 1900 RPM, 100% Load.....	58
5.6-1	Effect of Injector Orifice Size on CWS Burnout and Ignition Delay: Autoignition @ 1900 RPM, 50% Load...	59
6.1-1	Series 149 Coal-Fueled High Speed Diesel Engine (Concept #1).....	60
6.1-2	Series 149 Coal-Fueled High Speed Diesel Engine (Concept #2).....	61
6.1-3	Series 149 Coal-Fueled High Speed Diesel Engine (Concept #3).....	62
6.2-1	The CWS Injector Simulation Results at 1900 RPM, 100% Load Point.....	63
6.2-2	Pump-Line-Nozzle Injector for CWS.....	64
6.4-1	Photograph of the 8V-149TI Engine.....	65

1.0 EXECUTIVE SUMMARY

This report presents progress on a program (DOE/METC contract number DE-AC21-MC27222) to study the feasibility of operating a Detroit Diesel Series 149 engine at high speed with fuel injection and combustion systems designed for coal-water-slurry (CWS). The objective is to achieve engine operation for sustained periods over the load range at speeds from 600 to 1900 rpm with reasonable fuel economy and coal burnout rate. The 149 engine is being proposed for operation on CWS fuel for the mine-haul off-highway truck and work boat marine markets. Economic analysis studies indicate that the 149 coal-fueled engine applications for these markets could have sufficient operating cost savings, depending upon the future diesel fuel price, emission control system capital and operating costs, and maintenance and overhaul costs. A major portion of the maintenance costs is expected to be due to the anticipated lower life and higher cost of the CWS injectors.

Simulation results indicate that an average droplet size smaller than 20 - microns is important to obtain high carbon burnout at engine speeds of 1900 rpm. The results predict autoignition of coal fuel at various loads at 1900 rpm with an average droplet size of 18 microns and 19:1 compression ratio. Based on these results, the performance of the CWS injection system will be crucial in achieving acceptable coal burnout in the short time available in the 149 high-speed engine. The proposed engine will be modified to obtain high gas and surface temperatures in the cylinder, with the goal being the autoignition of coal fuel. This will be achieved by a combination of hot residual gases in the cylinder through control of the scavenge ratio with a bypass blower, ceramic insulated engine components and increased compression ratio. Ignition-improved methanol is being considered for the cold start, warm up and shutdown of the engine. A diesel pilot injector is a secondary option.

2.0 INTRODUCTION

During the past several years, DOE/METC has sponsored many programs to develop coal-fueled diesel engines. These efforts have made significant progress in developing technology for low and medium speed diesel engines capable of operating on these fuels. Currently, two major programs are being conducted by GE and A.D. Little/Cooper to commercialize coal-fueled diesel engines for locomotive and cogeneration applications, respectively. These programs have solved many of the difficult problems associated with burning coal in a diesel engine and have demonstrated single-cylinder engine operation on CWS. Excellent progress has been made in the areas of CWS injection and combustion at engine speeds of 1050 rpm (GE) and 300 to 450 rpm (Cooper). Also, the economic analysis indicates that coal-fueled engines can provide good return on investment assuming reasonable diesel fuel costs. Some of the problems associated with wear of the high pressure CWS injector nozzle holes, cylinder liners and piston rings are being pursued by GE, A. D. Little and other R&D companies (Webb et al., 1990). The problems faced by the medium-speed coal-fueled engines, namely, reliability of the fuel injection system, coal fuel ignition and combustion, and the wear of engine components, are expected to be more severe for high-speed engines operating on coal fuels. These problems need to be addressed by the use of innovative approaches to mixture preparation, ignition and combustion.

DOE/METC has also funded research programs to investigate combustion of coal powder and CWS in low heat rejection (adiabatic) engines. The objective of these programs was to enhance the coal combustion by higher in-cylinder gas temperatures achieved in these engines. In an uncooled single-cylinder prechamber diesel engine, combustion of fumigated coal powder fuel and CWS injected into the prechamber was achieved at speeds up to 1800 rpm (Kakwani et al., 1989 and 1990).

Detroit Diesel Corporation (DDC) is developing the Series 149 engine for operation on CWS fuel at high speeds for the mine-haul off-highway truck and work boat marine markets. The 149 diesel engine has a major share of these markets. Some of the economic and technical merits of a coal-fueled 149 engine are as follows:

- o Series 149 engines use a large amount of fuel. For mine-haul off-highway trucks (>170 tons), 16V-149TI engines are used at 1800 to 2200 hp. The average load factor for a typical mine-haul truck is about 35%, and it operates 5000 to 7000 hours per year. In the case of a work boat engine, the load factor is greater than 50% and there are 5000 to 8000 operating hours per year.
- o An increase in the engine cost due to the coal fuel injection system, wear-resistant components, and the emission control system should be tolerable on a Series 149 engine. In contrast, the cost of an on-highway coal-fueled engine would be expected to increase by an unacceptable amount compared to the cost of the basic diesel engine.
- o The 149 engine is a high-speed heavy-duty diesel engine with a large market share of the off-highway truck, industrial and marine applications. Therefore, a coal-fueled version of the engine has many possible applications.
- o A large off-highway engine, like the Series 149 engine, will be better suited to accept the additional space and weight requirements of the emission control system posed by the use of coal fuels. Also, emission regulations for off-highway are not expected to be as stringent as those for on-highway engines because off-highway trucks, industrial engines and marine vessels normally operate in remote areas away from population centers.

Based on this assessment and technical considerations, the 149 engine was selected for this program.

3.0 PROGRAM OBJECTIVES AND DESCRIPTION

The objectives of this program are to study combustion feasibility by running Series 149 engine tests at high speeds with a fuel injection and combustion system designed for coal-water-slurry (CWS). The following criteria will be used to judge feasibility:

- Engine operation for sustained periods over the load range at speeds from 600 to 1900 rpm. The 149 engine for mine-haul trucks has a rated speed of 1900 rpm.
- Reasonable fuel economy and coal burnout rate
- Reasonable cost of the engine design concept and CWS fuel compared to future oil prices.

The feasibility of running the 149 coal-fueled high-speed diesel engine is being investigated under a 2.5 year, four task program by DDC with Diesel Technology Corporation (DTC), Southwest Research Institute (SwRI), Ricardo North America and A. D. Little, Inc. (ADL) as subcontractors. Figure 3-1 presents the work tasks and schedule for this program. DDC is working on the injection and combustion system design and engine component development. DTC and SwRI are involved with the design and fabrication of the CWS injection system. SwRI will conduct the bench and engine tests by operating one cylinder of a DDC 8V-149TI engine on CWS fuel. Ricardo has performed cycle simulations of the 149 coal-fueled engine concepts. ADL has performed an economic analysis and market assessment for the 149 coal-fueled engine.

The proposed engine will use high gas and surface temperatures in the engine with the goal being autoignition of coal fuel. This will be achieved in the 149 engine by a combination of hot residual gases in the cylinder through control of the scavenge ratio with a bypass blower, warm inlet air, increased compression ratio and ceramic insulated engine components. Ignition improved methanol is being considered for the cold start and warm up of the engine. A diesel pilot injector is a secondary option. The CWS injection system design for the 149 engine is currently being finalized.

4.0 ECONOMIC ANALYSIS AND MARKET ASSESSMENT

4.1 OBJECTIVES AND APPROACH

The objective of this task was to identify the market potential and economic feasibility of the future coal-fueled, high-speed diesel engine. The results of this task were expected to determine what specific heavy duty engine and application(s) are most attractive for coal fuel, and also define basic economic factors for the engine to be competitive, such as:

- o Source coal location and price
- o Coal cleaning cost, given assumed CWS specifications
- o Engine parts cost increase to permit operation with coal fuel
- o Load factor (equivalent full load hours per year)
- o Maintenance and operating costs
- o Emission control system capital and operating costs and size
- o Diesel fuel costs at which the coal-fueled engine is cost competitive

Several applications for heavy duty diesels in the 300-2,500 hp output class were surveyed, including off-highway mine-haul trucks, construction and industrial, marine and generator set applications. A set of "attractiveness criteria" for coal fuel operation was developed to screen the full range of engine applications. This preliminary screening resulted in two applications which appeared particularly well suited for coal fuel:

- o Mine-haul trucks equipped with a 16V-149TI diesel engine of 1,600 to 2,200 hp. These 170-240 ton trucks operate in overburden and open pit mines including coal, iron ore and copper.
- o Barge, tug and work boat engines in the marine inland water-way sector. Most of the existing 4,000 engines are naturally aspirated with 800 to 2,000 hp and have relatively long service lives.

