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Coal-Fueled Diesel System for Stationary Power Applications -- Technology Development

Final Report
March 1988 - June 1994

October 1995

Work Performed Under Contract No.: DE-AC21-88MC25124

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

By
Arthur D. Little, Inc.
Cambridge, Massachusetts

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Morgantown, West Virginia 26507-0880

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October 1995

List of Abbreviations

A&E	Architect and Engineer
AFB	Atmospheric Fluidized Bed
AISI	American Iron and Steel Institute
ATC	After Top (Dead) Center
BACT	Best Available Control Technology
BHP	Brake Horse Power
BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
BTC	Before Top (Dead) Center
CAA	Clean Air Act
CAAA	Clean Air Act Amendments
CFBC	Circulating Fluidized Bed Combustion
CFD	Computational Fluid Dynamics
CQ	Coal Quality, Inc.
CQDC	Coal Quality Development Center
CWS	Coal Water Slurry
DDC	Detroit Diesel Corporation
DOE	Department of Energy
DSCF	Dry Standard Cubic Foot
EA	Energy Associates
ECS	Emissions Control System
EI	Energy International, Inc.
EIA	Energy Information Administration
EPA	Environmental Protection Agency
FGD	Flue Gas Desulfurization
GE	General Electric Corporation
GM/EMD	General Motors/Electro-Motive Division
IC	Internal Combustion
ID	Induced Draft
IGCC	Integrated Gasification Combined Cycle
IPP	Independent Power Producer
LHV	Lower Heating Value
MAF	Moisture and Ash Free
METC	Morgantown Energy Technology Center
MIT	Massachusetts Institute of Technology
MW_e	Megawatt (Electric)
MW_t	Megawatt (Thermal)
NESCAUM	North East States for Coordinated Air Use Management
NUG	Non-Utility Generator
PC	Pulverized Coal
PFBC	Pressurized Fluidized Bed Combustion
PURPA	Public Utilities Regulatory Policy Act
RAL	Rotary Air Lock
RCRA	Resource Conservation and Recovery Act
RPM	Revolutions Per Minute
SCR	Selective Catalytic Reduction
SNR	Selective Non-catalytic Reduction
SS	Stainless Steel
STIG	Steam-Injected Gas Turbine
SwRI	Southwest Reasearch Institute
WC	Water Column

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Abstract

Under the sponsorship of the Department of Energy, Morgantown Energy Technology Center, Cooper-Bessemer and Arthur D. Little have developed the technology to enable coal-water slurry to be utilized in large-bore, medium-speed diesel engines. The target application is modular power generation in the 10 to 100 MW size, with each plant using between two and eight engines. Such systems are expected to be economically attractive in the non-utility generation market after 2000, when oil and natural gas prices are expected to escalate rapidly compared to the price of coal.

During this development program, over 1000 hours of prototype engine operation have been achieved on coal-water slurry (CWS), including over 100 hours operation of a six-cylinder, 1.8 MW engine with an integrated emissions control system. Arthur D. Little, Inc., managed the coal-fueled diesel development, with Cooper-Bessemer as the principal subcontractor responsible for the engine design and testing.

Several key technical advances which enable the viability of the coal-fueled diesel engine were made under this program. Principal among them are the development and demonstration of (1) durable injection nozzles; (2) an integrated emissions control system; and (3) low-cost clean coal slurry formulations optimized for the engine. Significant advances in all subsystem designs were made to develop the full-scale Cooper-Bessemer coal engine components in preparation for a 100-hour proof-of-concept test of an integrated system, including emissions controls. Key achievements in 1992-1993 were:

- The full-scale (six cylinder, 1800 kW) Cooper-Bessemer Model LS engine was assembled and demonstrated on coal-water slurry fuel. Two hundred hours of full-scale engine testing were achieved at the Cooper-Bessemer test facility.

- An improved, lower-cost slurry preparation approach, in which an "engine grade" coal cleaning module is integrated with conventional mine-mouth cleaning, was developed and demonstrated by CQ Inc. A full scale, 6500 gallon slurry storage and handling system was fabricated and demonstrated.

- The full-scale 1.8 MW emissions control system was installed and demonstrated at the Cooper-Bessemer test facility. NO_x emission data indicated that the coal diesel is competitive with gas turbines (coal diesel NO_x emissions were 0.11 lb/MMBtu). SO_2 and particulate emissions were below those of competitive, coal-based power generation technologies.

The Clean Coal Diesel power plant of the future will provide a cost-competitive, low-emissions, modular, coal-based power generation option to the non-utility generation, small utility, independent power producer, and cogeneration markets. Combined cycle efficiencies will be approximately 48 percent (lower heating value basis) and installed cost will be approximately $1300/kW (1992 dollars).

I. Introduction and Summary

A. Background and Summary

Interest in coal-fueled heat engines revived after the sharp increase in the prices of natural gas and petroleum in the 1970's. Based on the success of micronized coal water slurry combustion tests in an engine in the 1980's, Morgantown Energy Technology Center (METC) of the U.S. Department of Energy initiated several programs for the development of advanced coal-fueled diesel and gas turbine engines for use in cogeneration, small utilities, industrial applications and transportation.

Cooper-Bessemer and Arthur D. Little have been developing technology since 1985, under the sponsorship of METC, to enable coal water slurry (CWS) to be utilized in large bore, medium-speed engines. Modular power generation applications in the 10-100 MW size (each plant typically using from two to eight engines) are the target applications for the late 1990's and beyond when, according to the U.S. DOE and other projections, oil and natural gas prices are expected to escalate much more rapidly compared to the price of coal.

As part of this program over 1050 hours of prototype engine operation have been logged on coal water slurry, including over 100 continuous hours operation of a six-cylinder full-scale engine with Integrated Emissions Control System in 1993. Under DOE-METC support, the technology has made rapid progress toward commercial readiness. This novel diesel engine-based technology offers 45% simple cycle efficiency and NO_x emission rates below 0.2 lb/MMBtu with selective catalytic reduction (SCR) (comparable to natural gas-fueled gas turbines). The fuel is a low-cost, coal-based liquid with the consistency of black paint, composed of 12-micron mean size premium 2% ash coal dust mixed 50/50 with water. Arthur D. Little, Inc., managed the coal-fueled diesel development, with Cooper-Bessemer as the principal subcontractor responsible for engine design and testing.

The Clean Coal Diesel Plant of the future is targeted for the 10-100 MW non-utility generation (NUG) and small utility markets, including independent power producers (IPP) and cogeneration. A family of plant designs will be offered using the Cooper-Bessemer 3.8, 5.0, and 6.3 MW Model LS engines as building blocks. In addition, larger plants will be configured with an engine in the 10-25 MW class (Cooper-Bessemer will license the technology to other large bore stationary engine manufacturers).

Although somewhat rare in the U.S., it is quite common worldwide to install larger diesel plants configured with engines in the 10-25 MW capacity range; in fact, the 10-25 MW class diesel engines offer a heat rate efficiency of 45% vs. 40% simple cycle for the smaller engines. While a plant can be built as small as 2 MW (based on the Cooper-Bessemer Model LS-6 engine), our cost projections indicate that an

8 MW plant is likely to be at the lower end of what is economically attractive. It should also be noted that the coal diesel plant can also be configured for cogeneration applications. Figure I-1 shows a typical plant layout for a 14 MW diesel plant with SCR and heat recovery boiler.

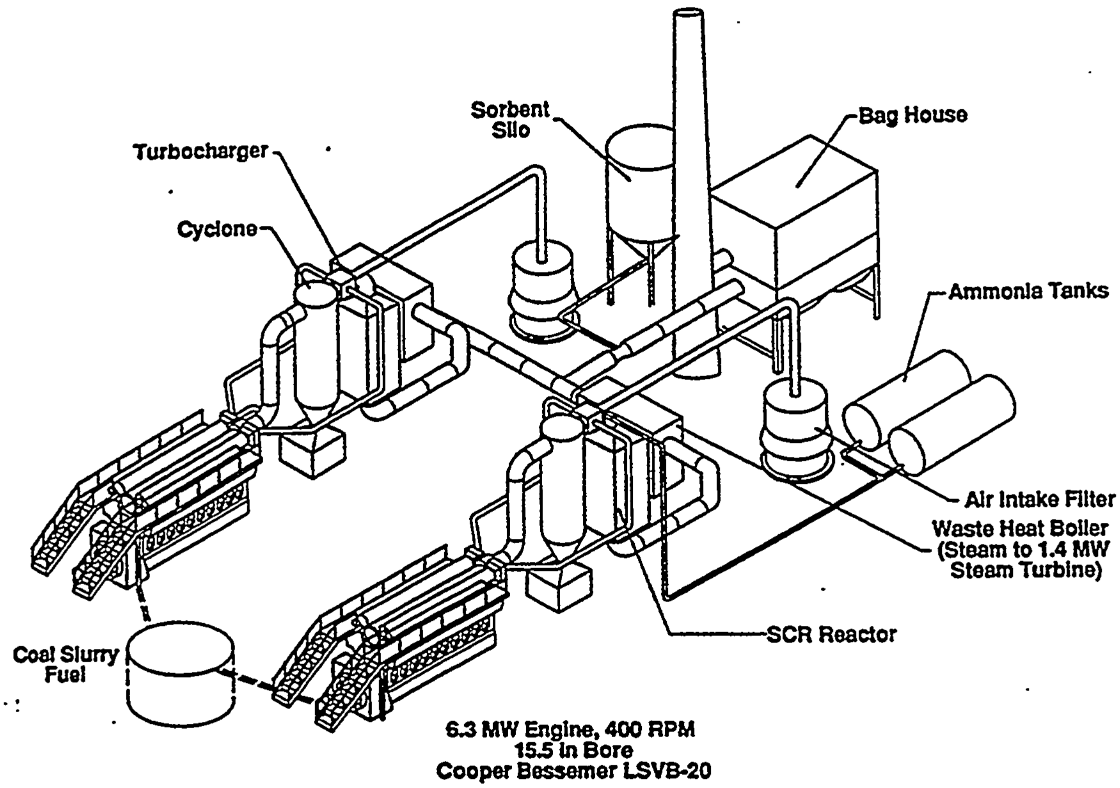

Figure I-1. 14 MW Coal-Fueled Diesel Powerplant with Steam Turbine Bottoming Cycle

The reciprocating engine offers a remarkable degree of flexibility in selecting plant capacity. This flexibility exists because the engines are modular in every sense (fuel cell stacks have similar modularity). Scale-up is accomplished simply by adding cylinders (e.g., 20 vs. 16) or by adding engines (4 vs. 3). There is no scale-up of the basic cylinder size. Thus, there is essentially no technical development needed to scale-up the Cooper-Bessemer Clean Coal Diesel Technology all the way from 2 MW (one 6-cylinder engine) to 50 MW (eight 20-cylinder engines), other than engineering adaptation of the turbocharger to match the engine.

The emissions control system for the coal diesel plant includes the following:

- Cyclone separators remove the large particulate upstream of the turbocharger.
- NO_x control is achieved by combustion optimization, selective catalytic reduction (SCR) and reduction across the duct injection system.
- SO_x control is achieved by duct injection of sodium bicarbonate followed by sorbent separation in a fabric filter.
- Final particulate control is achieved by use of a fabric filter.

Use of advanced diesel engines which operate at high brake mean effective pressure, suitably converted for coal-water slurry firing, in future commercial coal diesel plants can lead to combined-cycle generating efficiencies of up to 50 percent.

B. Coal Diesel Advantages Compared to Other Clean Coal Power Plant Technologies

The Clean Coal Diesel will offer the following performance characteristics beginning in the 2005-2010 timeframe:

- Installed cost $1300/kW (1992 dollars)
- Efficiency 48.2% (LHV, combined cycle) (demonstrated: 41% - LHV, simple cycle)
- NO_x emissions 0.11 lb/MMBtu (demonstrated: 0.18 lb/MMBtu)
- SO_x emissions 0.2 - 0.4 lb/MMBtu, depending on coal sulfur content (demonstrated: 0.1 - 0.2 lb/MMBtu)

The advantage of this 10-100 MW clean coal diesel technology is that it is targeted for NUG and small utility capacities, whereas all other clean coal technologies have been designed for the central station utility market (generally 200-500 MW).

•	IGCC	200-500 MW
•	PFBC	100-300 MW
•	Fuel cell with Integrated Gasification	200-500 MW

Fuel cell technology is under rapid development and, although initially more costly, fuel cell power plants using natural gas reformers will eventually compete in the 100 kW-10 MW range. Figure I-2 illustrates the unique market position of the clean coal diesel with respect to its competitors.

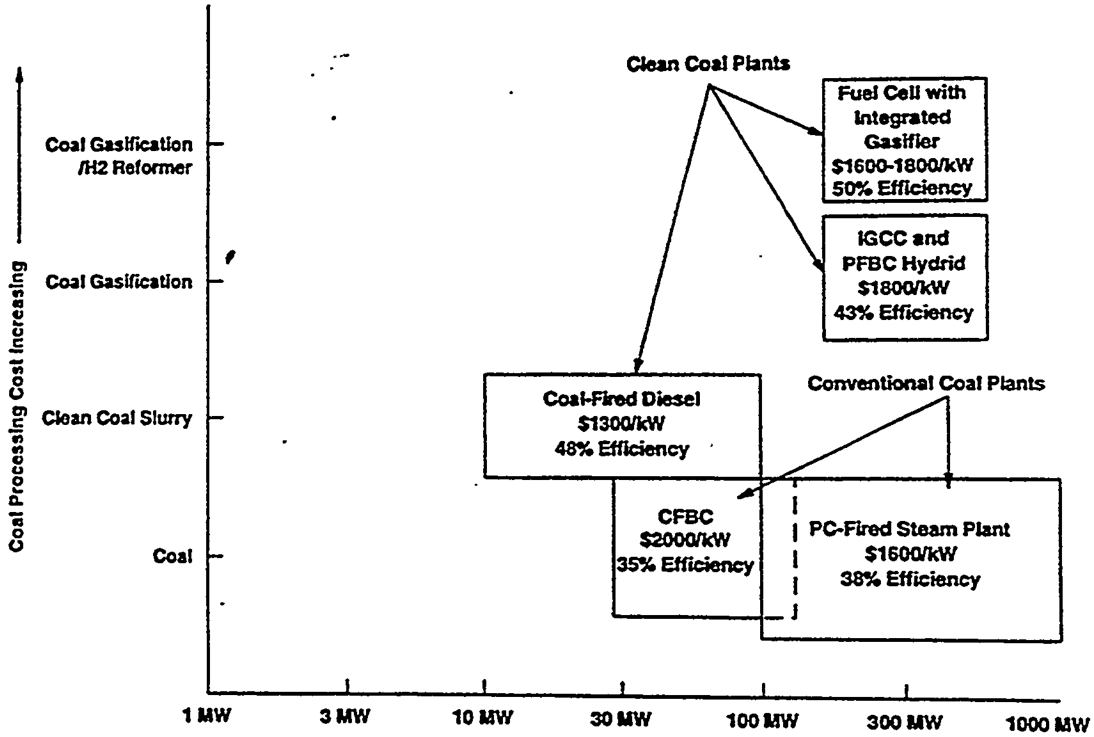

Figure I-2. Coal Diesel Advantages in Sub-100 MW Applications

In the early market introduction (2000-2010), the clean coal diesel will compete against both natural gas technologies and small coal plants (10-100 MW) of the PC, AFB, and stoker variety. The more favorable the price difference between coal and gas (or oil), the more competitive the coal diesel will be.

C. Technology Status/Program Accomplishments

Past DOE/METC coal-fueled diesel engine programs by Cooper-Bessemer/Arthur D. Little (ADL), Sulzer, General Electric (GE), Adiabatics, General Motors/Electro-Motive Division (GM/EMD), Detroit Diesel Corporation (DDC), and Southwest Research Institute (SwRI) have not only defined economic and technical conditions under which future coal engines can be commercialized, but also advanced the state of the art through exploratory engine testing with novel injectors. Many component problems had been uncovered by the Germans between 1920 and 1945, when they attempted to utilize coal dust in large stationary diesel engines.

Three technology advances which enabled the coal-fueled reciprocating engine to be viable have been made in the current METC program:

- Durable injection nozzles were developed which show virtually no wear to the extent they could be tested (hundreds of hours).

- An emissions control subsystem was perfected to bring levels of NO_x particulate, and SO_2 emissions below those of competitive coal power technologies.