Next, more detailed information for these two selected market applications was prepared to better assess their feasibility and relative merit. This information included market share, duty cycle for engine operation, historical annual sales, emission control requirements, and specific geographical locations for

the engines now in use (which enabled us to select suitable coal mines nearby). A detailed economic feasibility analysis was then performed using a spreadsheet model for life-cycle engine and fuel costs. Parametric studies were performed to explore the relative impact of various parameters such as:

- o Load factor
- o Clean coal fuel cost relative to diesel fuel
- o Engine efficiency
- o CWS injector nozzle life
- o Financing rate

4.2 SELECTION OF ENGINE APPLICATIONS

4.2.1 Candidate 300-2,500 hp Engine Applications

Heavy-duty diesel engines in the 1,700-2,100 rpm rated speed class are widely recognized for their use in the on-highway trucks and buses, where they account for approximately 6% of U.S. fuel oil usage. However, there are other applications for heavy-duty diesels in which Detroit Diesel enjoys a significant market share. These applications are less widely known, but may be more suitable for adaptation to coal-water slurry fuel, such as:

- Off-highway, including mine haul trucks
- Marine propulsion, including inland waterway vessels
- Generator sets and industrial drives

4.2.2 Selection Criteria for Coal Fuel Feasibility

Previous work on clean coal fuels for reciprocating engines has shown that the emission control requirements and the total annual fuel usage per engine are two factors of primary importance for economic feasibility. Annual fuel usage is critical because the benefit for the engine operator will be the lower cost of coal fuel compared to diesel fuel oil. In brief, the engine must burn enough coal fuel each year to pay for the extra cost of emission control (which is technically feasible, but relatively expensive on a \$/engine basis). For example, an application with 1,500-2,000 hp engines operating 2,000 to 4,000 full-load equivalent hours/year and moderate emission control requirements will

be preferred to a 500-1,000 hp engine application with only 1,500 full-load equivalent hrs/year and stringent emission controls. This is why the highway truck application is not attractive for coal fuel.

And the engine application must not be so rare that there are not enough engines operating in a given region to justify a coal-cleaning plant (here the threshold level is about 500,000 hp, or 250 engines in the 2,000 hp class). Preferably, those regions with large concentrations of coal-fueled engines will be in close proximity to "engine-grade" source coals (that is, coals which can be economically cleaned to about 2% ash or better).

Also, the future potential displacement of oil by coal must be sufficient to justify the DDC coal engine development, and this implies that the total U.S. installed horsepower must be significant. For example, an engine application with an installed base of 4 million hp in the U.S. will account for 0.1 quad/year (10^{14} Btu/year) of fuel usage, assuming 3,000 full-load hours and 8,000 Btu/hp-hr specific energy consumption rate. This is about 3% of total U.S. oil consumption (6% of oil imports) which is quite significant.

Another factor favoring large engines is the cost of special hardware to burn coal fuels. This may include the injection system, costly wear-resistant parts for the injector tips, the piston rings and liners, the turbocharger, and the valves. This special hardware is expected to cost \$30,000 to \$100,000 per engine (dependent on engine size), which is tolerable for 1,500-2,000 hp engines but exceeds the engine cost itself for engines of less than 500 hp.

Past development work on coal-fueled diesel engines has shown that these engines favor steady operation with minimum load and speed changes. Cold starts should also be kept to a minimum. Therefore, applications are favored which offer a large amount of engine operating time at steady load and speed, such as the marine propulsion application.

4.2.3 Screening of Engine Applications

Table 4.2-1 lists the evaluation criteria discussed above, and shows how four candidate diesel engine applications were evaluated. A simple numerical rating system was used, and the outcome was so definitive that it was unnecessary to

develop a more sophisticated evaluation method. The marine and mine-haul applications received scores of +14 and +12, respectively (far above the other two candidates). The generator set industrial engines and bus/truck applications did not score well in economic feasibility, largely because of the relatively small engine size, low fuel usage per engine and stringent emission control requirements. In many U.S. locations, there is pressure to replace industrial drives with electric motors because of emission regulations. The bus/truck application, although attractive for its large national fuel usage, also suffers from a non-steady duty cycle and the need to have a clean coal fuel infrastructure throughout the highway system.

As a result of this screening, the inland waterway tow boat and mine haul truck engine applications were selected as attractive for adaptation to coal slurry fuel. These applications are examined in more detail in the following sections.

4.3 MINE HAUL TRUCK APPLICATION

4.3.1 Background

Mine haul trucks in the 170-240 ton class are the largest trucks produced and are about 20 ft. tall, 40-50 ft. long, with tires standing over 10 ft. tall. Empty, the vehicle weighs up to 350,000 lbs. Figure 4.3-1 shows typical mine haul trucks in use for open-pit mines to transport overburden and coal (or ore) out of the mine pit. The trucks cost about \$1.5 million, with the engine itself accounting for about 15% of the total cost.

Both electric drive models (diesel engines powering an alternator) and mechanical drive models (electronic transmission) are currently offered with key truck manufacturers including Marathon LeTourneau, WISED, VME-Americas, Unit Rig, and Dresser-Komatsu (Haulpak). The mine haul truck is suited for the coal fuel operation because 1,500-2,200 hp engines are used for 16 to 24 hours per day with a high load duty cycle. DDC Series 149 engines enjoy a significant market share in the U.S. (approximately 50%), and also are exported to mines in South America, Australia, South Africa, China, and India. This opens up the possibility of export sales of the U.S. clean coal technology and coal fueled diesel engines.

The mine haul truck market in the U.S. is in the range of 600 to 800 trucks per year, with 40% going to coal surface mines in ND, TX, WY, KY, and WV; 25% going to iron ore mines mainly in Minnesota; and 35% going to copper mines in AZ, NM, NV, and UT. Currently, there are an estimated 6,000-8,000 trucks in service with an average engine life of about 50,000 operating hours (typical useful engine life is estimated to be 10 years at 5,000 hours per year).

The larger mines operate a fleet of 15 to 30 active large trucks for 24 hours per day, with a diesel fuel consumption of 2 to 5 million gallons per year. Fuel is typically 15-20% of the operating cost per ton for a 100,000 ton/day mine. The large fuel oil budget of each major mine operator means that penetration of this market segment by coal engines can be accomplished readily, provided the cost savings with the use of coal-fueled engines are demonstrated.

The overall fuel usage in the U.S. by this engine application can be estimated as follows. If 2,000 active trucks are assumed to use 25 gallons per hour of diesel fuel for 4,000 hours per year, the total estimate is 200 million gallons per year (0.03 quad). Alternatively, if the total annual tonnage in surface mines (coal, ore, and overburden) is 1,000 million tons, and an average fuel use of 0.15 gallons per ton is assumed, one obtains 150 million gallons per year (0.02 quad). These estimates are reasonably consistent and give the total potential national impact of coal fuel substitution for the mine haul truck market.

4.3.2 Mine Locations and Coal Slurry Logistics

Three major types of surface mines use the mine haul trucks, as described below:

- Coal mines (Wyoming, North Dakota, Montana, and Texas),
- Iron ore mines (Minnesota), and
- Copper mines (Arizona, Utah, Nevada, and New Mexico).

For the coal mines in the Powder River Basin (Wyoming, North Dakota and Montana), it is clear that the preferred source coal for a clean coal slurry powered truck fleet would be in that same region. It has been demonstrated that several coals in this region are indeed cleanable to engine grade quality.

The AMAX/ADL coal cleaning plant model suggests a typical clean coal slurry price produced near the mine site in the Powder River Basin would be \$2.92/MMBtu in 1991 dollars.

For the copper mines in Arizona and neighboring states, the western source coals are also most convenient, but there would be a transportation cost to the slurry plant, because this plant must be located 20-50 miles from the mine haul truck fleets. For example, assuming a Salt Lake City slurry plant location, the AMAX/ADL coal cleaning plant model suggests a slurry price of \$3.55/MMBtu, including \$0.55/MMBtu for transportation.

4.3.3 Schematic Design of Coal Engine System for Mine Haul Truck

The 149 coal engine system would incorporate the same package as for the 149 diesel engine with the exception of special coal tolerant injection system and in-cylinder components. Also, this will include two new modules:

- Coal slurry fuel tank (about 1,200 gallons capacity), and
- Emission control system.

The coal slurry fuel tank will replace one of the two 300-600 gallon diesel fuel tanks currently found on the truck underbody. The emission control system is expected to be located beside the cab under the overhang, so as to minimize interference to the vision of the operator. Figure 4.3-2 illustrates the schematic design of the emission control system components.

4.4 INLAND WATERWAY TOWBOAT APPLICATION

Inland waterway transport is a major U.S. mode of freight traffic consisting mainly of barges pushed or pulled by diesel powered tugs or towboats. Each boat typically moves 10 to 16 barges with the engine operating close to full load. Figure 4.4-1 shows a typical inland waterway towboat in operation. These engines operate on a continuous basis for about 5,000-6,000 hours per year and are rated more conservatively than the mine haul truck engines. This is potentially an excellent application for coal fuel because of steady operation with a high load factor.

4.4.1 Existing Towboat Engines - Population and Fuel Usage

Based on data provided by Waterways Journal, Arthur D. Little, Inc. compiled an engine database of 3,700 work boats active on inland waterways in the U.S. The database shows vessel serial number, engine manufacturer and model, vessel horsepower, and year built. The results are summarized in Table 4.4-1. The retrofit potential is noted in the last column, largely based on engine size. It is clear that of the 4.5 million horsepower presently installed, 50-75% of towboat engines have the potential to be replaced by the 149 coal-fueled engines, when developed.