- Lower cost clean coal slurry formulations were optimized for the engine.

In the final two years of this program (1992-1993), the Cooper-Bessemer prototype engine was demonstrated with all three of these technology advances, including over 100-hours of proof-of-concept testing.

Significant subsystem advances were made to develop the 6-cylinder, 1.8 MW Cooper-Bessemer coal engine components in preparation for the 100-hour proof-of-concept testing of the integrated system. The key achievements in 1992-1993 were as follows:

- The full-scale (six cylinders--1800 kW) Cooper-Bessemer Model LS engine (coal configuration) was assembled and demonstrated on clean coal slurry. This engine burns 15 times the amount of coal as the previous single-cylinder JS-1 engine (2.5 x per cylinder). This same engine can be built in 6.3 MW configuration without any scale-up of cylinder size--simply by increasing the number of cylinders from 6 to 20.

- An improved lower-cost slurry preparation approach was developed by CQ Inc. Integration of the "engine grade" coal cleaning module with conventional mine mouth cleaning lowers cost. Now, with this revised approach, the clean coal can be shipped dry (by truck, rail or barge) economically over longer distances and simple modular on-site slurry plants are economical (5-20 ton/hr typical size).

- A full scale, 6500 gallon slurry storage and handling system was fabricated and demonstrated at the test facility.

- The full-scale 1.8 MW emissions control system was demonstrated; NO_x emission data indicated that the coal diesel is competitive with gas turbine NO_x emissions (0.11 lb/MMBtu).

- 200 hours of engine testing was achieved on coal water slurry. The engine tests have provided valuable insights related to component design. Figure I-3 summarizes the cumulative 1050 hours of engine testing achieved.

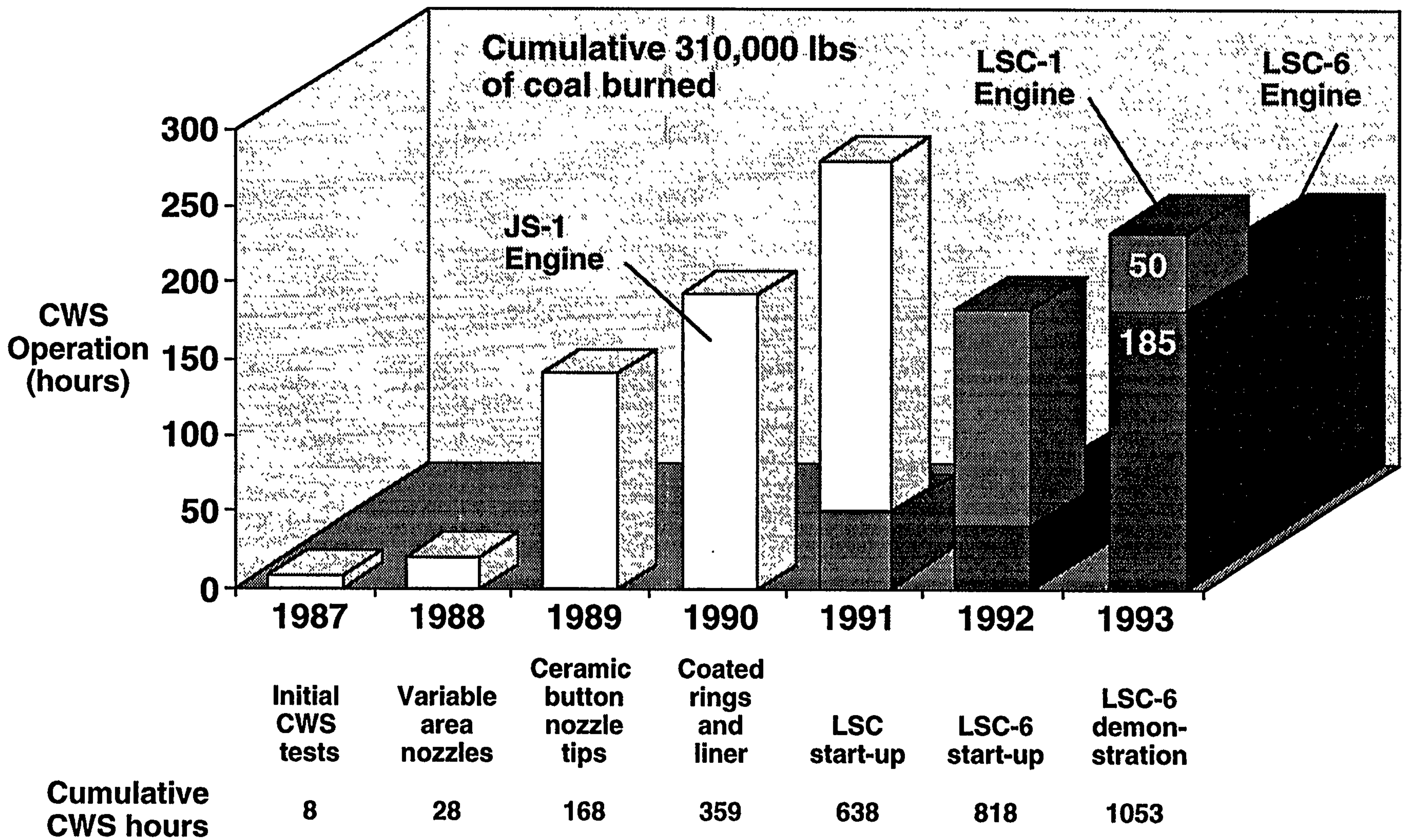

Figure I-3. History of Engine Test Experience

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II. Overview of Coal-Fuels Development Activity

The objective of the Coal-Fuels activity was to support the development of the coal-fuel diesel engine by optimizing the fuel characteristics so as to reduce wear while promoting satisfactory combustion and fuel handling. The main accomplishments were as follows:

- Specification, preparation, characterization, and delivery of coal-water slurry fuel (CWS) in several-thousand-gallon batches to support tests of the engine, injection system, and other components;

- Design, fabrication, and testing of subsystems for CWS storage, handling, and transfer for use at Cooper Bessemer's R&D facility in Mt. Vernon, OH, and at CQ Inc.'s CWS production facility;

- Identification of appropriate fuel specifications for producing low-cost CWS and corresponding coal feedstocks; and

- Development of a conceptual design for a coal cleaning/slurry preparation plant that could produce engine-grade CWS on a commercial scale (typically 250 ton per hour) at a processing cost of $30/ton or less (including amortization of the coal cleaning plant). Depending on the cost of the coal feedstock, this would correspond to CWS prices of $3.00/MMBtu or less delivered to the powerplant.

Over the course of the coal-engine development program, four CWS producers participated in these activities. They were: AMAX Research and Development Center, Energy International, CQ Inc. and Otisca Industries. The areas of contributions of these companies and the schedule of their involvement in the program is summarized in Table II-1.

AMAX produced small batches of chemically cleaned coal fuels (<0.5% ash) for early single cylinder tests on the JS engine. They also produced larger, (1000 gallon) batches of physically cleaned coals (ash content <1.5% ash) for development work on the JS-1 from 1988 - 1989. AMAX developed formulations for well stabilized slurries and conducted a study of storage and handling characteristics of slurries prepared with various additive packages. AMAX also contributed to the economic analysis of CWS plant design alternatives with the development of a spreadsheet-based, CWS plant costing tool. In addition, AMAX R&D prepared a survey of potential feedstocks for the production of engine-grade CWS.

Energy International (EI) worked with CQ Inc. to prepare fuels according to conventional coal cleaning and grinding techniques. EI expanded on the formulation work conducted by AMAX to specify the grinding and additive requirements for slurries prepared in large batches. EI explored a variety of alternative cleaning techniques and prepared a series of CWS fuels used to test the effect of particle size, viscosity, coal type and ash content on engine and injection system performance.

CQ Inc. served as our main fuel supplier and produced over 44,000 gallons of CWS for testing of the LS-1 and LS-6 engines. They have designed, tested, and operated large-scale CWS preparation circuits and CWS storage equipment. They developed a conceptual design for a multi-product coal cleaning facility that produces conventional boiler fuel and engine grade coals at potential cost advantages compared to a single-product plant. Also, CQ Inc. developed and executed the logistics of delivering CWS by tanker truck to Cooper-Bessemer over the 100-hour continuous run.

Otisca Industries produced over 34,000 gallons of CWS for JS and LS engine testing. Otisca contributed to the design of the CWS storage and transfer system at Cooper Industries in Mt. Vernon. Otisca provided design guidelines for the jet mixing system which is the basis of the 500 gallon and 6,500 gallon storage tanks at Cooper. They also modified the pumps for use with slurry. Otisca assembled and tested the 6,500 gallon storage system at their plant in Syracuse NY prior to its installation at Cooper. Otisca provided detailed process and costing information that enabled us to include the oil-agglomeration process in the spreadsheet-based costing model developed by AMAX.

Table II-1. Companies Participating in CWS Development

Company	Role	Dates
AMAX R&D	CWS production • chemical cleaning • physical cleaning Cost modelling CWS handling study Coal feedstock survey	1986 - 1990
Energy International	CWS formulation Alternative feedstock study	1990 - 1992
CQ Inc.	CWS production • heavy media cycloning Commercial plant conceptual design CWS logistics	1990 - 1994
Otisca Industries	CWS production • oil agglomeration CWS storage facility ◦ design /testing CWS plant costing evaluation	1990 - 1993

The results of the coal-fuel development activities in CWS production, handling, specification, and conceptual plant design are presented in the following sections.

A. Coal Slurry Production

At the start of the engine development program, exploratory engine tests were run with ultra-low (<0.5%) ash coal fuels produced by chemical cleaning. Early fuel specifications were stringent, requiring small particle size (<44 microns), high solids loading (>55%), low viscosity (<100 cp) and high rank bituminous coal. Although these coal slurries performed well in terms of handling, injection and burning, their projected cost for use in a commercial coal-engine facility was too high. To compete with oil and gas in the 2000-2010 timeframe, slurry fuels must cost no more than $3.00/MMBtu delivered to the powerplant, including coal cost, which allows only $1.00 to $1.50/MMBtu for processing. Therefore, the team examined avenues for relaxing CWS specifications and identified alternative coal cleaning and CWS preparation technologies that could produce engine grade CWS on a commercial scale for much lower cost.

1. CWS Specification for Engine Testing

Based on extensive testing, the specifications for lower cost coal slurry for the LSC 6-cylinder engine tests were established as follows: 2% ash or less, 88 micron top size, 12-15 micron mean size, 51% max solids, and <200 cP viscosity. (The process by which these specifications were developed and the impact of these specifications on CWS cost and engine performance are discussed below in Section IIC, p. II-21.). Over 44,000 gallons of slurry were produced at CQ Inc. for engine testing. Clean coal for this slurry was produced using conventional, heavy media cyclones in the circuit described below. The grinding circuit and additive package used by CQ Inc. to produce the fuel was developed in partnership with Energy International. This development effort is also discussed in Section IIA-2 which follows. Additional coal-water slurry prepared by Otisca was used for LS-1 engine tests and was stored as a "back-up" fuel for LS-6 engine tests. The Otisca oil-agglomeration coal cleaning circuit and the slurry preparation circuit used to prepare this back-up fuel is also described in the section below.

2. Coal Cleaning and Preparation Circuit (CQ)

Starting in 1990, CQ Inc. produced and delivered coal-water fuel to Cooper-Bessemer to meet scheduled test requirements of the coal-engine development program. All CWS fuel obtained from CQ was produced at CQ Inc.'s Coal Quality Development Center located near Homer City, Pennsylvania.

Coal Feedstock for CQ Baseline Fuel

CQ Inc. procured coal from the Wentz Mine of Westmoreland Coal Company located in Wise County, Virginia. This Taggart Seam coal was cleaned at the commercial cleaning plant at the mine to approximately 3.0 percent ash content (on a dry basis).

The coal was delivered by truck to CQ Inc.'s Coal Quality Development Center (CQDC) in Indiana County, Pennsylvania. The quality (dry basis) of a typical shipment of this coal received at the CQDC was:

Ash (Wt %)	2.69
Sulfur (Wt %)	0.63 (equivalent to 0.83 lb/MMBtu of SO_2)
Heating Value (Btu/lb, MAF)	15,149

A complete description of the raw coal characteristics is included in Appendix A. Coal shipped from the mine had a nominal top size of two inches. When the coal was received at CQ Inc., it was dumped into an in-ground receiving hopper, crushed to 3/8-inch topsize in a Gundlach two-stage, four-roll crusher, and stored in one of five 100-ton storage bins. The coal receiving system is equipped with a mechanical sampling system which was used to collect a representative sample of the coal for laboratory analysis.

Description of Heavy-Media Cyclone Circuit at CQ Inc.

The coal was cleaned using a heavy-media cyclone circuit. Feed coal was metered from the storage bins using weight feeders and conveyed at 15 ton/hr to the coal cleaning plant, which was configured to clean the coal using the heavy-media cyclone circuit (see Figure II-1). The coal was mixed with water; the minus 28 mesh (0.6 mm) coal was removed and discarded using a combination of a sieve bend and vibrating screen. The 3/8 inch x 28 mesh feed coal was then mixed with a suspension of magnetite and water and pumped to a 14-inch diameter heavy-media cyclone. The specific gravity of the media was controlled to effect a separation at the appropriate specific gravity (1.28-1.30) to provide the low ash product.

The cyclone separated the feed coal into two streams, one containing specification quality clean coal and the other containing higher ash coal suitable for use in a conventional coal-fired power plant. Magnetite was drained and rinsed from the clean coal and rejects, also using a combination of sieve bends and vibrating screens. The drained media was recirculated. The media, diluted by rinse water, was reconcentrated using a drum magnetic separator and recirculated. The clean coal was dewatered in a basket centrifuge and conveyed to a ground pile to be transferred to storage. The coal was cleaned to 1.5-2.0 percent ash (dry basis) with a yield of approximately 65%.

The quality of a typical batch of clean Taggart Seam coal produced by this circuit was as follows (on a dry basis):

Ash (Wt %)	1.50
Sulfur (Wt %)	0.59 (equivalent to 0.78lb/MMBtu of SO_2)
Heating Value (Btu/lb)	15,074

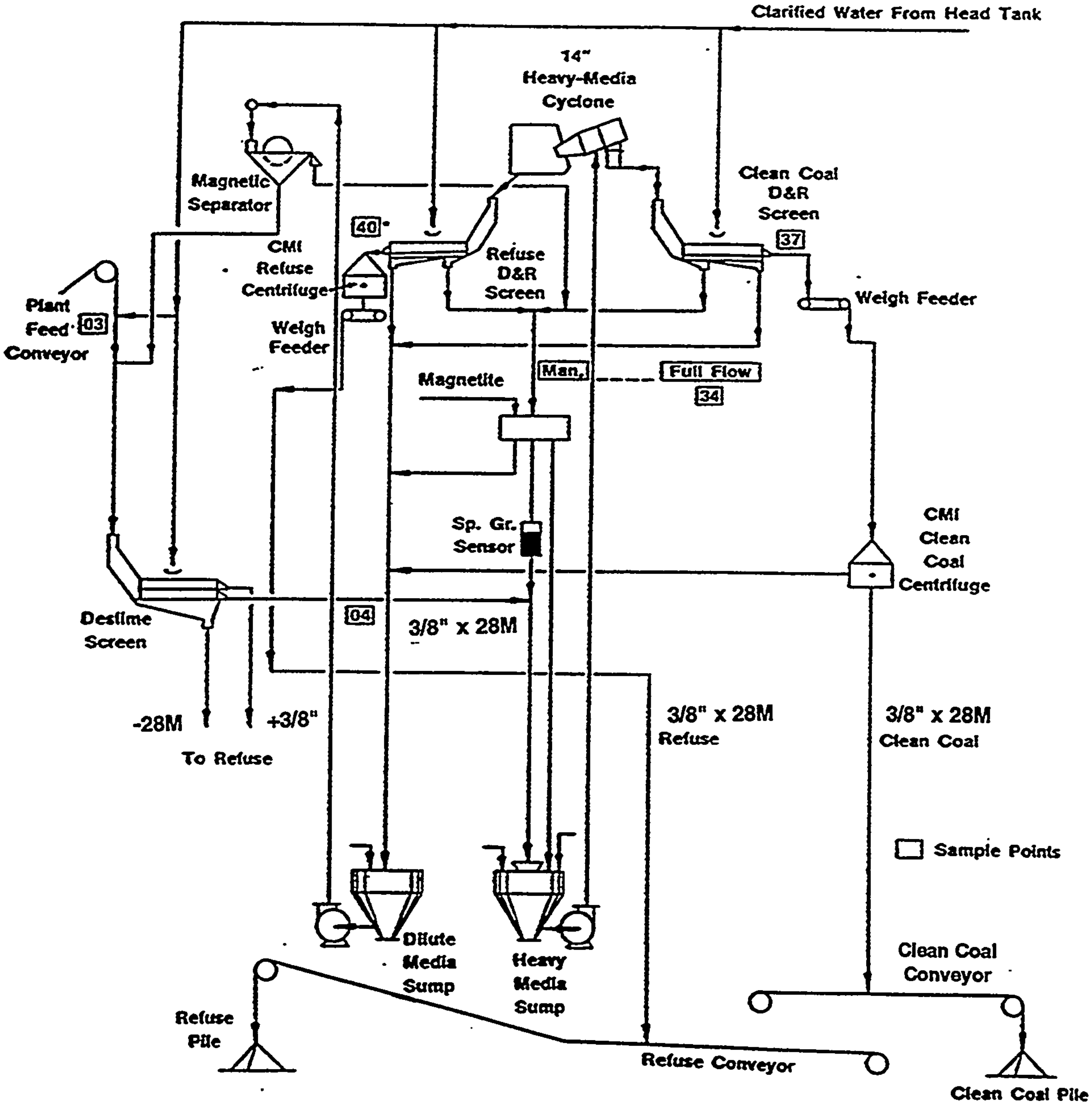


Figure II-1: Heavy-Media-Cyclone Flowsheet

The re-cleaned coal was loaded into storage bins to be fed to the coal-water fuel preparation circuit.