Combining the duty cycle data with the operator fuel use data, we estimated that active towboat engines operate an average of 3,200 (+/- 300) hours per year. This is about a 40% overall load factor on an annual basis. The average specific fuel consumption over the duty cycle was assumed to be 0.05 - gallon/hp-hr in agreement with the operating data. The total U.S. fuel usage for this engine application was estimated at 765 million gallons per year (0.11 quad). This is about three times the estimated fuel usage by the mine haul sector. DOE/EIA statistics (Petroleum Marketing Annual, 1988) show distillate fuel oil vessel bunkering averaged 765 million gallons for 1987 and 1988, distributed as follows:

<u>State</u>	<u>Million gallons</u>
Kentucky	85
Missouri	44
Tennessee	91
Louisiana	460
Mississippi	85

This excludes Florida, Alabama, and Texas, which together account for an additional 500 million gallons per year, much of which is used for coastal and Gulf of Mexico activities.

4.4.2 Towboat Operating Routes and Coal Slurry Logistics

The majority of inland waterway activity is along the Mississippi River and its

tributaries, as noted by the fuel usage statistics cited above. Towboats typically carry 20,000 gallons of fuel oil which can allow them to operate up to four weeks without refueling.

Source coals which are readily cleanable are available from mines which are in close proximity to the Mississippi river waterway system (e.g., Kentucky, West Virginia, Virginia, Ohio, Missouri, Pennsylvania, Illinois). The slurry plant serving a towboat fleet can be most logically placed at an existing bunkering terminal, with coal transported at low cost by barge to the slurry plant. This system of coal fuel logistics lends to a relatively low cost engine-grade slurry. For example, if Pittsburgh is selected as a slurry plant site and a 1-2% ash specification is assumed, the coal fuel price would be \$3.00/MMBtu, according to the AMAX/ADL coal cleaning model.

4.5 EMISSION CONTROL SYSTEM FOR MINE HAUL TRUCK AND TOWBOAT ENGINES

Emission control options were considered and preliminary design sketches prepared as a basis for the economic analysis. The design criteria assumed are presented in Table 4.5-1.

For each of the two high-speed engine applications, adequate emission control can most likely be achieved through: in-cylinder NO_x control; coal cleaning and post-combustion SO₂ removal; and a two-component particle removal scheme, such as a cyclone followed by a fabric filter. Since the flyash from coal-fueled diesels operating on highly beneficiated coals contain a large percentage of carbon (e.g., 50-70%), the feasibility of periodic incineration of captured particulate matter must be considered.

Task 2 Topical report of this program, being submitted to the DOE/METC, contains the itemized equipment list for the hypothetical coal engine emission system which meets the above specifications. Two configurations are illustrated for comparison; the differences are not great. Estimated costs are given for a 2,000 hp engine (mine haul truck) and a pair of 1,000 hp engines feeding a similar system (towboat). The emission system is estimated to cost \$93,500 to \$111,600 without installation, depending on the application.

4.6 ENGINE SPREADSHEET MODEL FOR LIFECYCLE COSTS

A Lotus 1-2-3 spreadsheet model was developed to analyze the total lifecycle cost of using the 149 coal-fueled engine under various assumptions. The spreadsheet model includes all aspects affecting the cost of owning and operating an engine, including

- (a) source coal costs including transportation
- (b) capital and operating costs of the slurry plant by coal type and its location
- (c) engine capital costs including installation
- (d) maintenance costs including parts replacement intervals
- (e) emission control costs (capital and operating)

Both coal-fired and oil-fired engines are analyzed in the cost calculation tables. The model takes the user specified input parameters and constants and estimates (1) the engine and emissions capital costs, (2) the engine and emissions operating costs, and (3) the maintenance costs in dollars per year (\$/year) for the oil and coal engines. These costs are totalled and the difference between these values determines the cost savings realized by using the coal engine for a given application and a given scenario. The details of the spreadsheet model are included in the topical report submitted to the DOE/METC.

4.6.1 Engine Efficiency and Fuel Cost

In each of the model runs, the baseline case for the oil-fired engine assumed \$.60/gallon oil costs, which represents typical costs of diesel fuel in 1991. Scenario 1 for the oil-fired engine assumed \$1.10/gallon oil costs, which might be anticipated in the next 5 to 15 years when oil prices rise, and when the coal engine becomes available. The oil-fired engine efficiency over the duty cycle was arbitrarily set at 34%. A typical average cycle efficiency of the 149 engine over a duty cycle for the mine haul truck or towboat may range from 32% to 37% (based on HHV), depending on the amount of part load and idle operation. Note that engine manufacturers conventionally quote the engine efficiency based on lower heating value (LHV). The 149 engine has a peak efficiency of 40% (LHV), and the operating efficiency over the duty cycle would range from 34 to 38% (LHV).

The Coal Slurry Cost Summary section of the spreadsheet model was used to calculate the itemized costs of the coal, slurry plant operations and slurry plant capital costs. Table 4.6-1 presents an example of the slurry costs which are presented in dollars per million BTU higher heating value (\$/MMBTU). The delivered coal price is made up of the cost of the coal at the minemouth and the transportation costs to the slurry plant. The operations and capital costs of the slurry plant (excluding the coal costs) were determined from the data entered in the Slurry Plant Operating and Capital Cost section of the model.

Table 4.6-1 Slurry Cost Summary

<u>Description</u>	<u>\$/MMBTU (HHV)</u>
Delivered Coal Price	1.79
Total Opns Cost (excluding coal)	1.21
Capital Recovery	0.46
Total Slurry Cost	3.47

4.6.2 Engine Capital and Operating Costs

The Engine Capital Cost section of the spreadsheet model uses financial statistics, operating statistics, and the capital cost of the components of the engine including the emissions equipment costs and determines an annual capital charge in dollars per year (\$/year). The annual capital charge is converted to a cost per output (\$/kWh) and then apportioned as engine costs and emissions costs in \$/kWh. Tables 4.6-2 and 4.6-3 display the capital cost information for the coal and diesel engines, respectively.

The Engine Maintenance Cost section in the spreadsheet model uses the maintenance intervals of major components of the engine in hours/cycle and cost per maintenance service for each of the engine components, and the life of the engine to determine the number of maintenance cycles (parts replacement) for each component and the life cost. The total maintenance cost in dollars per engine life is converted into a cost in dollars per kWh. Tables 4.6-4 and 4.6-5 present an example of the maintenance costs for the coal and diesel engines, respectively.

The total capital and operating costs were calculated for the coal and diesel engines to determine the cost benefit for using coal-fired engines in place of oil-fired engines. Tables 4.6-6 and 4.6-7 display an example of the engine capital and operating costs on \$/kWh and \$/year basis for the coal and diesel engines, respectively.

4.7 ANALYSIS OF ECONOMIC FEASIBILITY

The spreadsheet model, summarized in Section 4.6, was used to perform an analysis of the economic feasibility of operating a coal-fueled engine in the two selected applications. More than 50 runs of the model were made, resulting in several plots comparing the cost savings per year generated by using a coal engine in place of an oil-fired engine for different assumed scenarios. In each plot, fuel oil price was the basic independent parameter used on the horizontal axis.

In order to evaluate the two selected applications, representative locations were assumed. Salt Lake was assumed to be a representative Western site for the mine haul truck fleet (Copper mines and surface coal mines are in Arizona, New Mexico, Utah, and Powder River Basin). Pittsburgh was chosen as a typical inland waterway tow boat refueling location - it is one of many cities on the inland waterway. The proximity of these application sites to coal sources was taken into account. The AMAX slurry plant model was run to determine slurry plant and transportation costs. Colowyo coal was used for the Salt Lake site for mine haul trucks, due to its proximity. Blue Gem coal from Kentucky was selected in the model to determine the slurry plant and transportation costs for the Pittsburgh site for refueling tow boats.

The effect of several parameters was studied to determine their effect on the cost savings resulting from using coal engines instead of oil-fired engines for these applications. The parameters modified are as follows:

- o Nozzle life - how maintenance costs of critical parts affect savings
- o Coal cleaning costs - how slurry fuel costs affect savings
- o Interest rates - how the cost of money affects savings
- o Diesel fuel use - how the use of diesel fuel for idling affects savings
- o Engine efficiency - how a drop in engine efficiency would affect savings

The results of this work for the mine haul truck and inland waterway engines is summarized below. The detailed results of this task are described in a topical report submitted to the DOE/METC.

4.7.1 Mine Haul Truck Application

The model assumed a single 2,000 hp (1.5 MW) engine for a mine-haul truck. Parametric studies were conducted at three load factors, diesel fuel cost from \$0.60 to \$1.10 per gallon, two coal cleaning costs, two CWS injector nozzle lifetimes, and two financing rates. Figures 4.7-1 to 4.7-3 present results of this study. As one would expect, the load factor and coal cleaning cost have a significant effect on the annual savings. Assuming a coal cleaning cost of \$1.21/MMBtu and an average engine efficiency of 34% over the vehicle operating cycle, a mine-haul truck application in Salt Lake City breaks even at diesel fuel price of \$1.00 per gallon at a load factor of 25% (see Figure 4.7-1). As the load factor increases, the break even cost of diesel fuel is reduced to \$0.95 per gallon. As expected, a reduction in coal cleaning costs increases the annual savings.

The CWS injector nozzle life was varied in the model to give an indication of the relative effect of maintenance costs of a high wear component of the coal-fueled engine. Figure 4.7-2 summarizes the yearly savings for the mine haul truck application for two nozzle life values of 500 hours and 1,000 hours. It was seen that a doubling of the nozzle life has a smaller cost savings than reducing the coal cleaning cost in half. For a load factor of 25%, the break even diesel fuel cost is reduced from \$.98/gallon to \$.95/gallon by doubling the nozzle life.

The financing rate was varied to study the effect of the cost of money on the annual cost savings generated by using a coal engine in place of an oil-fired engine for a mine haul truck application. For a load factor of 25% and a coal engine efficiency of 34%, the diesel fuel cost break even point is \$.95/gallon for an interest rate of 12% as compared to \$1.06/gal for an interest rate of 16%. Figure 4.7-3 presents the annual cost savings for the mine haul truck fleet in Salt Lake City for interest rates varying from 12% to 16%.

For the above parametric runs, the engine model assumed that the mine haul truck coal engine was fueled with 100% coal at all operating conditions. Since the mine haul truck engine operates at idle speed and load during the downhill and loading periods, it may be necessary to use diesel fuel in addition to the coal fuel at these low loads (Task 4 of this program will also examine the use of ignition-improved methanol for this purpose). In order to account for this possibility, the spreadsheet model was run to calculate the effect of using 5%, 10% and 20% diesel fuel. These runs were made assuming the baseline values for all other parameters and the location of the mine haul truck to be in Salt Lake City. Figure 4.7-4 summarizes the effect of diesel fuel use on the annual savings for coal engine efficiency of 34% and a load factor of 25% for the mine haul truck. The coal engine cost savings break even point ranges from \$1.03/gallon at 5% diesel fuel use to \$1.09/gallon at 20% diesel fuel use.

4.7.2 Inland Waterway Tow Boat Application

The model for a tow boat with two 1,000 hp engines shows more annual cost savings than the mine-haul truck application, largely because of higher load factors. Pittsburgh was chosen as the refueling site for the coal-fueled tow boat. Figure 4.7-5 shows the effect of load factor and coal cleaning costs for this application. At \$1.21/MMBtu coal cleaning cost and 34% engine efficiency, the break even diesel fuel cost was \$0.82/gallon at a load factor of 40%. As the load factor increases from 40% to 50%, the diesel fuel cost break even point reduces to \$.79/gallon. This is quite significant because diesel fuel prices increased to this level in recent years on an intermittent basis. A reduction in the coal cleaning cost from \$1.21/MMBtu to \$.61/MMBtu at a load factor of 40% reduces the diesel fuel cost break even point from \$.82/gallon to \$.74/gallon for a coal engine efficiency of 34%.

Figure 4.7-6 presents the annual savings caused by doubling the injector nozzle life from 500 to 1,000 hours for the coal engine application. It can be seen that a doubling of the nozzle life has a smaller cost savings than reducing the coal cleaning in half. For the current nozzle life estimate of 500 hours and a coal engine efficiency of 34%, the break even diesel fuel cost ranges from \$.80/gallon for a load factor of 50% to \$.86/gal for a load factor of 30%.

Figure 4.7-7 shows the effect of varying the financing rate from 12% to 16% on the annual cost savings generated by using a coal engine in place of an oil-fired engine. For a load factor of 50%, the diesel fuel cost break even point is seen to increase from \$.89/gallon at 12% interest rate to \$.93/gallon at 16% interest rate.

Figure 4.7-8 summarizes the effect of diesel fuel use on the annual savings for coal engine efficiency of 34% and a load factor of 40% for the inland tow boat application. The coal engine cost savings break even point ranges from \$.84/gallon at 5% diesel fuel use to \$.89/gallon at 20% diesel fuel use.

5.0 ENGINE MODELING

In order to identify problems with the CWS ignition and combustion at high engine speeds and assess the effectiveness of possible design solutions, a coal combustion model developed at Ricardo North America was used to simulate the 149 diesel engine (Wahiduzzaman et al., 1990). The model was calibrated for the 149 two-stroke engine by performing a simulation of the diesel-fueled engine at the 1900 rpm, 2000 bhp operating condition for which engine test results and cylinder pressure data was available. A comparison of the heat release data obtained from the engine test and the model (see Figure 5-1) indicated reasonable agreement. The predicted brake power was within 3% of the measured value.

The coal combustion model combines the unique heat and mass transport and chemical kinetic processes of CWS combustion with the normal in-cylinder processes of a diesel engine. It includes a two-stage evaporation submodel for drying of the CWS droplet, a global kinetic submodel describing devolatilization, and a char combustion submodel describing surface gasification by oxygen, carbon dioxide and water vapor. The combustion volume is divided into multiple zones, whose thermochemistry is determined by in-situ equilibrium calculations. This provides accurate boundary conditions for the CWS droplet combustion submodels and gas phase heat release. A CWS fuel jet development, entrainment and mixing submodel is used to calculate the mass of entrained air in each of these zones. For diesel pilot assist, a separate combustion model was incorporated such that pilot fuel combustion can influence the CWS ignition and combustion processes.

One of the challenges of fueling a high-speed diesel engine with CWS is the time available for ignition and combustion. Since the CWS fuel is inherently difficult to ignite, attention was initially focussed on different ignition concepts. Among the ignition aids and options evaluated were: autoignition, glow plug assist, diesel pilot assist, increased in-cylinder temperature through exhaust residuals, preheating of CWS fuel, increased compression ratio and smaller droplet sizes.

A set of operating points for a 16V-149 diesel fueled engine for a typical mine-haul truck were simulated which spanned the range from 100% to 25% load at the rated speed of 1900 rpm and an idling load at 700 rpm speed. For simulations performed for autoignition studies, the predicted coal burnout was unsatisfactory with large droplets and at lower loads. In order to improve the burnout fraction, simulations were performed with 5 to 10% of the total fuel energy provided by diesel pilot fuel. This was seen to improve CWS ignition and combustion significantly and higher burnout rates were predicted.

Parametric studies were then carried out to determine whether burnout rates could be improved by modifying compression ratio, droplet size, injection timing, swirl, cylinder surface insulation, injector orifice spray angle and size and fuel temperature. The results of these studies with autoignition are discussed below.

5.1 EFFECT OF COMPRESSION RATIO

The simulation results show that autoignition of CWS can be achieved at 1900 rpm if the average droplet size is less than 20 microns. However, the model predicted a poor burnout for CWS at low compression ratios despite an elevated fuel temperature. Figure 5.1-1 shows the effect of compression ratio on the coal burnout at 1900 rpm, full load operating point with a Sauter mean diameter (SMD) of 16-18 microns. These results indicate that a compression ratio of 19 is necessary to achieve satisfactory burnout. At compression ratios higher than 19, excessive peak cylinder pressures were predicted. The simulation also predicted that a compression ratio higher than 20 may be necessary to autoignite CWS fuel at lower loads. However, a high compression ratio for the 149 engine may necessitate derating of the full load power.

5.2 EFFECT OF INJECTION TIMING AND DROPLET SIZE

The simulation predicted higher burnout rates for smaller droplets (<20 microns) and advanced injection timings of 30-35 deg before top dead center. Figures 5.2-1 and 5.2-2 present the effect of droplet size and injection timing at 1900 rpm, 100% and 50% load, respectively. An average droplet size in the range of 16 to 18 microns (small droplets) and 27 to 31 microns (large droplets) were used for these cases. In addition, a higher compression ratio of 23 was used for the large droplets compared to 19 for the small droplets. For the large droplet size, the burnout was predicted to be extremely poor and net positive work could not be achieved for any of the operating cases. On the other hand, for the simulations using small CWS droplets, with the exception of the idling case, autoignition is predicted and a significant amount of CWS is consumed and, consequently, net positive work is obtained. It should be noted here that simulation results are obtained using fixed air box and exhaust conditions. However, the exhaust condition varied significantly for the different CWS injection timings. For example, for full load and small droplet size the average exhaust gas temperatures are 653 K, 672 K, 739 K and 732 K for injection timings of 23, 25, 35, 40 deg BTDC, respectively. Incidentally, these temperatures are lower than the typical values (850 K) for a diesel fueled engine at the same load. Hence, it is unlikely that the existing air system will deliver the same intake boost when the engine is fueled with CWS, so a revised turbocharger match will be required.

5.3 UNIFORM VS. MULTIPLE DROPLET SIZES

In all the simulations reported previously, uniform droplet sizes were assumed within a zone. It was envisioned that if smaller particles exist in a zone, the ignitability of the rest of the CWS fuel in that zone will be significantly improved. In order to address this issue, it was decided to explore the effect of droplet size distribution. However, the model currently does not accommodate multiple droplet sizes within a zone. An alternative approach involved the use of a large number of zones and varying the droplet size between contiguous zones. It was decided that for each simulation three sizes of droplets (small, medium and large) would be chosen and any particular zone will receive droplets of a single size only. The frequency of injection of any one

of the droplet sizes will depend on the fraction of fuel mass associated with that droplet size. The SMD of the multiple droplet size distribution was set to be equal to the diameter of the medium sized droplet.

Figure 5.3-1 presents a comparison of the heat release rates for the uniform and multiple droplet sizes with SMD of 22.1 micron. The multiple size distribution had 9.8% of 17.5 micron small droplets, 80.1% of 22.1 micron medium droplets, and 10% of 30 micron large droplets. The heat release rates indicate earlier ignition and higher peak rate for the multiple droplets with 76.6% burnout as compared to 73.8% for the uniform droplets. Although detailed modeling is required to accurately predict the effect of droplet size distribution, the simulation results indicate that small droplets play a smaller role than seemed likely. The reason for this is that the in-cylinder temperature prior to combustion is highly influenced by the evaporation load of the CWS fuel and even if small droplets ignite early, much of this fuel energy is utilized in dehydration of larger droplets.