CWS Preparation Circuit at CQ Inc.

CQ Inc. devised a continuous grinding circuit to produce coal water fuel with similar properties to the coal-water fuel developed by Energy International for the coal-diesel application. The specifications for the coal-water fuel were:

- Mass mean diameter of less than 15 μm, maximum less than 88 μm (measured by Microtrac™ particle size analyzer).
- 50.0 to 53.0% (51.5%target) solids loading.
- Viscosity of <200 centipoise @ 100-1000/sec. (measured by a Haake rotoviscometer).

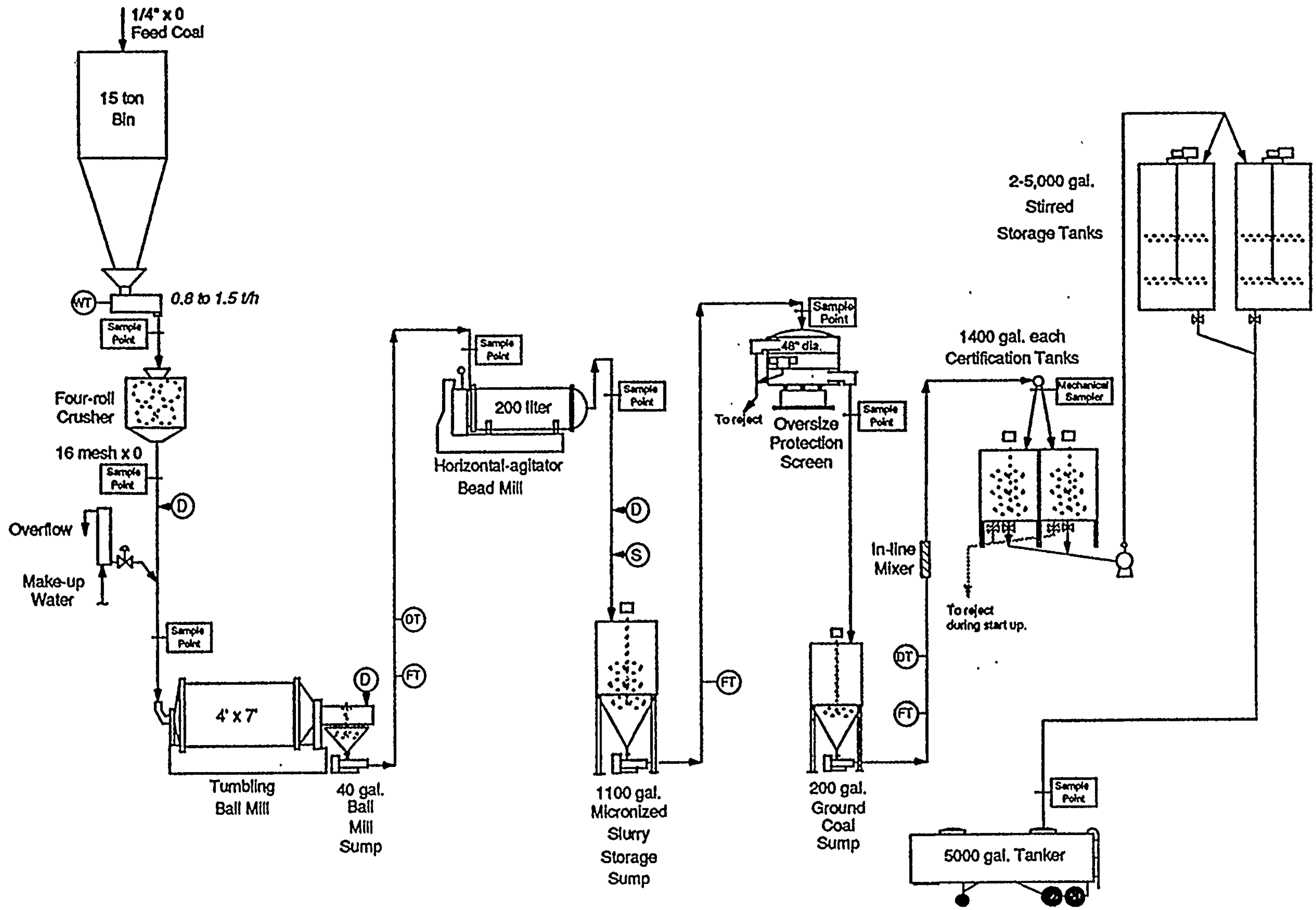
The grinding circuit used is shown in Figure II-2. Clean coal was metered from the 15-ton bin at a rate of about 1.1 tons/hour. The coal (3/8" top size) was crushed to 16 mesh (.04") in a Roskamp two-stage, four-roll crusher. The crushed coal was then mixed with half the total dispersant loading and enough water to obtain about one percentage point over the desired solids content. The coal-water-dispersant mixture was fed to a MPSI 4 ft X 7 ft tumbling ball mill. The ball mill product discharged via a trommel screen to prevent loss of the grinding media. A quarter of the total dispersant is added to the ball mill discharge.

The ball-milled coal was pumped to a 200-liter Netzsch horizontal-agitator bead-mill where its size was further reduced. Stabilizer (xantham gum) and the last quarter of the dispersant was added to the bead-milled discharge coal and pumped to a 48-inch diameter oversize protection screen. The screen was fitted with a 48 Tyler mesh (295 μm) deck. The oversized material was discarded.

The slurry was sampled and transferred to one of two 1400-gallon certification tanks. The slurry was sampled and analyzed for size distribution and density.

Once the slurry was certified, it was drained to storage tanks for storage until time for shipment to Cooper-Bessemer. Samples of coal-water fuel were analyzed by CQ Inc. for the following:

- Particle size (mean, max, and distribution)
- Low-shear Viscosity
- Solids Concentration
- Ash Content
- Heating Value
- Total Sulfur



t/h - Dry solids ton per hour
DT - Density transmitter
FT - Flow transmitter

WT - Weight transmitter
D - Dispersant additive(s)
S - Stabilizing additive(s)

Revision 4.0
26 September 1991

Figure II-2. Coal-Water Fuel Preparation Flowsheet

Coals Produced at CQ Inc.

CQ Inc. produced approximately 42,000 gallons of CWS for engine testing. These fuels were prepared in batch sizes ranging from 2000 gallons to 15,300 gallons each from November, 1991 to August 1993. A complete list of CQ slurries prepared is included in Table II-2. The properties of the CQ-prepared CWS used in the 100-hour engine test are included in Appendix A.

CWS Preparation Circuit at Energy International

Energy International (EI) used a 2x 2 ft wet ball mill to prepare small batches (2 - 8 barrels) of coal water slurry for engine testing at Cooper-Bessemer. EI prepared slurries that were used to evaluate the impacts of top particle size (65 microns compared to 44 microns), additive package, and coal type on engine performance. The coals used for the grinding and formulation studies were cleaned at CQ Inc. The specialty coals used to assess the impact of coal type and volatile content on engine performance were cleaned at Energy International and Virginia Polytechnic Institute using column flotation, froth flotation, oil agglomeration and heavy liquid separation techniques.

Additive Evaluation Studies at Energy International

Energy International conducted bench-top studies of CWS formulation as part of an effort to transfer formulation technology from AMAX R&D and improve the formulations where possible. They investigated the effects of additive quantities and points of introduction to the slurry on CWS viscosity and stability. The additives they used are listed below in Table II-3.

Table II-3. Additives Used in CWS Formulation

Additive	Trade Name	Constituent
Dispersent	MCG-32A-4S	Napthalene sulfonic acid polymer ammonium salt
Stabilizer	Flocon	Xanthan gum
Biocide	-	Formaldehyde

EI developed the 'distributed' additive strategy for CWS formulation. This strategy involved adding dispersant (1) before the ball mill, (2) between the ball mill and the bead mill, and (3) after the bead mill, as a means of completely mixing the additive into the slurry.

CWS Characterization by Energy International

Energy International characterized slurry properties for the slurries that they and CQ

Table II-2. CWS Batch Summary

Producer/ Designation	Date Produced	Amount	Purpose/ Description	Test Facility
AMAX F	12/88	8 barrels	Baseline; Chemical Clean, 0.4% ash	JS-1 engine
AMAX G	2/89	12 barrels	Baseline; Chemical Clean, 0.4% ash	JS-1 engine
AMAX H	6/89	30 barrels	Baseline; Chemical Clean, 0.5% ash	JS-1 engine
AMAX I	6/89	1 barrel	Test High Viscosity; (600 cP instead of 120 cP)	AMBAC Injection Rig
AMAX J	7/89	2 barrels	Test Coal Type; Upper Freeport; Low Volatile	JS-1 engine
AMAX K	8/89	2 barrels	Test Coal Type; Colowyo; Sub-bituminous	JS-1 engine
AMAX L	8/89	1 Barrel	Test Coal Type; Raton Creek; Western Bit	JS-1 engine
AMAX M	8/89	1 Barrel	Test Coal Type; Minnehaha; IL Basin	JS-1 engine
AMAX N,O,P	9/89	14 barrels	Baseline; Chemical Clean; 0.4% ash	JS-1 engine
AMAX Q	11/89	2 barrels	Physical Clean; Blue Gem; 1.2% ash	JS-1 engine
Otisca 2 - 5	2/90	8,000 gallons (160 barrels)	Baseline; 1.3% ash; Taggart	JS-1 engine
AMAX Q2, R	3/90	8 barrels	Particle Size Tests; Viscosity Tests;	SwRI rig
EI UE3-310	5/90	2 barrels	Check-out test; HMC clean at CQ, 1.4% ash	JS-1 engine
EI UE3-311	6/90	2 barrels	Check-out test; HMC clean at CQ, 1.9% ash	JS-1 engine
Otisca Solids Tests	6/90	12 barrels	Test effect of solids content; 50% - 42%	JS-1 engine
EI-UE3-317,320, 321	7/90, 8/90	8 barrels	Test effect of larger particle size (75 micron top vs. 44 micron top)	JS-1 engine
CQ Check-out UE3-336,337,338	11/90	6 barrels	Check-out fuel. Test effect of water quality on CWS characteristics	JS-1, storage tank tests
Otisca	12/90	25,000 gallons	For 100-hr test, 2.4% ash; Blue Gem coal	LS-6 engine (not used)
CQ Inc.	11/91	2,000 gallons	Baseline Fuel	LS-1 engine
CQ Inc.	1/92	400 gallon 1600 gallon	Baseline Fuel; (higher stabilizer content in larger delivery)	LS-1 engine
CQ Inc.	1/92	5,000 gallons	Storage tank tests; Baseline fuel	4,500 gal to Otisca for tank tests, 500 to CB for LS-1 engine tests
EI	2/92	6 barrels	Coal Feedstock tests; Brookville; Lower Block; Pocahontas	LS-1 engine (not tested)
CQ Inc.	9/92	10,000 gallons	Baseline fuel	LS-6 engine
CQ Inc.	6/93 - 8/93	25,300 gallons	Baseline fuel; 100-hour engine test	LS-6 engine

Inc. prepared. The characterization they provided included coal particle size distribution, high shear viscosity and low shear viscosity. These analyses were conducted using a laser diffraction particle size analyzer (Leeds and Northrup Microtrac); a Burrell extrusion tube viscometer, and a Haake viscometer, respectively.

Coal Cleaning and Preparation Circuit at Otisca

Otisca Industries, Inc. developed a coal cleaning process that relies on ultra-fine grinding of coal to liberate ash followed by selective agglomeration to separate coal macerals from minerals at very high levels of energy recovery. Otisca Industries operates a 15 ton/hr coal cleaning and CWS production plant for relatively large-batch slurry production (>20,000 gallons per batch) as well as laboratory scale cleaning facilities for smaller batch preparation. Otisca patented, developed and operated a 7.5 ton/hr stirred ball mill for energy efficient and cost-effective fine coal grinding.

Otisca Industries prepared CWS for engine and injector testing. Some of the slurry was prepared at the 15 ton/hr coal preparation facility in Jamesville, NY, while other batches were prepared in the laboratory facilities in Syracuse, NY. One order of CWS (20,000 gallons, December, 1990), was prepared by grinding and slurrying a Blue Gem coal that was received from the mine at less than 2% ash.

In addition to their coal cleaning and slurrying facilities, Otisca provided the program with expertise in slurry storage and handling. The 6,500 gallon CWS storage tank and circulation system used at Cooper-Bessemer's Mt. Vernon, OH facility was assembled and tested at Otisca Industries.

3. CWS Batches Prepared

Table II-2 summarizes the batches of coal-water slurry prepared between 1988 and 1993 for JS and LS engine tests, injection system tests, and coal cleaning/preparation tests.

4. CWS Production Lessons Learned

The following key lessons in engine-grade CWS production were learned from the development activities described above.

- **The use of conventional coal cleaning equipment (heavy media cyclones) is preferable to produce engine-grade slurry, but will be economical only if middlings are sold as a boiler fuel.** Conventional heavy media cyclone yields were found to be 40-70% for the most cleanable of coal feedstocks. The use of this type of low cost, conventional equipment is desirable but must be incorporated into a multi-product plant strategy to be cost efficient.

- **Avoiding contamination to the raw coal stream, clean coal stream and coal slurry storage is critical to maintaining CWS quality.** Small quantities of contamination can raise the ash content of these slurries significantly. High quality coal feedstocks and cleaned coal should be stored in covered bins with concrete or asphalt floors or in silos. CWS production must be quality controlled to ensure no foreign matter (rags, wood, metal flakes, etc.) contaminate the CWS stream. CWS transfer trucks must be cleaned before use.

- **Water quality can impact the slurry handling characteristics.** Water used to produce CWS should be low in minerals, particularly calcium, sodium, and iron ions.

- **Additives can be optimized by selecting both proper quantity and the process point at which they are added.** Dispersant is more effective when some is added upstream of the grinding process, to ensure complete mixing of the additive with the coal. Stabilizer appears to loose its effectiveness after about a year of CWS storage in a high-shear mixing environment. Addition of supplemental stabilizer can reduce the rate of settling but cannot eliminate it. The oil-antiagglomerant additive is always best added at the point of use (i.e., at the engine site, immediately prior to slurry use). Slurry prepared by heavy-media cycloning requires oil-antiagglomerant only during periods close to slurry start-up or shut-down when a slurry/diesel fuel interface could be present in the fuel lines.

- **The screening of CWS to remove oversize particles and tramp material is necessary to prevent large particles from contaminating the slurry.** The screening process, however, is not 100% efficient and does remove some undersize coal. A grinding circuit should be configured such that the reject coal from the final screen is returned to the ball mill.

- **Variability in CWS handling characteristics can occur from batch to batch even through process variables and additive package are tightly controlled.** The variability in CWS stability and consistency appears to be a property of CWS and must be incorporated into the design assumptions for CWS storage, handling and injection equipment.