5.4 EFFECT OF IN-CYLINDER INSULATION

In an effort to assess the effect of in-cylinder insulation, simulations were performed with various levels of insulation. For this purpose, the piston top and fire deck surfaces, valve faces and the top 2/3 of the liner surface were assumed to be coated with an insulating material having a thermal conductivity of 0.8 W/m-K. An operating point corresponding to 50% load at 1900 rpm speed was simulated for coating thickness ranging from 0.5 to 2.5 mm. These results indicated that the burnout and brake power would be improved with increased level of insulation. The average in-cylinder wall temperature increased from 580 K to 743 K with an increase in the insulation thickness from 0.5 to 2.5 mm. This produced favorable in-cylinder conditions for a more complete and earlier combustion of the CWS fuel. This resulted in an increase in the brake thermal efficiency for the insulated 149 CWS-fueled engine. Incidentally, the commercial 149 diesel engine for high power levels already has insulated piston and firedeck surfaces.

5.5 EFFECT OF SWIRL LEVEL

The effect of intake swirl on CWS combustion was also examined. At an operating condition of 100% load at 1900 rpm speed with swirl numbers ranging from

1.07 to 2.5, the results predicted that an increase in the swirl decreased the burnout and consequently reduced the power output (see Figure 5.5-1). The increase in heat transfer rate due to higher swirl level was predicted to be marginal and, evidently, was not the reason for combustion degradation. Upon closer examination, it was observed that the increased entrainment associated with swirl tends to quench the combustion for the CWS fuel injected during the later part of the injection period. This is a rather surprising result. However, this effect is expected to be less pronounced at higher loads.

5.6 EFFECT OF INJECTOR ORIFICE SPRAY ANGLE AND SIZE

Engine modeling was performed to determine the effect of injector parameters on the engine performance. The plume/wall interaction model allows unimpeded growth of the fuel plume until it reaches a bounding wall. The entrainment rate of fresh air into the portion of the plume affected by the wall is based on an empirical scheme. Under this scheme, the entrainment into the affected zones is reduced by a factor proportional to the extent of interaction (degree of wetting). However, the entrainment rate of an affected zone is never allowed to fall below 50% of the rate prescribed by the unmodified model, i.e., the entrainment modification factor is bounded by the values between 0.5 and 1.0. In addition, the model also accounts for the plume/wall heat transfer and its impact on structural and burned zone temperatures. It should be emphasized that the above model is a simple one and mechanisms such as interaction between plumes, self-entrainment (entrainment of a portion of a plume by another portion of the same plume), loss of fuel past the rings and reentrainment of fuel captured by the cylinder walls have not been modeled. It is expected that a change of injection parameters will significantly affect these mechanisms and alter the engine performance, thus underscoring the need for a more accurate model.

In order to assess the effect of injector orifice spray angle and size, the fuel impingement model was also used. For 1900 rpm, 50% load operating point, the CWS-fueled engine performance improved with an increase in the spray angle from 130 to 160 degrees. For a smaller spray angle, impingement occurs earlier and the fuel plumes remain confined for a longer duration. This results in poor combustion and consequently, lower power output. At higher spray angle, the extent of plume/wall interaction is reduced and hence its effect is much

less prominent, as indicated for spray angles of 150 and 160 degrees. At yet higher spray angles, the effect of cylinder liner and plume interaction would be a dominant mechanism affecting the entrainment rate. It is believed that reentrainment of the fuel from the wall jets impacting the cylinder liner is comparatively less efficient due to the presence of crevices. However, this mechanism was not included in the present model and such simulations were not performed at spray angles higher than 160 degrees.

Simulation results of the effect of injector nozzle orifice size on CWS fuel combustion indicated that the engine performance and burnout improves considerably with a reduction in the orifice size. Figure 5.6-1 shows that a decrease in the orifice size results in reduced ignition delay and increased burnout. For orifice sizes of 0.3 mm or less, the CWS burnout fraction and power output tend to approach a maximum value. This is due to opposing trends caused by higher injection velocity, which on one hand promotes entrainment and dehydration processes, and on the other hand can cause early plume/wall interaction, which adversely affects the entrainment process. However, practical problems such as orifice clogging and excessive wear may preclude the use of smaller orifice sizes.

5.7 EFFECT OF CWS FUEL TEMPERATURE

The simulation also assessed the effect of injected CWS fuel temperature on the burnout rates and ignition delay periods. It was observed that CWS burnout was improved by an increase in the fuel temperature from 310 K to 350 K. This gain was achieved because of reduction of the heatup load and time for the injected CWS fuel, which constitutes a significant fraction of the physical ignition delay.

6.0 DESIGN OF INJECTION AND COMBUSTION SYSTEM

6.1 SELECTION OF THE CONCEPT

The conceptual design of the combustion and injection systems for the 149 coal-fueled high-speed engine was performed. As seen in Table 6.1-1, the following three concepts were considered:

- 1) High pressure injection (8,000 to 12,000 psi) of CWS to achieve autoignition at high loads and the use of ignition assistance at low loads (see Figure 6.1-1).
- 2) Medium pressure injection (3,000 to 4,000 psi) of CWS with impingement on the piston and glow plug or diesel pilot ignition (see Figure 6.1-2).
- 3) Low pressure admission (less than 500 psi, air or steam assist) of CWS after closing of exhaust valves and intake ports with diesel pilot ignition (see Figure 6.1-3).

Based upon the simulation results that predicted high coal burnout rates at 1900 rpm speed with autoignition of coal fuel, it was decided to pursue the first concept by injecting CWS into the combustion chamber near top dead center using a high pressure injector. Autoignition of coal fuel will be achieved at high engine loads by high in-cylinder and surface temperatures in the engine. These temperatures will be achieved by a combination of hot residual gas in the cylinder, compression ratio of 19, insulated combustion chamber components and intake air temperatures to the cylinder of about 200°F. At lower loads, an ignition assist such as ignition-improved methanol or diesel pilot fuel will be used, if necessary, to achieve coal ignition and combustion. The CWS injection system performance is expected to play a major role in the successful outcome of this program. Simulation results indicate that the CWS injection system is required to deliver relatively small droplets because of the limited time available at the high engine speeds.

6.2 CWS INJECTION SYSTEM

Injection system design has, and continues to be, a key issue in the development of CWS-fueled diesel engines. The problems that were initially encountered by Ryan et al. (1980, 1982, 1984) consisted primarily of reliability factors associated with sticking of moving parts, packing of coal particles in tight clearances, and plugging of flow passages. Most of these problems were associated with loss of water from the slurry as it passed through the injection system components and came into contact with tight clearances.

The work of Rao et al. (1989) and Hsu (1991) have contributed significantly to the solution of many of the reliability problems listed above. The system design that appears to be evolving as the best approach to the reliability problems involves the use of a free shuttle piston to separate a clean pumping fluid from the slurry (Rao, 1989). In this approach, a standard plunger and barrel assembly is used to meter and pressurize a clean fluid such as diesel fuel. This high pressure fluid acts on one side of a free piston. The slurry is supplied through a check valve to the other side of the free piston where it is pressurized and directed through an injector nozzle. In the current designs, oil is supplied to both the guide area of the needle valve in the nozzle and to the center of the free piston. The oil is supplied at pressures exceeding the peak injection pressure so that it acts as both a lubricant and a sealing fluid.

The current design approach has greatly improved the reliability of the systems and has allowed researchers to develop fairly extensive engine experience with slurries. Sufficient experience has been gained to indicate the next set of problems related to system durability, in particular the durability of the spray holes in the injection nozzle tip. Excellent work by both Flynn et al. (1990) and Mayville et al. (1990) suggests that the spray hole durability problem can be solved by selection of the appropriate material.

6.2.1 Design of the 149 CWS Injection System

The design strategy for the CWS injection system for the 149 engine is to build upon the results of the previous work and to adapt the current technology to the unique requirements of the 149 engine. The goal is to ultimately develop an electronic unit injector which will fit into the cylinder head in place of the diesel injector. As an intermediate step, a pump-line-nozzle (PLN) injection system is being developed as a test bed for new concepts and as a flexible (injection timing control) tool for engine development. In addition, a mechanical unit injector (MUI) for CWS has been designed which will inject CWS directly (without a check valve) using an oil lubricated plunger.

The unique design considerations for the 149 engine include the fact that the engine runs at speeds from two to five times greater than the medium speed engines used by Hsu et al. (1989) and Rao et al. (1990), respectively. In addition, the system must be able to fill much faster than the other systems because it must inject from four to ten times as often as the other systems due to the fact that the 149 engine operates on a two rather than a four stroke cycle. Another constraint on the system was that the injector nozzle should fit in place of the existing diesel injector so that cylinder head modification would be minimized.

Simulation of the CWS injector was conducted to calculate injection parameters such as pressure, rate and duration, and the camshaft surface (Hertz) stresses for the MUI. The effect of nozzle tip flow area and the CWS properties (density, bulk modulus and viscosity) on the above parameters was investigated. The peak injection pressure was found to be in the range of 10,000 to 18,000 psi, depending upon the nozzle tip flow area. Figure 6.2-1 presents the injection parameters and cam hertz stress for a test run at 1900 rpm, full load condition. Based upon these simulation results, the number and size of spray holes was selected for the CWS injector and the following specifications were developed:

CWS Injection amount	- 465 cu mm/stroke at 1900 rpm, 100% load
CWS injection pressures	- 10,000 to 18,000 psi
Injection duration	- 30 to 35 degrees (maximum)
Beginning of injection	- 30 to 40 deg BTDC (maximum)

Figure 6.2-2 presents a schematic of the PLN injector for CWS. The external package dimensions of this injector fit within the current constraints of the EUI for the 149 diesel engine. The shuttle assembly, the slurry check valve, and the lubricating/seal oil supplies are all incorporated within the injector body. The slurry passages through the body to the tip have been sized to accommodate the slurry flow with acceptable pressure drop. The passages have been designed to present the simplest flow path for the slurry, eliminating dead end passages, annular flow passages, and flow area changes that are often found in diesel fuel injection systems. The injection pulse will be provided using an engine driven injection pump which has been shown to produce the desired injection characteristics at 2000 rpm.

The design of the PLN injector components has been completed. Efforts were made to incorporate as many of the standard injector parts as possible in the design. The design for the spray tip and spring cage was done by DTC, and these are compatible with the MUI injector for CWS. The PLN injector design work also included connecting the AMBAC check valve to the injector body, installing a high pressure relief valve in the diesel injection line, and deciding on the spray tip hole configuration.

Several different tips with different spray hole geometries (sizes and numbers) are being fabricated to provide a selection for optimizing the engine performance. The hole selection options are infinite, but the initial designs range from doubling the number of holes and keeping the diameter the same as standard to keeping the number the same and doubling the flow area of each hole. The goal is to examine the extremes in injection pressure and hole size. It was decided to fabricate 7-hole and 8-hole (in two rows) spray tips with different orifice diameters for the initial bench testing.

The fabrication of the PLN injector components is expected to be completed by the end of November. DTC is fabricating the spray tip, spring cage, shuttle plunger and barrel, while the remaining components are being fabricated at SwRI.

6.3 SET-UP TEST FACILITIES

The injection system is currently mounted on a test stand where it will undergo preliminary optimization prior to installation and test on the engine. The goals of the bench test are to achieve reliable operation of the system up to 2000 rpm and to refine the selection of the spray hole geometries (sizes, numbers and spray angle) to achieve desirable injection rate, duration and spray jet characteristics (such as penetration, cone angle, and droplet size distribution).

SwRI installed the AMBAC diesel injection pump and the seal oil supply pump on the test stand. The slurry supply pump is scheduled for delivery during the month of November. The CWS injector will initially be tested with water in place of the slurry. Procedures for slurry preparation, storage and pumping,

solids concentration measurement, and flushing and cleaning of the equipment will be established before starting bench tests of the CWS injector.

6.4 TEST ENGINE PREPARATION

An 8V-149TI engine, rated at 1000 bhp @ 1900 rpm, was built at DDC. This engine was equipped with the Detroit Diesel Electronic Control (DDEC) system and electronic unit injectors (EUI) for diesel fuel. Also, several engine components were installed which are normally used for high power output Series 149 engines and for use with high sulfur content diesel fuel. These components included: plasma spray ceramic-coated pistons (compression ratio of 15), cylinder heads and exhaust valves; thermal wrap exhaust manifolds and turbo-charger turbine housings; and wear-resistant ceramic-coated fire rings. The engine was tested in DDC's production engine test facilities prior to its shipment to SwRI. Figure 6.4-1 presents a photograph of this engine.

One cylinder of this engine will be operated on CWS fuel during Task 4 of this program. SwRI will initially use this engine to decide on the mounting and location of various components including the diesel and pilot injection pumps and CWS injector. Based on the preferred location of the pumps and other technical considerations, one cylinder of this engine will be chosen and modified for operation on CWS fuel.

7.0 CONCLUSIONS

Economic Analysis and Market Assessment

The economic feasibility analysis predicted that the future use of coal fuel in the mine-haul truck and tow boat applications is feasible and has the potential to offer significant annual savings of up to \$100,000 per year for each vehicle. With an improvement in the cost of CWS fuel and wear-resistant engine components required for coal fuel operation, these savings are expected to increase. However, the future diesel fuel price will be critical in determining the economic feasibility. It was predicted that the inland tow boat application has greater potential than the mine-haul truck because of higher load factor and lower CWS fuel costs for the tow boat engines.

Engine Modeling

Simulation results for the 149 coal-fueled engine indicate that an average droplet size smaller than 20 microns is important to obtain high coal burnout at engine speeds of 1900 rpm. The results predict autoignition of coal fuel at various loads at 1900 rpm with an average droplet size of 18 microns and compression ratio of 19. Based on these results, the performance of the CWS injection system is expected to be crucial in achieving acceptable coal burnout in the short time available in the 149 high-speed engine. At lower engine loads, ignition assist or higher compression ratio may be required.

Injection and Combustion System

A combustion and injection system for the 149 coal-fueled engine is being pursued which relies on injecting CWS into the combustion chamber near top dead center using a high pressure injector. Autoignition of coal fuel will be achieved by high in-cylinder and surface temperatures obtained by a combination of retained hot residual gas, ceramic insulated cylinder components and increased compression ratio. A pump-line-nozzle injection system is currently being developed as a flexible tool for engine development work. Ignition-improved methanol is being considered for the cold start, warm up and shutdown of the engine. A diesel pilot injector is a secondary option if satisfactory coal burnout can not be achieved with autoignition.

8.0 FUTURE WORK

The future work involves the fabrication of the CWS injection system for the 149 engine. Also, engine components required for the coal-fueled engine operation are being fabricated or procured. These components include: cylinder head incorporating a glow plug or diesel pilot injector and a pressure transducer to acquire cylinder pressure data, high compression ratio pistons, bypass blower controls and the pilot injector.

SwRI is setting up facilities to bench test the injection and combustion system at their facilities. After successful bench testing, one cylinder of an 8V-149TI engine will be operated on CWS fuel at SwRI.

REFERENCES

- Flynn, P.L., Hsu, B.D., and Leonard, G.L., 1990, "Coal Fueled Diesel Engine Progress at GE Transportation," ASME Publication ICE-Vol. 12, pp. 1-9.
- Hsu, B.D., and Confer, G.L., 1991, "Progress on the Investigation of Coal-Water-Slurry Combustion in a Medium Speed Diesel Engine: Part 4 - Fuel Effects," ASME Publication ICE-Vol. 14, pp. 1-8.
- Kakwani, R.M., Kamo, R., Cutlip, R.G., and Smith, W.C., 1989, "Combustion Characteristics of Dry Coal Powder-Fueled Adiabatic Diesel Engine," ASME Publication ICEVol. 7, pp. 27-33.
- Kakwani, R.M., Badgley, P.R., and Smith, W.C., 1990, "Development of Fuel Injection and Combustion Systems for CWS-Fueled Diesel Engines," ASME Publication ICE-Vol. 12, pp. 41-50.
- Mayville, R.A., Rao, A.K., and Wilson, R.P., 1990, "Cooper-Bessemer Coal-Fueled Engine Systems: Recent Developments in Durable Components," ASME Publication ICE-Vol.12, pp. 17-22.
- Rao, A.K., Wilson, R.P., Balles, E.N., Mayville, R.A., McMillian, M.H., and Kimberley, J.A., 1989, "Cooper-Bessemer Coal-Fueled Engine System - Progress Report" ASME Publication ICE-Vol. 7, pp. 9-17.
- Ryan, T.W., Likos, W.E., and Moses, C.A., 1980, "The Use of Hybrid Fuels in a Single-Cylinder Diesel Engine," SAE Paper 801380.
- Ryan, T.W., Callahan, T.J., Dodge, L.G., and Moses, C.A., 1982, "Injection, Atomization, and Combustion of Carbon Slurry Fuels," SAE Paper 821199.
- Ryan, T.W., and Dodge, L.G., 1984, "Diesel Engine Injection and Combustion of Slurries of Coal, Charcoal, and Coke in Diesel Fuel," SAE Paper 840119.
- Wahiduzzaman, S., Blumberg, P.N., Keribar, R., and Rackmil, C.I., 1990, "A Comprehensive Model for Pilot-Ignited Coal-Water Mixture Combustion in a Direct Injection Diesel Engine," Journal of Engineering for Gas Turbines and Power, Vol. 112, No. 3, pp. 384-390.
- Webb, H.A., Rekos, N.F., Kothari, V.P., and Bedick, R.C., 1990, "Proceedings of the Seventh Annual Coal-Fueled Heat Engines and Gas Stream Cleanup Systems Contractors Review Meeting," Morgantown, West Virginia.
- Wilson, R., 1991, "Market Assessment and Economic Feasibility of High-Speed Coal-Fueled Diesel Engines," Topical Report submitted to DOE/METC.