- **An ideal slurry preparation plant will maximize the use of gravity flow and minimize the use of slurry pumps (to minimize maintenance costs).** All slurry lines need to be equipped with clean outs and must be drained when not in use.

- **When operating a heavy-media cyclone to make a low-specific gravity separation, the circulating media must be held within 0.0005 specific gravity points because of the high amounts of near gravity, clean coal material.**

B. Coal Slurry Storage, Handling, and Transportation

A CWS storage and transfer system for use with the LSB-6 engine was designed, fabricated, tested and operated for almost three years. A schematic of the system is shown as Figure II-3. The storage tank used is a modified tanker truck with a jet mixer to keep the slurry suspended. The heat generated from operation of the jet mixer is sufficient to keep the slurry above freezing temperatures in the winter. A

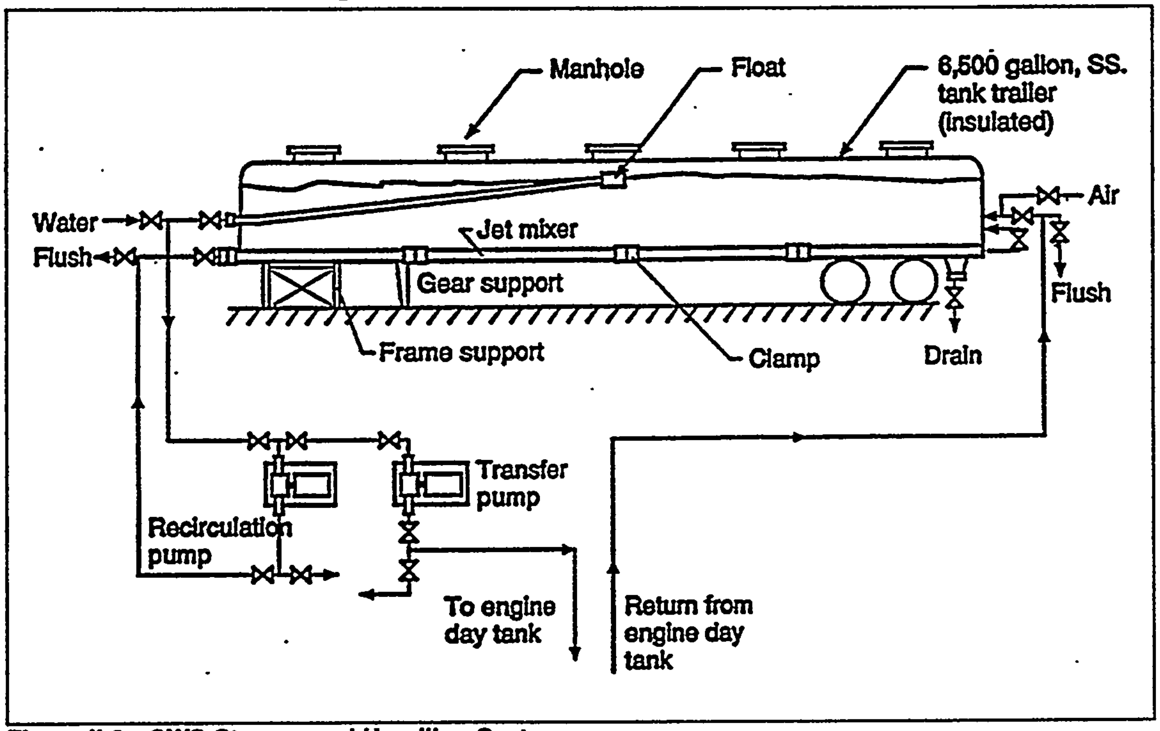


Figure II-3. CWS Storage and Handling System

control system was developed that automated the recirculation function, flushed the pumps to avoid CWS clogging, provided system diagnostics in the event of a failure, automated the refill of the engine day tank, and provided warning and safety signals for CWS loading into the tank.

A second CWS storage facility was designed, tested and operated at CQ Inc. This storage facility included two, vertical, 5,000 gallon tanks, each stirred continuously with a low speed mixer. The operation of both these facilities are described in the sections below.

1. Recirculating Tank Design and Operation (Cooper-Bessemer Facility)

Jet Mixer

The jet mixing system used high velocity jets to entrain coal particles that began to settle, and remixed them into the bulk slurry. The jet mixer was configured to be able to remix settled slurry, or keep slurry suspended. This mixer was a round pipe, 4" in diameter, with small holes (3/16" diameter) drilled along the length, on both sides of the pipe. The discharge end of the mixer was sealed. Slurry was drawn off the top of the tank, and pumped through the 'dead-headed' jet mixer positioned along the bottom of the tank. Slurry was then forced through the holes, causing a vortex flow pattern in the tank, as shown in Figure II-4. The recirculation flow rate, hole number and hole size were designed so that the jet velocity of the slurry exiting the holes was approximately 20 feet per second. In addition, the recirculation rate was sized to provide a tank 'turn-over' of at least 3 volumes per hour.

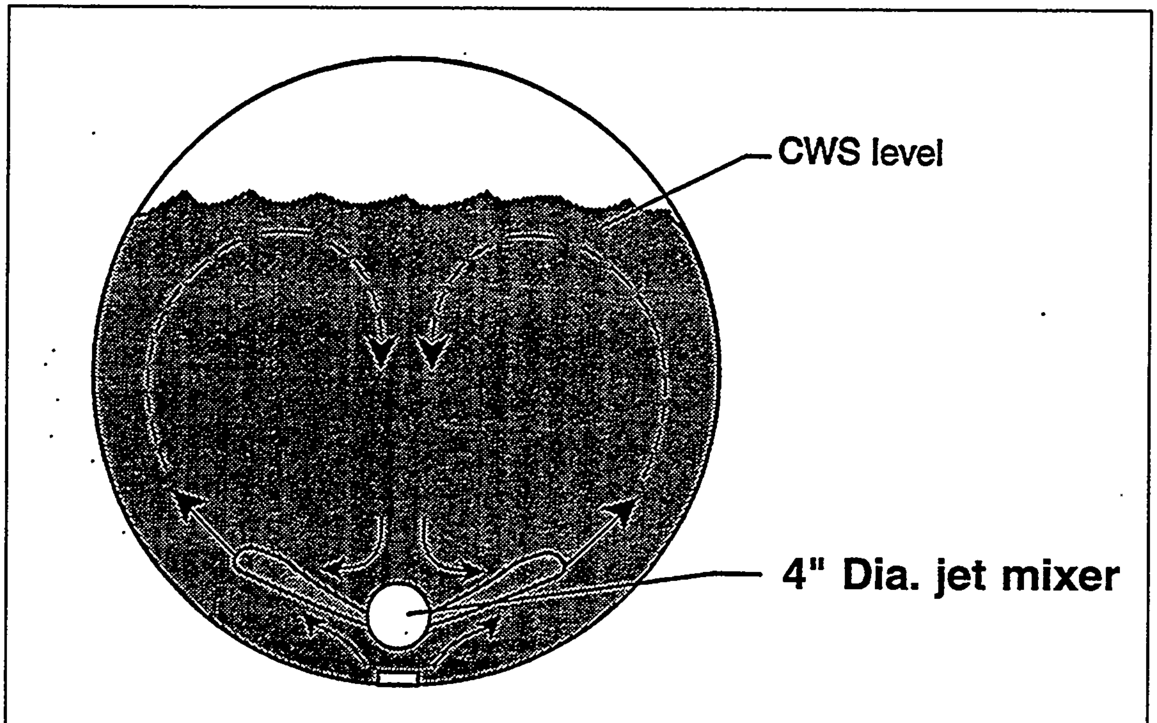


Figure II-4. CWS Storage and Handling System--Jet Mixer Operation

Pumps

The slurry was recirculated through the jet mixer with a positive displacement, vane-type Blackmer pump rated at 325 gpm at 25 psi head. The pump seals were modified to reduce the chance that CWS could harden in the seal area and/or penetrate into the bearing area. These modifications utilized a brief water flush interval to ensure a very low solids loading of any slurry left standing in small passages after the pump had been shut off. The water flush was automated by the CWS tank controls system.

The slurry was delivered to the day tank through a Blackmer vane-type pump rated at 30 gpm mounted so that in case of a pump failure, a fully functional pump would be immediately available.

2. Engine 'Day Tank' Design and Operation

The refill system for the engine tay tank is shown in Figure II-5. The slurry was continuously circulated through the transfer loop to avoid plugging in the lines. Solenoid operated valves opened to fill the mixed, 100-gallon day tank when its level dropped below a set point. Slurry consumption was monitored by the weight scale.

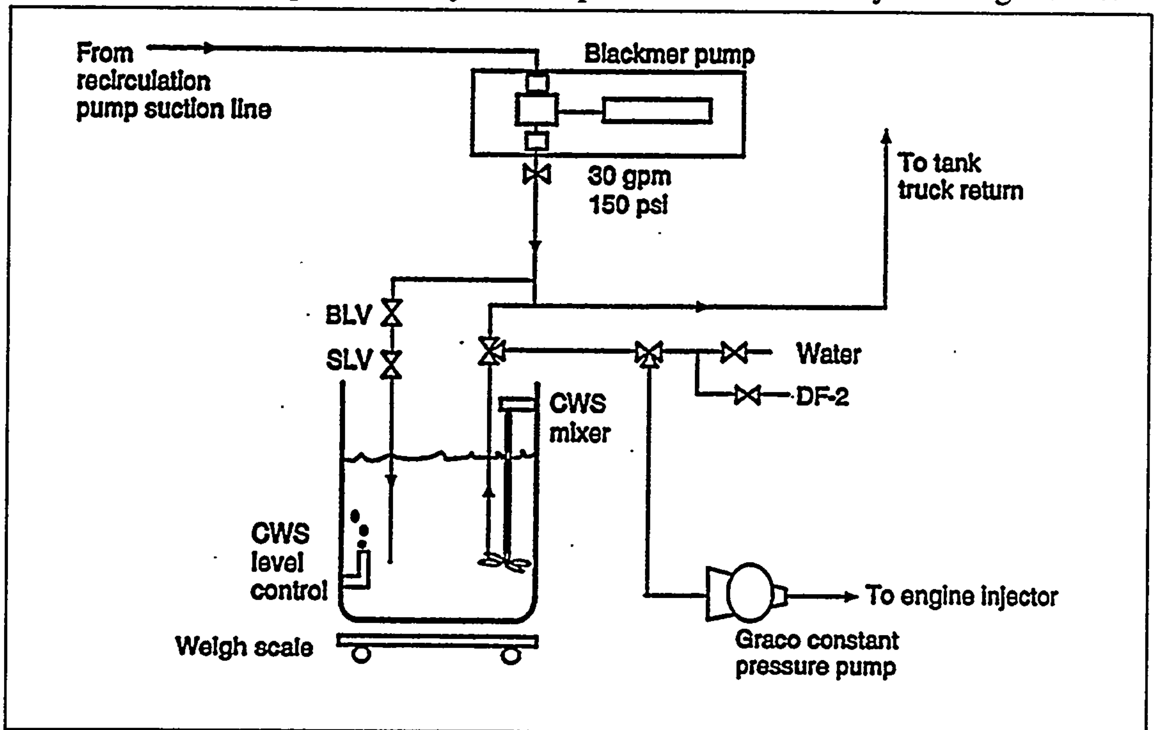


Figure II-5. CWS Transfer Loop from Storage to Day Tank

System Controls

An Allen-Bradley programmable controller was the core of the automated system controls. This controller provided the timing for tank occulation, pump flushing, day-tank refilling. It also controlled safety shut offs for the pumps, and motors, as well as temperature in the storage tank.

Photographs of the system are shown in Figures II-6 through II-14.

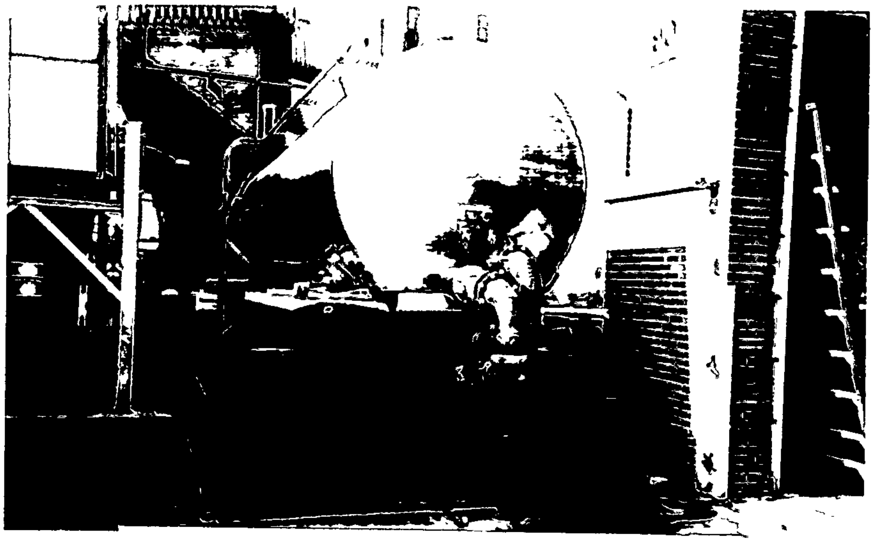


Figure II-6. 6,500 Gal CWS Storage/Mixing Tank

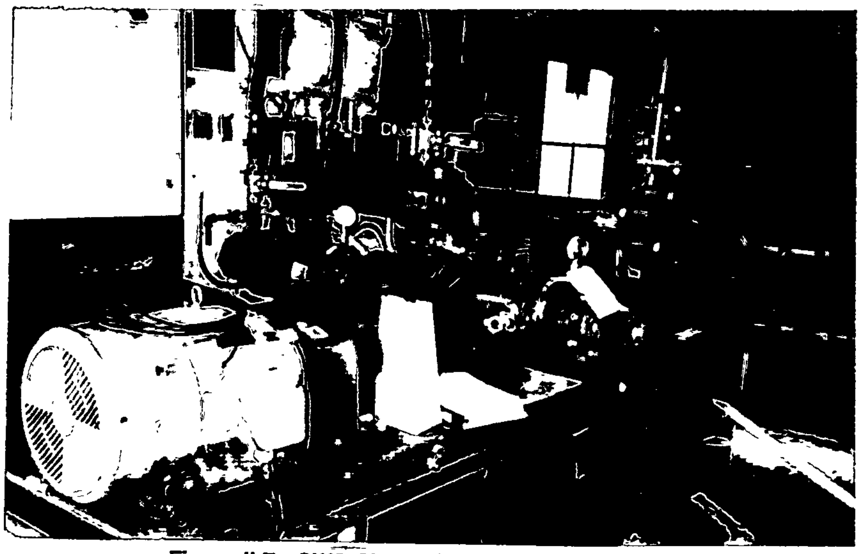


Figure II-7. CWS Circulation Pump (5" Hose, 15 Hp)