Table 4.2-1 Evaluation Criteria for DDC Coal Engine Application

		Relative Ability to Accommodate Clean Coal Slurry			
		Marine	Gen Set	Mine Haul	Bus/Truck
A. Economics					
• Annual fuel usage per engine		+2	+2	+2	-1
• Sufficient engine population for CWS plant		+1	0	+1	+2
• Total number of engines in use		+1	0	+1	+2
• Cost of emission control to satisfy (future) requirements		+1	-1	+1	-2
• Cost of special engine hardware to burn coal		+1	-1	+1	-2
• Proximity to low-cost source coals		+2	0	+1	-1
	Subtotal	8	0	7	-2
B. Technical Feasibility					
• Duty cycle - load changes and turnaround; number and frequency of cold starts		+2	+2	+1	-2
• Impact of unscheduled downtime		-2	-1	-2	-2
	Subtotal	0	+1	-1	-4
C. Logistics and System Design Constraints					
• Fuel processing facility and local bunkering facility		+1	+2	+2	-1
• Space available next to engine for emissions control equipment		+2	0	+1	-2
• Accommodation of other on-vehicle subsystems		+2	+1	+1	-1
• larger CWS fuel tank/separate DF2 loop for pilot/load fuel		+1	-1	+2	+1
• injection system changeout/frequent maintenance		+6	+2	+6	-3
	Subtotal	+14	+3	+12	-9
	Total Score				

Ratings

- +2 No problem; or superior fit with CWS
- +1 Probably acceptable, but needs engineering
- 0 Unknown; to be determined
- 1 Problem identified
- 2 Significant problem identified which may be insurmountable

Table 4.4-1 Inland Waterway Towboat Engine Database Results

Engine Model	Typical Vessel hp	Number of Vessels	Total Horsepower	Retrofit Potential for DDC 149 Coal Engine
DDC Model 92	600 or 1,200	151	162,000	Possible with 8V-149
DDC Model 149	1,350 or 1,800	190	287,000	Excellent
EMD Model 567	3,000 (avg)	160	418,000	Good
Miscell GM	2,000 (avg)	42	79,000	Good
DDC Model V-71	500-1,000	1340	851,000	Poor
6-110	Approx 400	75	27,000	Poor
EMD Model 645				
12 cyl.	2,800 or 4,300	124	444,000	Excellent
16 cyl.	3,400 to 9,000	244	1,210,00	Excellent
20 cyl.	7,000 to 10,500	32	253,000	Poor
8 cyl.	1,800 or 3,000	20	70,000	Good
Other manufac.				
Cummins, Caterpillar, Alco	varies: 500-4,000	<u>1,303</u>	<u>1,646,000</u>	Good for certain engines
Totals		3,690	4,454,000	

Table 4.5-1 Emission Control System Design Assumptions

Performance	
SO ₂	50-70% reduction (depending on fuel sulfur)
NO _x	20-40% reduction
Particulate matter	95-99.5% reduction (to meet California off-road standards)
Cost	< \$20/hp-yr
Size of Emission Control Module	< 8' x 10' x 10'
Weight	< 2 tons
Load range	Idle - 100%
Load transients	TBD
Vibration loads	TBD
Change in orientation	±9° from horizontal

Table 4.6-2 Coal-Fueled Engine Capital Cost

DESCRIPTION	UNITS	BASE CASE
FINANCIAL STATISTICS		
COST OF CAPITAL	%	12.00%
RECOVERY/ TERM	years	20
OPERATING STATISTICS		
POWER RATING	MEGAWATTS	1.5
ANALYSIS PERIOD	years	20
AVAILABILITY	%	90.00%
LOAD FACTOR	%	40.00%
FULL LOAD EQUIV. HOURS	HRS PER year	3154
POWER GENERATED	MWH	95,000
CAPITAL COST COMPONENTS		
LUBE SYSTEM		\$8,000
PISTONS & PISTON RINGS		\$21,300
CYLINDER LINERS		\$8,800
CYLINDER HEAD		\$28,800
VALVES		\$9,600
FUEL PUMP & INJECTOR		\$16,400
INSTRUMENTS/CONTROLS		\$15,000
TURBOCHARGER		\$15,300
CAM SHAFT & BEARINGS		\$7,600
MISCELLANY		\$34,650
SUBTOTAL CRITICAL PARTS		\$165,450
NON-CRITICAL PARTS		\$123,000
PRODUCTION ENGINE COST		\$288,450
INSTALLATION COSTS		\$144,225
INSTALLED ENGINE COST		\$432,675
EMISSIONS EQUIPMENT		\$130,000
TOTAL CAPITAL OUTLAY		\$562,675
ANNUAL CAPITAL CHARGE	\$/year	\$75,330
CAPITAL CHARGE/OUTPUT		
PRINCIPAL COMPONENT	\$/kWh	\$0.0059
COST OF CAPITAL	\$/kWh	\$0.0099
TOTAL CHARGE/OUTPUT	\$/kWh	\$0.0159
TOTAL CAPITAL COST/OUTPUT		
ENGINE	\$/kWh	\$0.0122
EMISSIONS	\$/kWh	\$0.0037

Table 4.6-3 Oil-Fired Diesel Engine Capital Cost

DESCRIPTION	UNITS	BASE CASE
FINANCIAL STATISTICS		
COST OF CAPITAL	%	12.00%
RECOVERY TERM	years	20
OPERATING STATISTICS		
POWER RATING	MEGAWATTS	1.5
LIFE	years	20
AVAILABILITY	%	90.00%
LOAD FACTOR	%	40.00%
POWER GENERATED	MW HOURS	95,000
CAPITAL COST COMPONENTS		
STANDARD ENGINE COST	\$	\$220,000
% COST OF CRITICAL PARTS	%	44.00%
VALUE OF CRITICAL PARTS	\$	\$96,800
ADJ FOR CRITICAL PARTS	%	100.00%
ADJ COST-CRITICAL PARTS	\$	\$96,800
PRODUCTION ENGINE COST	\$	\$220,000
INSTALLATION COSTS	\$	\$110,000
TOTAL CAPITAL OUTLAY	\$	\$330,000
ANNUAL CAPITAL CHARGE	\$/year	\$44,180
CAPITAL CHARGE/OUTPUT		
PRINCIPAL COMPONENT	\$/kWh	\$0.0035
COST OF CAPITAL	\$/kWh	\$0.0058
TOTAL CHARGE/OUTPUT	\$/kWh	\$0.0093

Table 4.6-4 Coal Engine Maintenance Cost

DESCRIPTION	UNITS	BASE CASE
MAINTENANCE INTERVALS		
OIL/FUEL/AIR FILTER	HRS/CYCLE	250
INJECTOR REPLACEMENT	HRS/CYCLE	500
TOP END OVERHAUL	HRS/CYCLE	8,000
MAJOR OVERHAUL	HRS/CYCLE	20,000
BLOCK OVERHAUL	HRS/CYCLE	40,000
COST PER SERVICING		
OIL/FUEL/AIR FILTER	\$	\$300
INJECTOR REPLACEMENT	\$	\$4,000
TOP END OVERHAUL	\$	\$20,000
MAJOR OVERHAUL	\$	\$40,000
BLOCK OVERHAUL	\$	\$80,000
SERVICING THRU 1st REBUILD		
OIL/FUEL/AIR FILTER	# OF CYCLES	160
INJECTOR REPLACEMENT	# OF CYCLES	80
TOP END OVERHAUL	# OF CYCLES	5
MAJOR OVERHAUL	# OF CYCLES	2
BLOCK OVERHAUL	# OF CYCLES	1
COST THRU 1st REBUILD		
OIL/FUEL/AIR FILTER	\$	\$48,000
INJECTOR REPLACEMENT	\$	\$320,000
TOP END OVERHAUL	\$	\$100,000
MAJOR OVERHAUL	\$	\$80,000
BLOCK OVERHAUL	\$	\$80,000
TOTAL THRU #1 BOTTOM END	\$	\$628,000
ALLOCATED MAINTENANCE %		
OIL/FUEL/AIR FILTER	%	7.64%
INJECTOR REPLACEMENT	%	50.96%
TOP END OVERHAUL	%	15.92%
MAJOR OVERHAUL	%	12.74%
BLOCK OVERHAUL	%	12.74%
LIFE SERVICE CYCLES		
OIL/FUEL/AIR FILTER	# OF CYCLES	252
INJECTOR REPLACEMENT	# OF CYCLES	126
TOP END OVERHAUL	# OF CYCLES	7
MAJOR OVERHAUL	# OF CYCLES	3
BLOCK OVERHAUL	# OF CYCLES	1
20 year COST		
OIL/FUEL/AIR FILTER	\$/LIFE	\$75,600
INJECTOR REPLACEMENT	\$/LIFE	\$504,000
TOP END OVERHAUL	\$/LIFE	\$140,000
MAJOR OVERHAUL	\$/LIFE	\$120,000
BLOCK OVERHAUL	\$/LIFE	\$80,000
TOTAL MAINTENANCE COST	\$/LIFE	\$919,600
	\$/kWh	\$0.0097