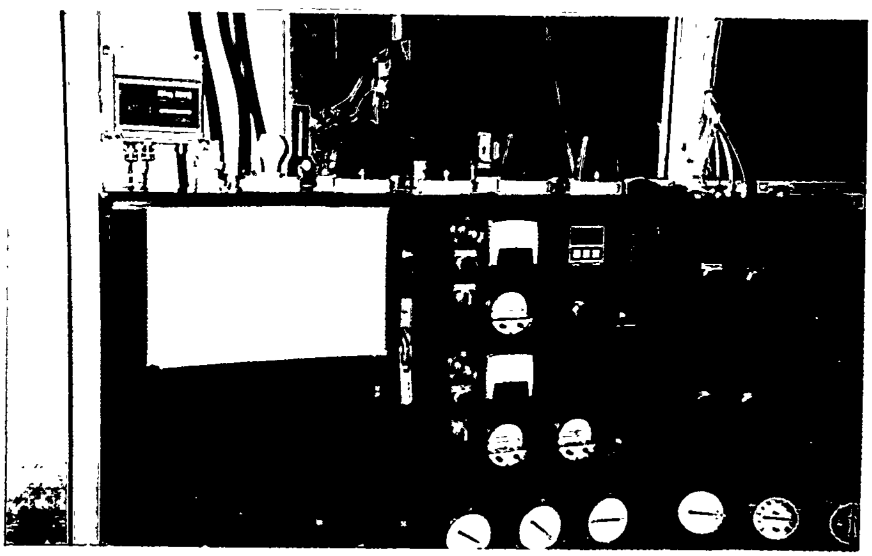


Figure II-8. Control Panel of CWS Handling System

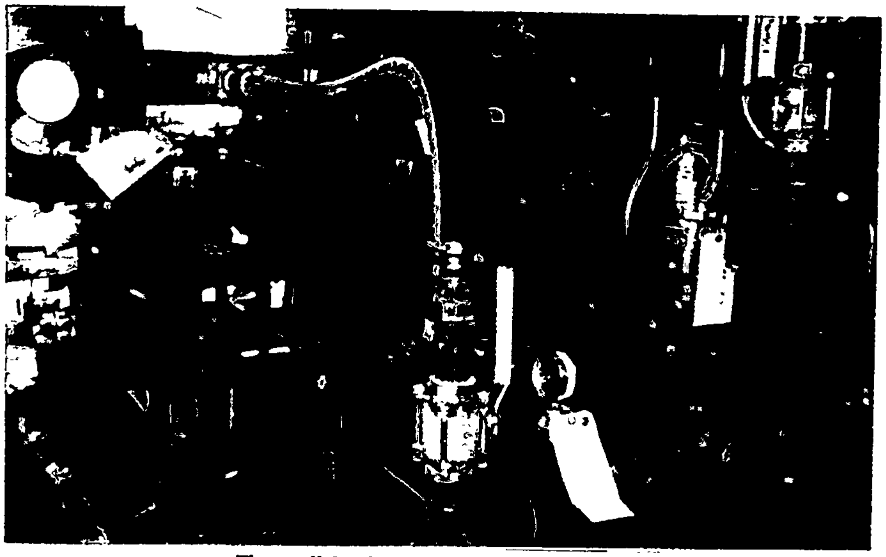


Figure II-9. CWS Control System

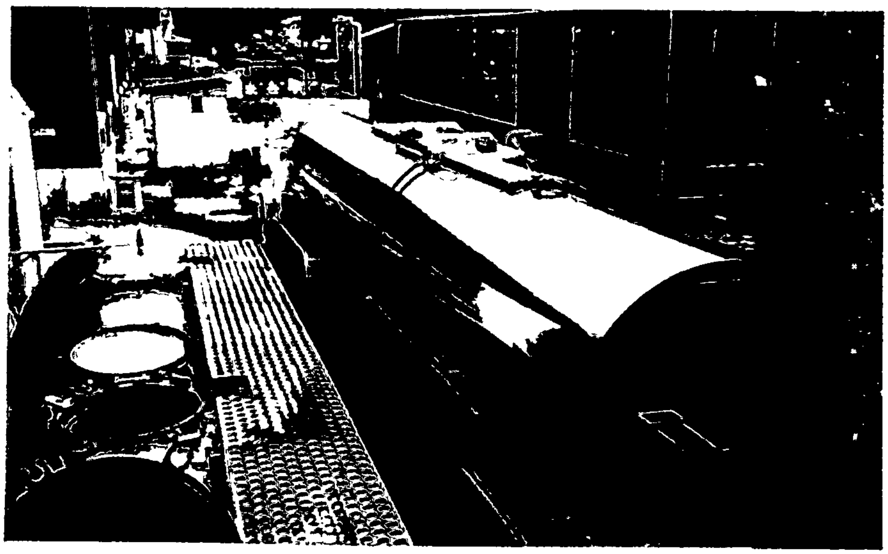


Figure II-10. Delivery of CWS from Tank Truck (on right) to Storage Tank (on left)

Figure II-11. Delivery of CWS from Tank Truck (on Left) to Storage Tank (on right)

Figure II-12. Suction and Discharge Lines

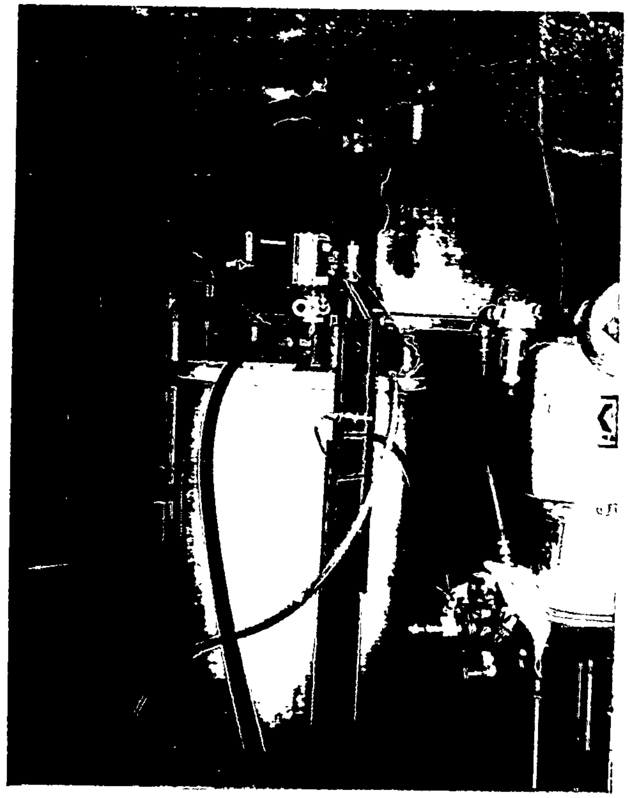


Figure II-13. 100 Gal Day Tank for CWS

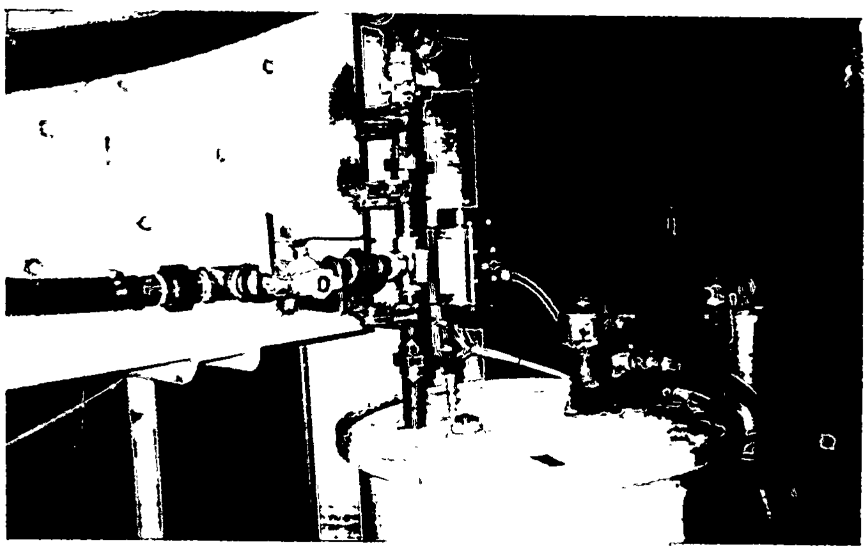


Figure II-14. 3-Way Valve with CWS Supply/Return Lines

Operating Characteristics

CQ slurry was stored in the tank for eleven months without significant variation in solids content. The tank was recirculated for one hour at intervals of eight hours (7 hours off/1 hour on). A graph showing solids concentration over three months of tank operation is shown in Figure II-15. After about 11 months of operation, the hose connected to the CWS inlet float sank, causing a disruption in the flow pattern in the tank. During this disruption, CWS solids content dropped from 49% to 46% over a three month period.

3. CWS Storage in Vertical, Stirred Mixing Tanks (CQ Facility)

CQ Inc. designed, fabricated and operated two, 5,000 gallon vertical CWS storage tanks that were stirred with low shear (60 RPM) mixers. As shown in Figure II-16, the tanks are 8 ft. in diameter and 15 ft. 6 in. high. Each mixer was powered by a 1½ hp motor. The bottom of the tank was sloped to allow for complete drainage and clean-out.

In practice, these tanks held the slurry at constant solids content over long periods of time (from one to twelve months, depending on engine test schedule). A mound of coal formed at the bottom of the tank, up to 9 inches thick, defining a quiescent region of the tank. This mound stayed at the bottom of the tank and did not interfere with mixing effectiveness. However, the slurry was drawn out of the top of the tank instead of the bottom of the tank because the settled coal blocked the tank outlet pipe at the bottom of the tank. Later, compressed air lines were added to clear the bottom discharges of blockage and thereafter the tanks were emptied from the bottom.

This low shear, stirred vertical tank is an alternative to the jet-mixer system used in the horizontal storage tank used at Cooper. The stirrers were eventually left on continuously; they consumed only 20 kWh of electricity per day and required only routine lubrication of gear reducers.

4. CWS Transfer and Transportation Logistics

During the 100-hour engine test, CWS was transferred from CQ Inc. to Cooper-Bessemer in tanker trucks filled with 4,000 - 4,500 gallons of fuel. The shipments to Cooper were timed to refill the storage tank at Cooper when there was a minimum of 2 hours operating time left in the tank. This corresponded to a quantity of 1000 - 1300 gallons of CWS in the tank (500 gallons of pumpable fuel, 500 - 800 gallons of non-recoverable fuel at the bottom of the tank).

Table II-4 shows the initial shipping schedule for the delivery of CWS during the 100-hour test in 1993. The final delivery times were determined on an hour-by-hour basis as the test progressed.

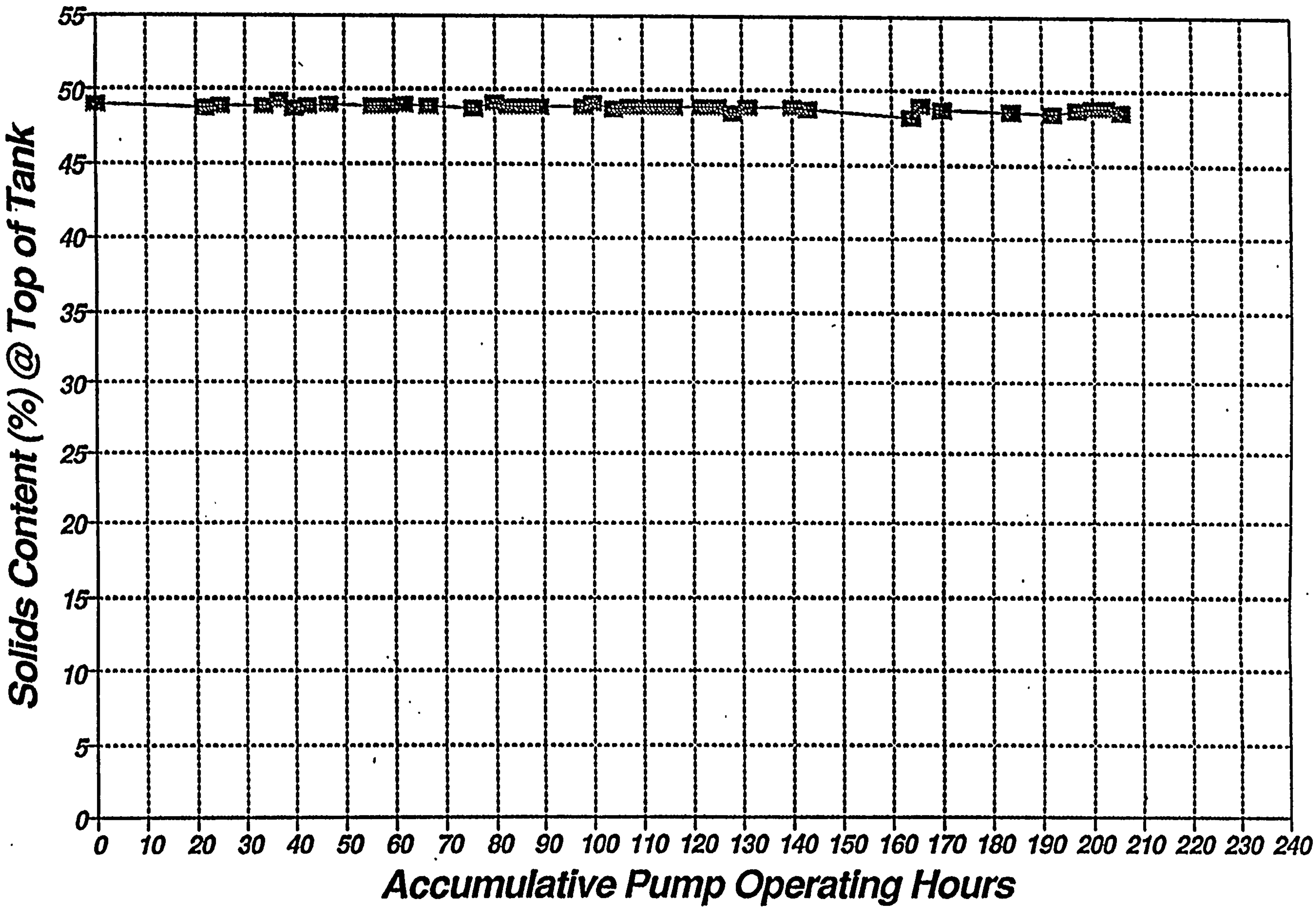


Figure II-15. CWS Solid % in 6,000 Tank (Mt. Vernon) - July 8 to Sept 7 (7 hr/off, 1 hr/on Cycle)

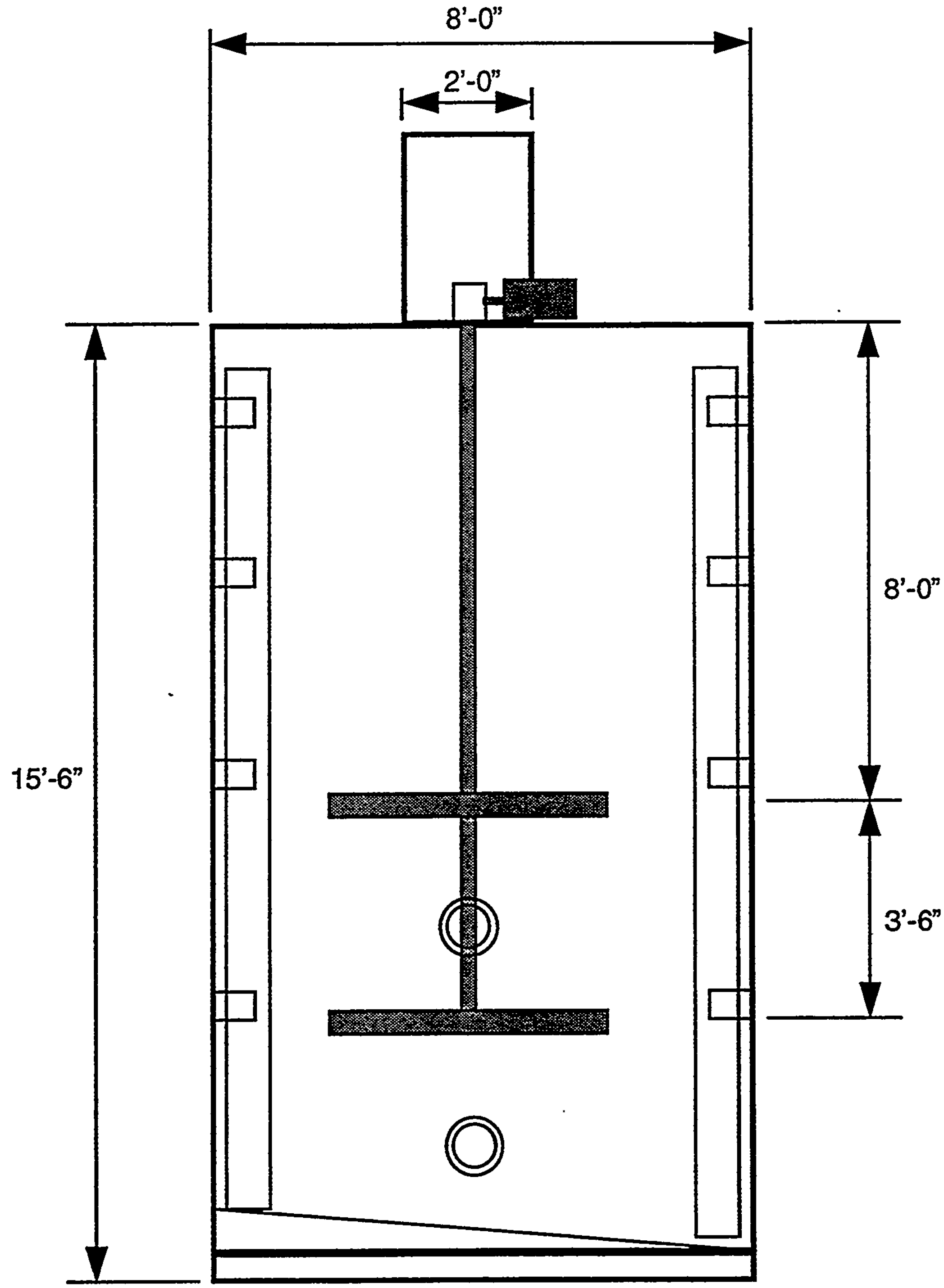


Figure II-16. Schematic of Homer City C.W.F. Storage Tank

Table II-4. Shipping Schedule, August 1993

Date and Time		Fuel Delivered at Cooper	Amount of Fuel Delivered	Amount of Fuel in Cooper Tank	Cumulative Amount of Fuel Used
6/23	6 am	0		5,200 gallons	0
	11 pm	17	Truck arrive	1,460 gallons	3,740 gallons
6/24	12 am	18	4,000 gallons	5,240 gallons	3,960 gallons
	5 pm	35	Truck arrive	1,500 gallons	7,700 gallons
	6 pm	36	4,500 gallons	5,780 gallons	7,920 gallons
6/25	1 pm	55	Truck arrive	1,600 gallons	12,100 gallons
	2 pm	56	4,500 gallons	5,880 gallons	12,300 gallons
6/26	9 am	75	Truck arrive	1,600 gallons	12,100 gallons
	2 pm	76	4,500 gallons	5,980 gallons	16,720 gallons
6/27	5 am	95	Truck arrive	1,800 gallons	20,900 gallons
	6 am	96	1,500 gallons	3,080 gallons	21,120 gallons

Notes: (1) Finished 100-hr test with #5 fuel delivery and leave 2,200 gallons in Cooper's tank

Loading and unloading the tanker truck proved to be straight-forward and reliable. Unloading at Cooper was accomplished by pressurizing the transfer tank with 15 psi compressed air from an on-board compressor (driven by the truck engine). Pressurizing the delivery tank took about 5 - 10 minutes. The slurry was transferred to the storage tank through a 6" diameter hose in 10 - 15 minutes. The entire transfer operation took about 20 minutes. The delivery truck was usually left with less than 50 gallons of residual fuel in the tank.

Figure II-17 shows the slurry being unloaded at Cooper from the transfer truck to the storage tank. Figure II-18 shows the slurry being loaded into the transfer truck from the storage tank at CQ. Inc.

CWS Storage and Handling-Lessons Learned

- **Left unattended and not agitated, all micronized CWS will eventually settle (in a matter of several months) and the separated solids will become hard packed.** Storage systems need to provide for regular mixing of the stored slurry. A jet mixing system can provide good mixing of the slurry and can maintain slurry solids content for periods up to 6 months. The jet mixing system can keep the slurry above freezing even in the coldest winter months. However, the jet mixing system includes a pump which does require considerable maintenance. A low cost, low maintenance solution is to equip storage tanks with low shear stirrers. These tanks would have to be in a heated location to prevent freezing, or would need to be partially under-ground.

- **If slurry will be fed directly to the engine and burned, the dispersant dose can be lower and there is no need for stabilizer.**

Figure II-17. Slurry Being Unloaded at Cooper-Bessemer

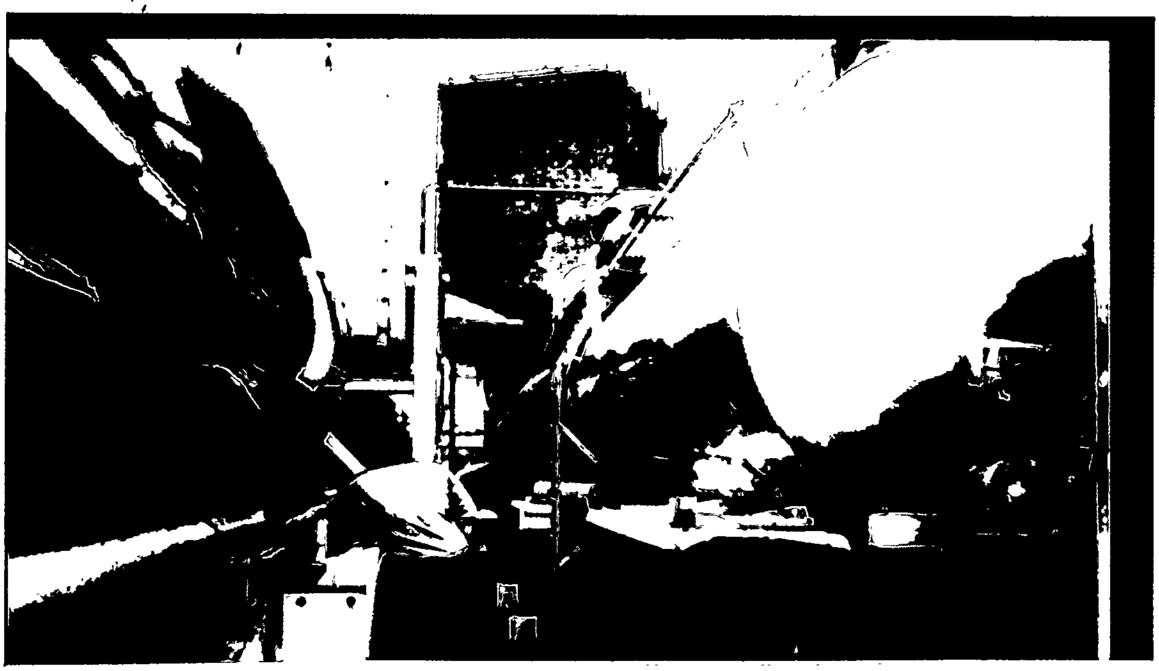


Figure II-18. Slurry Unloaded from Transfer Truck to Storage Trailer

C. Coal and Slurry Specifications Based on Test Results

The CWS properties that were examined for impact on coal-fired engine performance are: ash content, ash composition, particle size, solids loading, CWS formulation, sulfur and nitrogen content, and coal type. The basis of the determination of coal specifications was the tradeoff between the cost of fuel and the cost of engine modifications. For the properties listed, the tradeoffs are listed in the table below:

Table II-5. Cost Tradeoffs for Various CWS Properties

Ash Content and Composition	Coal Cleaning and Feedstock Cost vs. Durable Component Cost
Particle Size	Grinding Cost vs. Fuel Consumption and Durable Component Cost
Solids Loading	Additive Costs vs Fuel Consumption and NO_x emissions
CWS Formulation	Additive Cost vs. Storage and Handling System Costs
Sulfur and Nitrogen Content	Coal Feedstock, Cleaning Cost vs. Emission Control System Cost
Coal Type	Feedstock Cost (including transportation) vs. Fuel Consumption

The tests or analyses conducted to evaluate these tradeoffs are summarized in Table II-6 and discussed below.

Table II-6. Tests Conducted to Evaluate Cost Tradeoffs

CWS Property	Range Tested	Test Facility
Ash Content and Composition	0.5% - 3.8% 0.5% - 3.8% 0.5% - 22%	JS-1 engine AMBAC Injection Rig Sliding Wear Rig
Particle Size	3 - 15 micron mean 10 - 88 micron top	JS-1 engine
Solids Loading	42 - 53%	JS-1 engine
CWS Formulation	Triton X (0 - 2%) Dispersant Stabilizer (0.015 - 0.08%)	JS-1, LS-1, LS-6 engines JS-1 engine JS-1, LS-1, storage tanks
Sulfur and Nitrogen Content	No tests conducted	
Coal Type	High Volatile Bituminous Western Sub-bituminous Low Volatile Bituminous Appalachian Illinois Basin	JS-1 engine

1. Ash Content

A series of JS-1 engine tests were conducted in 1988 using AMAX prepared coals with (dry) ash contents of 0.5%, 1.2%, 1.5% and 3.8%. After each 2-hour engine test, the injection nozzle was inspected for wear. The nozzles used were AISI 8620 steel, carbonitrided multi-hole nozzles. Overall engine performance (pressure profiles, heat release, fuel consumption, and emissions) was also observed.

Slightly more nozzle wear was observed on runs with the 1.2-1.5% ash coals as compared to the 0.5% ash coal. Other engine performance was satisfactory. The differences in nozzle wear were not considered significant. The engine tests with 3.8% ash coal, however, indicated a significant fuel problem. The engine operated erratically and was unable to hold the set speed or load. Components upstream of the nozzle (the Moyno pump, pressure regulator, check valves, etc.) malfunctioned. Nozzle wear could not be compared to baseline because of the short duration of the test.

A second set of ash content tests were conducted in 1990 on fuels prepared by Energy International. The ash content of these fuels were 1.2% and 1.9%. No significant differences in engine performance or nozzle wear were observed over the duration of these tests (approximately 3 hours per tests).

Tests on the sliding wear rig and on the AMBAC injection rig showed that abrasive and erosive wear rate is proportional to ash content. Mineral matter composition did not appear to affect wear rate as significantly as total ash content.

It was decided that a baseline ash content of 2% (maximum) provided a reasonable tradeoff of coal cost and component wear. It is anticipated that this ash content could be increased if the projected wear rates of the advanced materials (sapphire nozzle tips, tungsten carbide rings) show sufficient operation life at 2% ash. The linear relationship between ash content and wear rate provides a starting point for future trade-off analyses.

2. Coal Particle Size

To test the impact of coal particle size on engine performance, AMAX prepared three batches on CWS from the same clean (0.5% ash) coal. The coal was cycled 2, 3, and 4 times through the ball mill to produce slurries with mean particle sizes of 12, 6, and 3 microns. No significant differences in engine performance (fuel consumption, peak pressure, location of peak pressure, etc.) were observed at the time.

A second set of particle size test was run with Energy International fuel in 1990. No degradation of engine performance was observed with particles sizes up to 15 micron mean, 80 micron top.

The particle size distribution in the engine-grade CWS will not likely be set by coal cleaning requirements, and coal combustion performance is apparently not a factor below about 80 micron top size. It is still important to screen out oversize particles and tramp material that could clog the nozzles. The limit of particle size that the engine can burn has not yet been found. Because of the significant cost of fine grinding and the associated additive costs required to formulate a fine particle slurry, it is worthwhile exploring this limit further.

3. Solids Loading

JS engine tests were conducted to investigate the effects of CWS solid content on engine performance. Of particular interest was:

1) Determining a reasonable CWS solids loading specification that takes into consideration both 1) potential high solids-loading penalties (e.g. injection problems, high viscosities and erratic engine performance) and 2) low solids-loading penalties (e.g. CWS instability, and poor engine BSFC).

2) Investigating the effect of solids loading on NO_x emissions.

Otisca slurry was diluted to solids content levels of 50%, 48%, 46%, 44% and 42%. Engine tests were conducted at four test conditions for each of these fuels. Fuel samples were frequently taken from the day tank and a 'tee' fitting close to the injector to evaluate CWS stability (settling behavior).

Key test results were as follows:

- Engine fuel consumption was fairly constant with solids content until the loading fell below 44% (see Figure II-19). The increase in BSFC at this point is probably due to the long injection duration at this low solids level.

- The CWS did not settle noticeably over the course of the runs, for any of the dilutions. (For the 50% solids slurry, solids loading fluctuated between 50.1 and 50.2 over a 2½ hour period). The solids content of CWS sampled at the day tank generally agreed with the solids content of CWS sampled at the injector to within 0.5%. There was no indication that there is a short-term handling 'penalty' associated with minor dilutions to the slurry. Potential problems with long term storage at lower solids loading were not investigated.

- Using fuel with 50% solids, fuel consumption at different engine test conditions varied by 500 Btu/bhp-hr (see Figure II-19). This variation was less than 100 Btu/bhp-hr at 46 and 48% loading. Previous tests had shown that engine

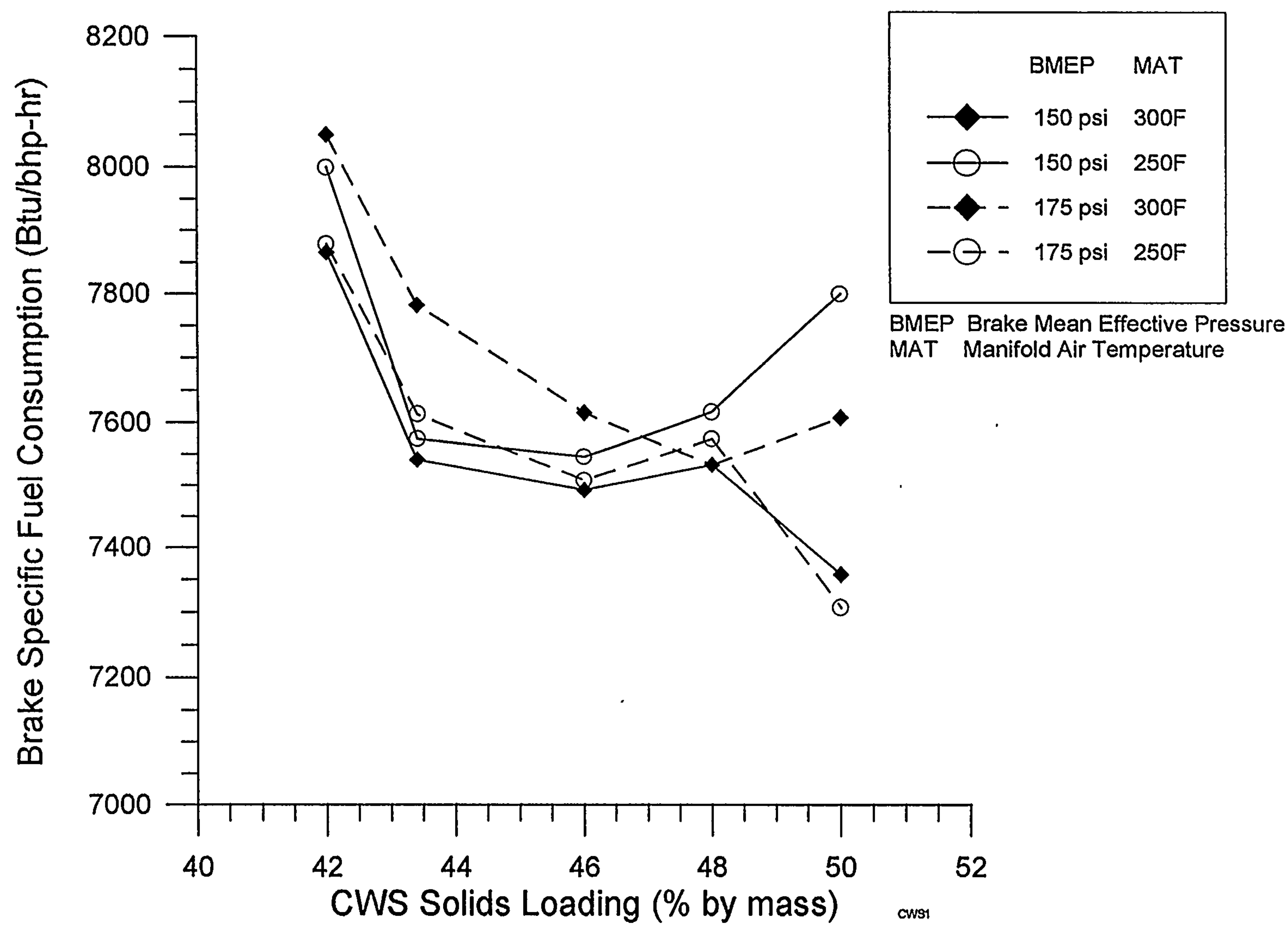


Figure II-19. Fuel Consumption vs. CWS Solids Content behavior was very erratic at 52% solids. Based on this data, 48% solids loading was chosen for the baseline Otisca fuel solids loading specification. The solids loading specification is closely tied to the particle size distribution of the slurry. Slurries produced by Energy International with a wider particle size distribution were loaded up to 54% solids without any injection or handling problems.

- NO_x emissions decreased with decreasing solids loading, as shown in Figure II-20. On average, NO_x emissions decreased 30-40% as solids loading was decreased from 50 to 42%.