Table 4.6-5 Oil-Fired Engine Maintenance Cost

DESCRIPTION	UNITS	BASE CASE
MAINTENANCE INTERVALS		
OIL/FUEL/AIR FILTER	HRS/CYCLE	250
INJECTOR REPLACEMENT	HRS/CYCLE	5,000
TOP END OVERHAUL	HRS/CYCLE	11,000
MAJOR OVERHAUL	HRS/CYCLE	22,000
BLOCK OVERHAUL	HRS/CYCLE	60,000
COST PER SERVICING		
OIL/FILTER/AIR FILTER	\$	\$350
INJECTOR REPLACEMENT	\$	\$6,500
TOP END OVERHAUL	\$	\$23,000
MAJOR OVERHAUL	\$	\$45,000
BLOCK OVERHAUL	\$	\$70,000
SERVICE CYCLES THRU 1st REBUILD		
OIL/FILTER/AIR FILTER	# OF CYCLES	240
INJECTOR REPLACEMENT	# OF CYCLES	12
TOP END OVERHAUL	# OF CYCLES	6
MAJOR OVERHAUL	# OF CYCLES	3
BLOCK OVERHAUL	# OF CYCLES	1
COST THRU 1st BOTTOM END REBUILD		
OIL/FILTER/AIR FILTER	\$	\$84,000
INJECTOR REPLACEMENT	\$	\$78,000
TOP END OVERHAUL	\$	\$138,000
MAJOR OVERHAUL	\$	\$135,000
BLOCK OVERHAUL	\$	\$70,000
TOTAL	\$	\$505,000
ALLOCATED MAINTENANCE %		
OIL/FILTER/AIR FILTER	%	16.63%
INJECTOR REPLACEMENT	%	15.45%
TOP END OVERHAUL	%	27.33%
MAJOR OVERHAUL	%	26.73%
BLOCK OVERHAUL	%	13.86%
LIFE SERVICE CYCLES		
OIL/FILTER/AIR FILTER	# OF CYCLES	252
INJECTOR REPLACEMENT	# OF CYCLES	12
TOP END OVERHAUL	# OF CYCLES	5
MAJOR OVERHAUL	# OF CYCLES	2
BLOCK OVERHAUL	# OF CYCLES	1
20 year COST		
OIL/FILTER/AIR FILTER	\$/LIFE	\$88,200
INJECTOR REPLACEMENT	\$/LIFE	\$78,000
TOP END OVERHAUL	\$/LIFE	\$115,000
MAJOR OVERHAUL	\$/LIFE	\$90,000
BLOCK OVERHAUL	\$/LIFE	\$70,000
TOTAL MAINTENANCE COST	\$/LIFE	\$441,200
	\$/kWh	\$0.0046

TABLE 4.6-6 Coal Engine Capital and Operating Cost

DESCRIPTION	\$/kWh	\$/YEAR
OPERATING (EXCL EMISSIONS)		
FUEL	\$0.0374	\$177,858
MAINTENANCE	\$0.0097	\$45,980
TOTAL OPERATING	\$0.0471	\$223,838
CAPITAL (EXCL EMISSIONS)		
EMISSIONS		
OPERATING	\$0.0100	\$47,500
CAPITAL	\$0.0037	\$17,404
TOTAL EMISSIONS	\$0.0137	\$64,904
TOTAL COST	\$0.0730	\$346,668

TABLE 4.6-7 Oil-Fired Engine Capital and Operating Cost

DESCRIPTION	\$/kWh	\$/YEAR
ENGINE CAPITAL	\$0.0093	\$44,180
EMISSIONS CAPITAL	\$0.0010	\$4,686
ENGINE MAINTENANCE	\$0.0046	\$22,060
FUEL	\$0.0421	\$200,109
TOTAL COST	\$0.0571	\$271,035

Table 6.1-1

**DDC 149 COAL-FUELED ENGINE
INJECTION AND COMBUSTION CONCEPTS**

	<u>#1A</u>	<u>#1B</u>	<u>#2A</u>	<u>#2B</u>	<u>#3A</u>	<u>#3B</u>	<u>#3C</u>
CONCEPT	HPD	HPG	MPG	MPD	LPD	LPA	LPS
INJECTOR	CONV.	CONV.	CONV.	CONV.	CONV.	AIR-ASSIST	STEAM
BOI (DBTC)	25	25	35	35	135	135	135
INJ. PRESSURE	12,000	12,000	3000	3000	500	500	500
IGNITION	PILOT	AUTOIGNITION*	AUTOIGNITION*	PILOT	PILOT	PILOT	PILOT
OTHER PARTS	--	--	IMPINGING PISTON	IMPINGING PISTON	--	AIR COMPRESSOR	SLURRY HEATER
INJ. WEAR	HIGH	HIGH	MEDIUM	MEDIUM	LOW	LOW	LOW
PROBLEMS	COMPLEX	-----	COLD START -----	COMPLEX	KNOCK	POWER, KNOCK	STARTING, KNOCK

*GLOW PLUG FOR THE COLD START AND WARM UP.

COAL-FUELED HIGH-SPEED DIESEL ENGINE DEVELOPMENT

METC CONTRACT NUMBER DE-AC21-90MC27222 (Program schedule revised 9/91)

Figure 3-1 Program Schedule

COAL3 9/12/91

Figure 4.3-1 Typical Mine Haul Trucks in Operation

Figure 4.3-2 Emission Control Device for Coal-Fired Diesel Mine-Haul Truck (2000 HP)

Figure 4.4-1 1800 HP Viking Class Towboat

MINE HAUL TRUCK COAL ENG. COST SAVINGS COAL CLEANING COST/LOAD FACTOR EFFECTS

SITE: SALT LAKE
 DF2 ENGINE EFFICIENCY=34%
 COAL ENGINE EFFICIENCY=34%
 COAL CLEANING COSTS: \$1.21/MMBTU, \$1.61/MMBTU
 COAL COSTS= \$1.74/MMBTU

Figure 4.7-1

MINE HAUL TRUCK COAL ENG. COST SAVINGS NOZZLE LIFE & LOAD FACTOR EFFECTS

SITE: SALT LAKE
DF2 ENGINE EFFICIENCY=34%
ENGINE EFFICIENCY=34%
NOZZLE LIFE: 500, 1000 HOURS
COAL COSTS= \$1.74/MMBTU

Figure 4.7-2

MINE HAUL TRUCK COAL ENG. COST SAVINGS INTEREST RATE/LOAD FACTOR EFFECTS

..... LOAD FACTOR=15% — LOAD FACTOR=25% --- LOAD FACTOR=35%

SITE: SALT LAKE
DF2 ENGINE EFFICIENCY=34%
COAL ENGINE EFFICIENCY=34%
INTEREST RATE= 12%, 16%
COAL COSTS= \$1.74/MMBTU

Figure 4.7-3

Figure 4.7-4

Figure 4.7-5

Figure 4.7-6

Figure 4.7-7

TOW BOAT COAL ENGINE COST SAVINGS DIESEL FUEL USE EFFECTS

— 0% DIESEL FUEL USE 5% DIESEL FUEL USE --- 10% DIESEL FUEL USE ---- 20% DIESEL FUEL USE

SITE: PITTSBURGH
DF2 ENGINE EFFICIENCY=34%
COAL ENGINE EFFICIENCY=34%
LOAD FACTOR=40%
COAL COSTS=\$1.79/MMBTU

Figure 4.7-8

Figure 5-1 Comparison of Heat Release Rate from Predicted and Measured Cylinder Pressure Data.

EFFECT OF COMPRESSION RATIO AUTOIGNITION @ 1900 RPM, 100% LOAD

Figure 5.1-1

EFFECT OF CWS INJECTION TIMING & DROPLET SIZE AUTOIGNITION @ 1900 RPM, 100% LOAD

Figure 5.2-1

EFFECT OF CWS INJECTION TIMING & DROPLET SIZE AUTOIGNITION @ 1900 RPM, 50% LOAD

Figure 5.2-2

UNIFORM & MULTIPLE DROPLET SIZE DISTRIBUTION AUTOIGNITION @ 1900 RPM, 100% LOAD (SMD = 22.1 MICRONS, CR = 19:1)

Figure 5.3-1

EFFECT OF SWIRL LEVEL ON CWS BURNOUT AND POWER OUTPUT AUTOIGNITION @ 1900 RPM, 100% LOAD

Figure 5.5-1

EFFECT OF INJECTOR ORIFICE SIZE ON CWS BURNOUT AND IGNITION DELAY AUTOIGNITION @ 1900 RPM, 50% LOAD

Figure 5.6-1

SERIES 149 COAL-FUELED HIGH SPEED DIESEL ENGINE (Concept #1)

(CWS INJECTION PRESSURE $\approx 8000-12000$ PSI)

Figure 6.1-1

SERIES 149 COAL-FUELED HIGH SPEED DIESEL ENGINE (Concept #2)

(CWS INJECTION PRESSURE $\approx$ 2000-3000 PSI)

Figure 6.1-2

SERIES 149 COAL-FUELED HIGH SPEED DIESEL ENGINE (Concept #3)

(CWS PRESSURE $\approx$ 200-500 PSI)

Figure 6.1-3

149 CWS INJECTOR 12.7 MM PLUNGER, 2000 CFM TIP,
RPM=1900 FPC=464.96

Figure 6.2-1 The CWS Injector Simulation Results at 1900 RPM, 100% Load Point

Figure 6.2-2 Pump-Line-Nozzle Injector for CWS

Figure 6.4-1 Photograph of the 8V-149TI Engine

END

**DATE
FILMED**

7 / 29 / 92