4. CWS Formulation

Three additives are used in the formulation of the engine-grade slurries: dispersant (naphthalene ammonium sulphonate) to decrease viscosity, stabilizer (xanthin gum) to decrease settling, and oil-antiagglomerant (Triton-X) to prevent the coal from agglomerating when it contacts diesel oil under high shear conditions. Numerous tests were conducted to identify the preferred levels of these additives to use and the process condition to add them. It was found that dispersant is most effective if its addition is distributed over the slurry preparation process. To satisfy the requirements of grinding, storage, handling, and injection, dispersant was added at three points: upstream of the ball mill, down stream of the ball mill and at the discharge of the bead mill (the final grinding step). A total of 1% dispersant is effective at maintaining viscosity below 200 cP.

Stabilizer amounts were reduced to 0.015% by weight (from 0.03%) without decreasing the ability to store and re-suspend the slurry. If storage requirements were minimal (that is, less than a few days) stabilizer may not be required. The use of too much stabilizer can lead to the formation of a thick, viscous layer on the top of storage tanks. This layer is easily remixable, however.

Triton X is necessary if the slurry contacts diesel fuel in lines upstream of the injection system. The LS-6 engine was operated successfully for hours without any Triton-X added to the stream. However, if there is an emergency shutdown, and the engine operation is rapidly switched to diesel fuel, there is a risk that CWS could clog in the check-valves and/or nozzles. It may be most economical to provide the opportunity to add Triton X to the fuel system at a point near to the injection system so that it can be mixed with the slurry only when needed for shutdown or start-up on CWS.

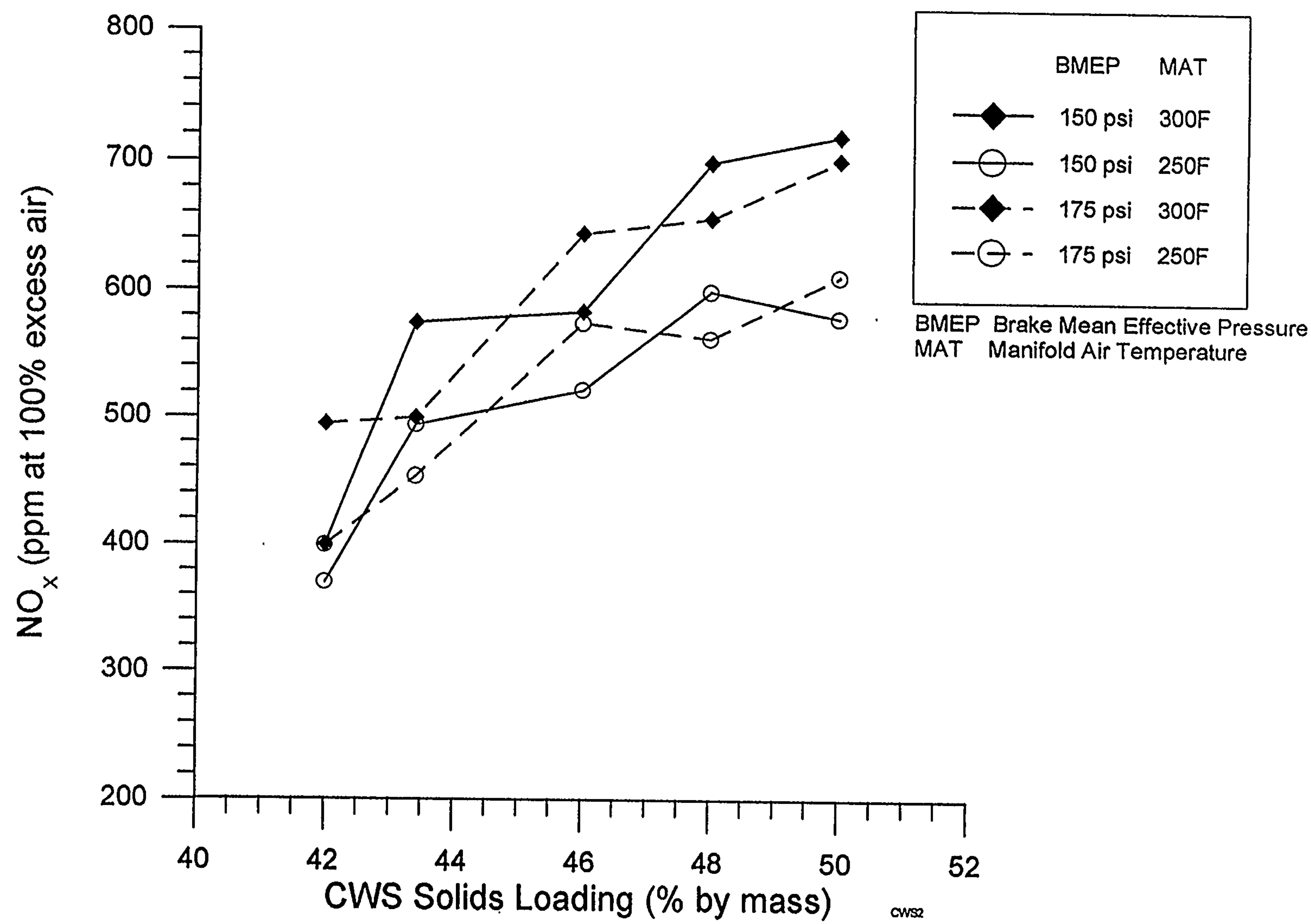


Figure II-20. NO_x Emission vs CWS Solids Content

5. Coal Type

The engine was very tolerant of changes in feedstock, as well as particle size. High volatile, eastern bituminous coal was used for almost all engine testing. Brief tests were conducted with a western bituminous, a western sub-bituminous, a low volatile eastern bituminous and an Illinois basin bituminous coal. All except the lower volatile bituminous coal burned well, with acceptable peak cylinder pressure, heat release rate and overall combustion efficiency. The lower volatile coal tested, Upper Freeport, (29% volatiles) is a sticky coal with a high swelling index. It is not known if the combustion problems on this coal were due to the low volatile content or the tendency of the coal to agglomerate.

Limits to the use of various coal types will be set by their cleanability. This issue was explored in the feedstock survey, and is discussed in Section II-D.

6. CWS Specifications-Lessons Learned

- **The 2% ash content specification is a reasonable balance point between CWS cost and engine component cost.** However, this ash content threshold should be re-evaluated as the wear resistance of hard components is established. Increasing the ash content specification can significantly reduce slurry cost.

- **The coal-diesel is very tolerant of increases in top particle size (up to 88 microns).** No slurry was tested that resulted in a decrease in engine efficiency because of excessively large particle sizes. This is another area that should be examined for relaxing the specification further to decrease in fuel processing cost.

- **The coal diesel engine is very tolerant of changes in coal type.** Based on test results, a wide range of coal feedstocks appear to be acceptable for engine use. Coal cleanability is likely a more limiting factor than engine performance.

- **The engine was not sensitive to amount of dispersant used in the slurry and the point of the slurrying process at which it was added.** The engine operated well without Triton X and with limited amounts of stabilizer.

D. Conceptual Commercial Plant Design

1. Potential Coal Feedstocks

The most important criteria for coal feedstock selection is coal cleanability. CWS ash contents of less than 2% appear to be necessary because of engine component wear caused by abrasive constituents in the coal (and/or the combusted particulate matter). Therefore, a coal feedstock survey was conducted to identify coals which can be cleaned to acceptable mineral matter concentrations utilizing currently known coal cleaning technologies. Other feedstock selection criteria, such as available reserves, mine size, seam location, and sulfur content were also considered.

The overall plan for identifying coal feedstock suitable for diesel engine applications was to:

(1) Identify potential sources of U.S. coal

(2) Eliminate unsatisfactory candidates, based upon a prioritized list of selection criteria.

(3) Select suitable candidates from different geographical locations for beneficiation testing.

(4) Identify availability and cost information for each candidate.

The portfolio of selected coals was intended to be representative of coal sources throughout the U.S. It was not intended to be comprehensive. The prioritized list of selection criteria are listed below.

(1) Attractive washability, indicating sufficient ash reduction may be attainable with fine coal beneficiation technologies.

(2) Demonstrated sufficient coal reserves (greater than 20 MM tons).

(3) Larger mines (greater than 100,000 tons per year).

(4) Location close to intended market.

(5) Coals with a lower organic sulfur content (less than 1.0 weight percent).

Figure II-21 shows an overlay of the locations of potential coal feedstock mines and the regions identified for potential coal diesel applications. Many states targeted for coal-diesel applications contain high grade coals that potentially could be suitable as engine-grade CWS feedstocks. These states include: Kentucky, Virginia, West Virginia, Pennsylvania and Ohio. The paragraphs below describe in general terms the types of coals available in these states.

Virginia

Coals mined in the Southwestern Virginia Coal Field are generally very high quality, low sulfur coals. Washability data shows seams in the Wise Formation such as Taggart, Dorchester, and Kelly can be cleaned to ash and sulfur levels suitable for diesel engine applications, often with only standard heavy media washing. The coal

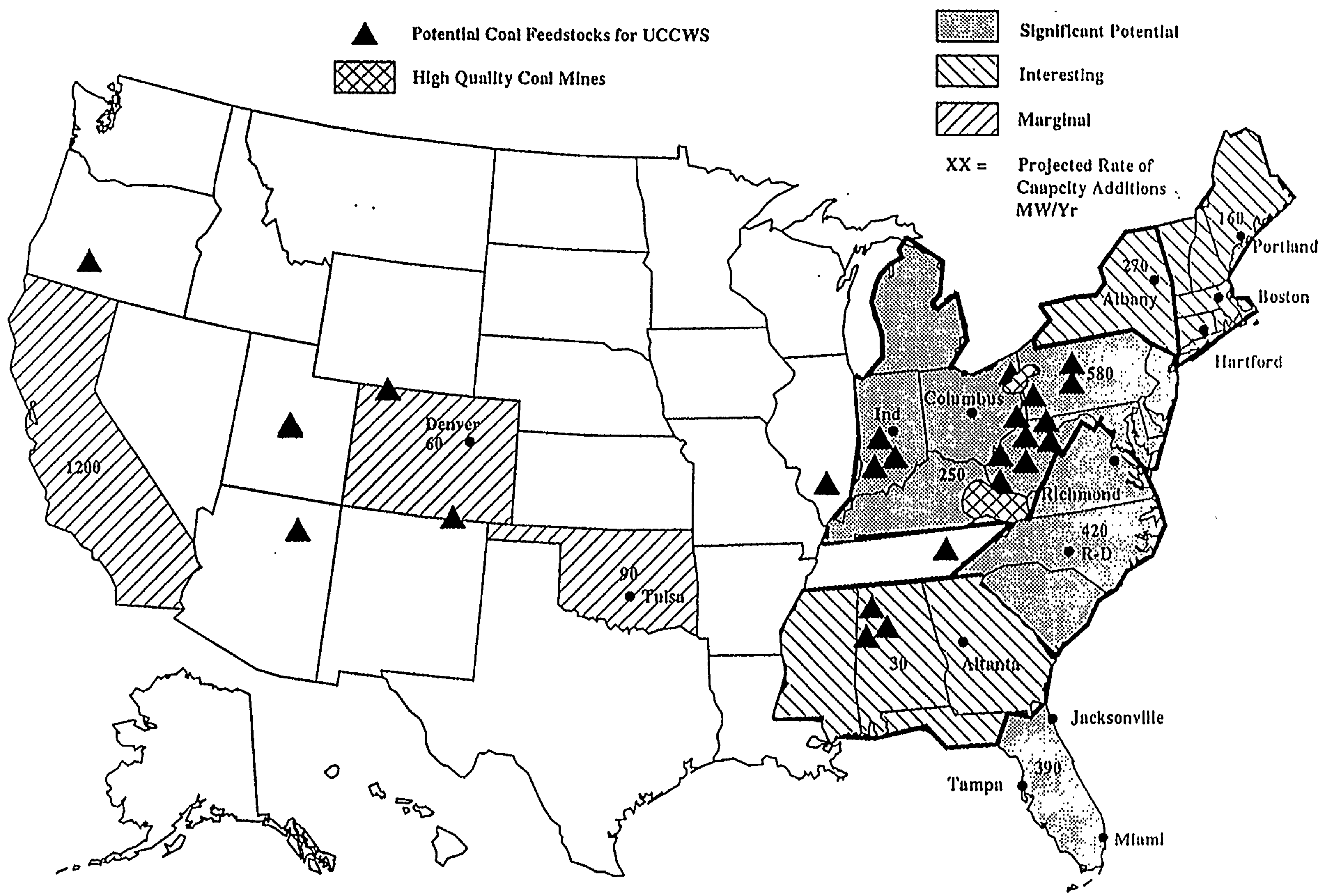


Figure II-21. Locations of Potential Coal Feedstocks quality in this formation is similar to that found in the Upper Cumberland District of Kentucky and the Pottsville Group in the Southern Coal Field of West Virginia. The three coal producing districts are adjacent to each other and have arbitrarily been divided by state lines.

West Virginia

There are two distinct coal regions in West Virginia: a North Coal Field containing thin seams of generally medium to high sulfur (1.5 to over 3.0 weight percent) and medium to high ash (higher than 6.0 weight percent), and a southern coal field containing thick seams of coal uniformly low in sulfur (less than 1.5 weight percent) and ash (less than 6.0 weight percent). The two coal fields are separated by a distinct "hinge line" which runs northeast to southwest through the middle of the state.

West Virginia ranks third in coal production behind Wyoming and Kentucky. Vast reserves of coals in the Pottsville Group and the Pocahontas Formation are similar to coals found in the Upper Cumberland District of Kentucky and the Wise Formation of Virginia. Pocahontas No. 3 is a good example of a coal with large minable reserves (2.8 billion tons), low as-received ash (average 4.0 percent), and shows good fine coal washability (ash reduction to 1.5 weight percent).

Kentucky

The state of Kentucky has two distinctly different coal fields separated by a non-coal bearing portion of the state. The Western Coal Field belongs to the Interior Coal Province (the Illinois Basin). This vast resource contains about 38 billion tons of medium to high sulfur coal (1.5-5.0 weight percent).

The Eastern Coal Field covers about twice the area of the Western Field and contains reserves of roughly 55 billion tons. The region is divided into six coal producing districts: the Princess District, the Licking River, the Bid Sandy, the Hazards, the Southwestern and the Upper Cumberland. Of the six districts, the coals of interest for diesel engine applications may come from the Upper Cumberland, Bid Sandy and Hazard districts, in that order. Some seams of interest with large reserves are: Hence, Harlan, Upper Elkhorn Nos. 1, 2, and 3, Lower Elkhorn and Blue Gem.

The coals found in the Upper Cumberland District are of the same high quality (high volatile, low ask, low sulfur, amenable to beneficiation) as those found in the Southwestern Virginia Coal Fields and the Southern Coal Fields of West Virginia. That is, these three coal fields are members of the same coal field which has been arbitrarily divided by state lines.

Ohio

Most Ohio coals are classified as either medium or high sulfur coals (1.5-6.0 weight percent). The relatively small reserves of low sulfur coal, such as Lower Freeport (Ohio 6A) have been depleted. The highest quality coal with any significant reserves is Lower Kittaning in Mahoning or Holmes County. Ash reduction suitable for diesel applications using physical cleaning methods is possible with sulfur in the 0.5-2.0 weight percent range.

Pennsylvania

A coal quality trend in Pennsylvania is that sulfur content increases northward and westward. Many of the familiar Pennsylvania seams (Brookville, Upper, Middle, and Lower Kittaning, Upper and Lower Freeport, Pittsburgh) are also mined in Ohio and West Virginia. These seams are extensive in the area they cover and have significantly different quality depending upon where it is mined. Parts of the Middle Kittaning seam wash very easily to about 1.0-1.5 weight percent ash and 0.5-0.7 weight percent sulfur while other locations can achieve no better (using the same cleaning technique) than 4.0 percent ash and 2.0 percent sulfur. Essentially the same thing can be said of any of the other seams mentioned above. It is quite possible for Pennsylvania coals to be used for diesel engine applications, but care should be taken in choosing the mine from which the coal will be purchased.

2. Plant Design for Premium Coal Cleaning

Two different coal cleaning and slurry preparation plant scenarios have been examined: a single and a multi-product coal cleaning plant. Both these approaches to producing engine-grade slurry are feasible, however, the multi-product approach has less technical risk and allows for a much broader selection of coal sources.

Single Product Plant

The first scenario is that of a one-product plant that produces engine grade coal-water slurry only. It was assumed that this plant would provide enough engine-grade CWS for up to 500 MW of power generation capacity. Five cleaning options were investigated for this plant scenario: heavy media separation, oil-agglomeration, coarse flotation, fine flotation and chemical cleaning. For a one product plant, the key plant characteristics to optimize are coal recovery (cleaning efficiency) versus plant capital cost. The preferred plant design depends on the coal cost, coal transportation cost, coal cleanability and interest rates.

A spreadsheet-based cost model was developed to examine process plant cost tradeoffs and sensitivities. This model and the results of the analyses conducted have been discussed in other reports (Benedek, et al., 1990, Benedek and Wilson, 1990).

The results of the single-product plant studies indicated that slurry could be produced eventually at costs between \$3.00 and \$3.50 per million Btu, but that the costs of slurry from the first plant could be higher.

Multi-Product Plant

A second, engine-grade, coal slurry preparation option was examined. In this scenario, shown in Figure II-22, the engine grade coal slurry is produced as a by-product of a commercial coal cleaning plant that produces cleaned coal for boiler use. A multi-product plant may produce a small output of engine grade fuel (approximately 10% or less of the total plant output.) The premium coal product from the plant is shipped to a slurrying facility located in the same region as several coal-diesel power plants. The slurry is then shipped by tank truck to the power plant.

As illustrated in Figure II-23, premium coal is extracted as a by-product of the existing coal cleaning operations using conventional coal cleaning technology such as heavy media cyclones. The specifics of the approach are as follows:

- A heavy media cyclone circuit is added to an existing coal cleaning facility. Some of the clean coal stream, (10 - 40%) exiting the existing coal cleaning operations, is fed to the new heavy media circuit.

- A premium coal product is extracted at low recovery efficiencies (10-40%) from the heavy media cyclones. Reject coal from these cyclones is fed back to the boiler grade fuel stream. Premium coal is dried in a centrifuge and stored separately.

- Premium coal is shipped to within 25 miles of the engine site for slurry preparation. This minimizes the transportation cost and additive requirements for the slurry.

This approach is low risk because of its use of conventional technology and its relative insensitivity to premium coal recovery efficiency. Because the middlings product is mixed back into the boiler grade coal stream, there are no significant cost penalties associated with 'skimming' coal product from the boiler grade coal. A final benefit of this approach is that it can be initiated more easily than a single product plant. In this way, the coal diesel market and the premium coal market can grow together. As the demand for premium coal grows, the percentage of boiler grade coal diverted to the premium coal circuit can increase.

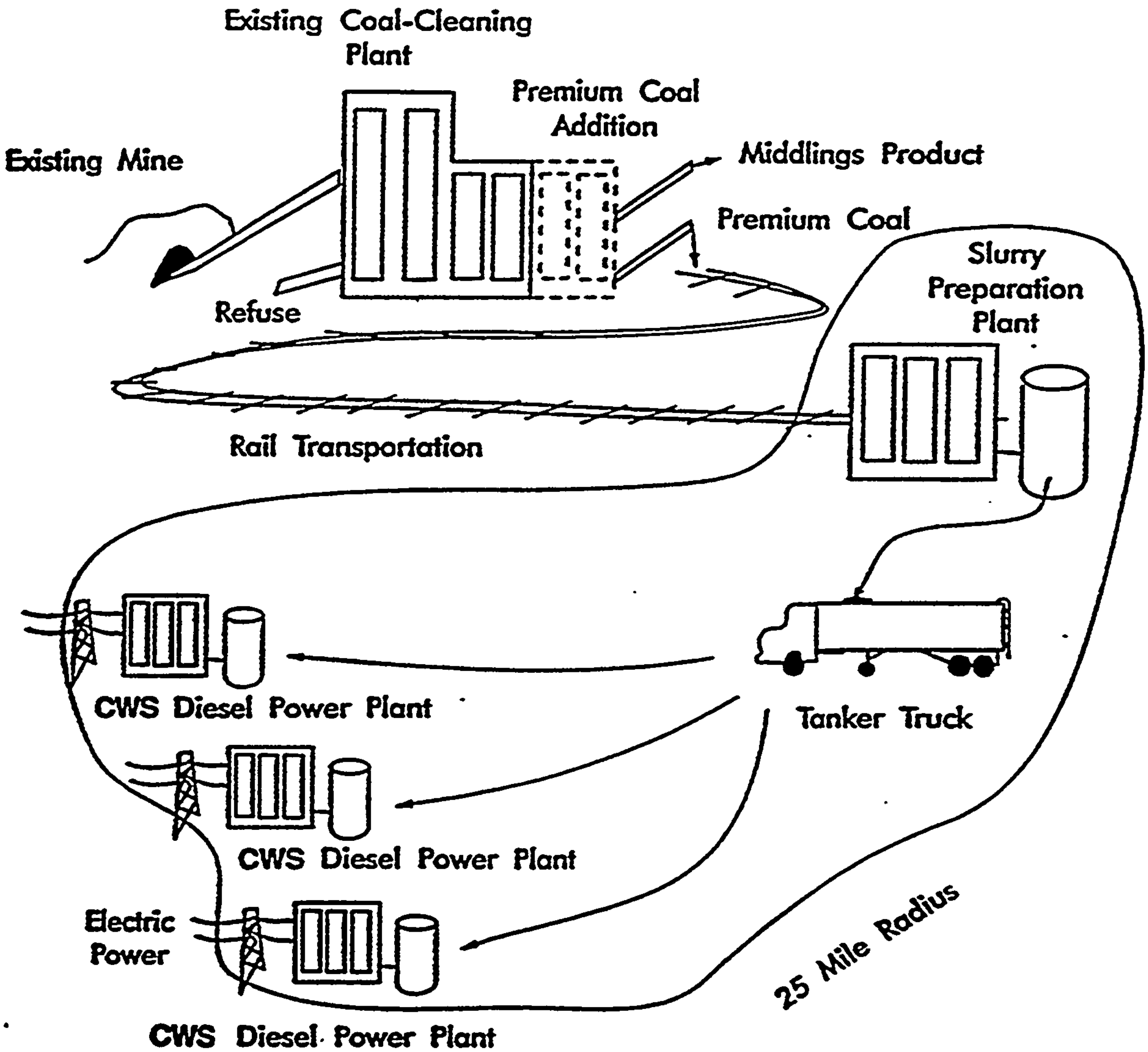


Figure II-22. Engine-Grade, Coal-Fuel Preparation Strategy

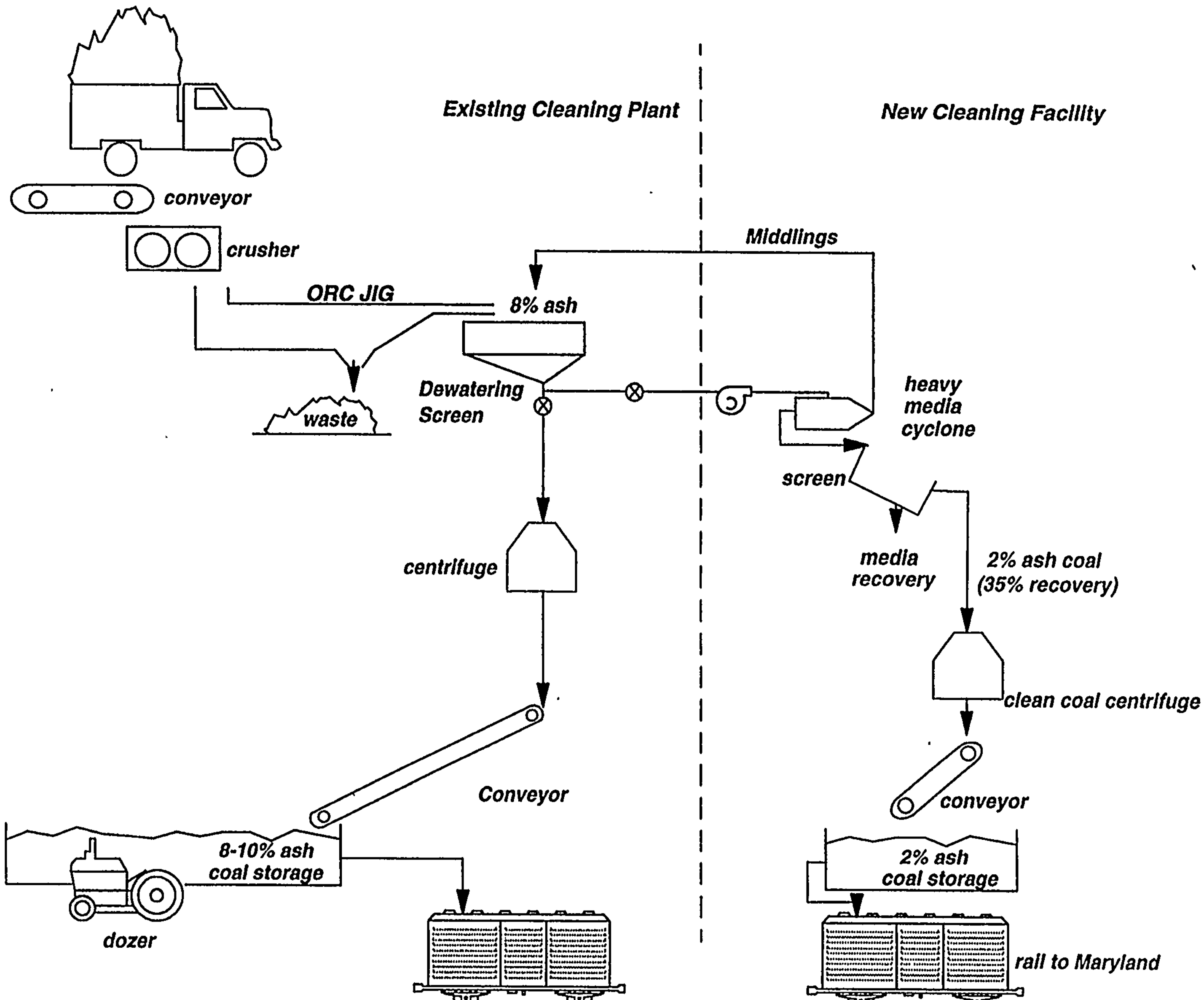


Figure II-23. Production of Engine-Grade Coal as By-Product of Boiler-Grade Coal Production

3. CWS Plant Economics and Projected Costs

Table II-7 shows the projected engine grade CWS price assuming a multi-product plant approach. This analysis shows that slurry can be delivered to the engine plant at under $3.00 per million Btu. Operating and maintenance cost breakdown for the slurry production is shown in Table II-8. Over 40% of the cost of the slurry goes to the additives. Much of the remaining costs (electricity, grinding media, maintenance) is associated with fine coal grinding. Capital cost assumptions for a 250 tons per hour coal cleaning plant addition are shown in Table II-9. The coal slurrying plant is far more expensive than the coal cleaning facility.

Table II-7: Projected CWS Price for Regional Production

Cost Component	Cost ($/MMBtu)	% of Total CWS Cost
Feed coal[1]	0.888	29.9
Coal cleaning		
O&M	0.188	6.3
Capital recovery[2]	0.112	3.8
Subtotal	0.301	10.1
Coal transportation	0.264	8.9
Coal slurrying		
O&M	0.513	17.3
Capital recovery[2]	0.542	18.3
Subtotal	1.055	35.6
CWS transportation	0.461	15.5
Total	2.979	100.0

1 Pre-cleaned feed coal is $24/ton. It is re-cleaned producing middlings that are sold for $22.98/ton and premium coal with a feed cost equivalent to $25.25/ton.

2 Capital recovery includes 112.5% cost of money.

Table II-8: Operating and Maintenance Cost Breakdown for CWS Production

Cost Component	Cost ($/ton Coal Feed)	% of Total O&M Cost
Labor	3.39	23.2
Electricity (78.8 kWh/ton)	3.15	21.6
Dispersant	4.91	33.7
Stabilizer	1.28	8.8
General maintenance	0.81	5.6
Grinding media and liners	1.05	7.2
Total	14.59	100.0

Table II-9: Capital Costs Associated with CWS Production

Component	Value
Coal cleaning plant addition (250 tph)	$7.7 million
Coal slurrying plant (100 tph)	$35.1 million
Financing term	20 years
50% debt	10.0% interest
50% equity	15.0% (after tax)

E. Conclusions and Recommendations

The objectives of the coal-fuels development task were to develop appropriate specifications for engine-grade CWS and to develop a CWS production scenario that could provide cost effective fuel for engine development and, ultimately, for the commercialization of the coal-diesel engine technology. The conclusions of the work conducted in these areas, and the recommendations for future development and demonstration efforts and for commercial plant design are listed below:

- **Ash content.** A coal ash content of 2% can be cost-effectively achieved with conventional technology in a multi-product, coal cleaning plant. It is recommended that this coal specification be applied to future efforts in coal-engine technology. If significant gains are made in engine durability such that a higher ash content in the fuel could be tolerated, the ash content specification should be re-examined with the aim of increasing the specification to the range of 3-5% ash. Ash content is not the most critical determinant of engine-grade CWS cost.

- **Coal particle size.** A top coal size of 80 microns with a mean of 12 to 15 microns has been successfully used in the coal diesel engine and can be achieved with a combination of crushing, ball milling and stirred ball milling. The grinding process is a significant element of the CWS cost and more effort should be expended to determine if this specification can be relaxed. A coal slurry with a larger top size would also require less additives. In future tests, the impact on the engine of increasing both the coal top size (to 120 - 150 microns) and the mean particle size (to 25 microns) should be examined. At this particle size distribution, it is possible that the coal could be prepared without expensive stirred ball milling.

- **Additive package.** An additive package has been developed that allows the slurry to be stored for long periods of time (up to 1 year) , transported over long distances (200 miles) and pumped or injected like diesel fuel. These additives contribute to almost 40% of the cost of the CWS (for a commercial product). A coal cleaning and slurrying scenario has been developed that reduced both the required storage time and the required transportation distance of the slurry. In future efforts, the suitability of a reduced additive package should be tested. This should be tested in conjunction with an increase in the coal particle size.

- **CWS storage.** Coal slurry can be stored with minimal energy and maintenance requirements in continuously stirred, vertical tanks. This configuration should be considered for future efforts.

- **CWS preparation.** A multi-product, coal cleaning plant can produce 2% ash coal at acceptable cost. A coal production technique that produces engine-grade coal as a by-product of boiler-coal production minimizes the importance of coal cleaning efficiency and increases the set of potential feedstocks for the process compared to a single-product plant design. It is anticipated that engine-grade, coal water slurry can be produced for less than $3.00/MMBtu.

III. Engine Tests and Performance Results

A. Overview

Significant progress has been made in developing components for the coal-fueled diesel engine. A total of more than 1050 hours of engine operation using coal-water slurry (CWS) was logged on Cooper-Bessemer engines as part of this program. Over 560 hours of CWS testing was accumulated on a single cylinder research engine (JS-1) in order to develop combustion system configurations, fuel specifications, and durable components. Then, in 1991-1992, an additional 125 hours of CWS testing was accumulated on the full-scale LSC-1 engine (one cylinder on coal). During the last two years of the program (1992-1993), the LSC-6 engine (all six cylinders on coal) was successfully run on CWS under full speed and full load conditions. In addition, the LSC-6 engine successfully achieved the 100 hour proof-of-concept "endurance" run on the first attempt at the conclusion of this program. These engine tests have provided valuable insights into coal-fueled diesel combustion phenomena and engine component design requirements. Table III-1 summarizes the key development efforts for the three test engines and shows the progression from subscale tests to the commercial scale. Figure III-1 illustrates the history of Cooper-Bessemer test experience from 1987 to date. Advances in the durability of critical components such as the nozzle tip and piston rings have enabled Cooper-Bessemer to accumulate hundreds of hours per year of engine test experience.

Table III-1. Coal-Fueled Diesel Engine Development

JS-1	13 inch bore 16 inch stroke single cylinder	**Exploratory CWS Tests** • Fuel handling • Injector system development • Durable components • Coal-fueled diesel combustion phenomenon
LSC-1	15.5 inch bore 22 inch stroke 1 cylinder CWS fuel 5 cylinders DF-2 fuel 436 bhp/cylinder	**Full-scale Development Tests** • Injection system • Emission controls • Combustion optimization
LSC-6	15.5 inch bore 22 inch stroke 6 cylinders CWS fuel 2,616 bhp (1800 kW)	**Proof-of-Concept Demonstration** • 6-cylinder fuel system • Additional combustion optimization • 100-hour proof-of-concept test

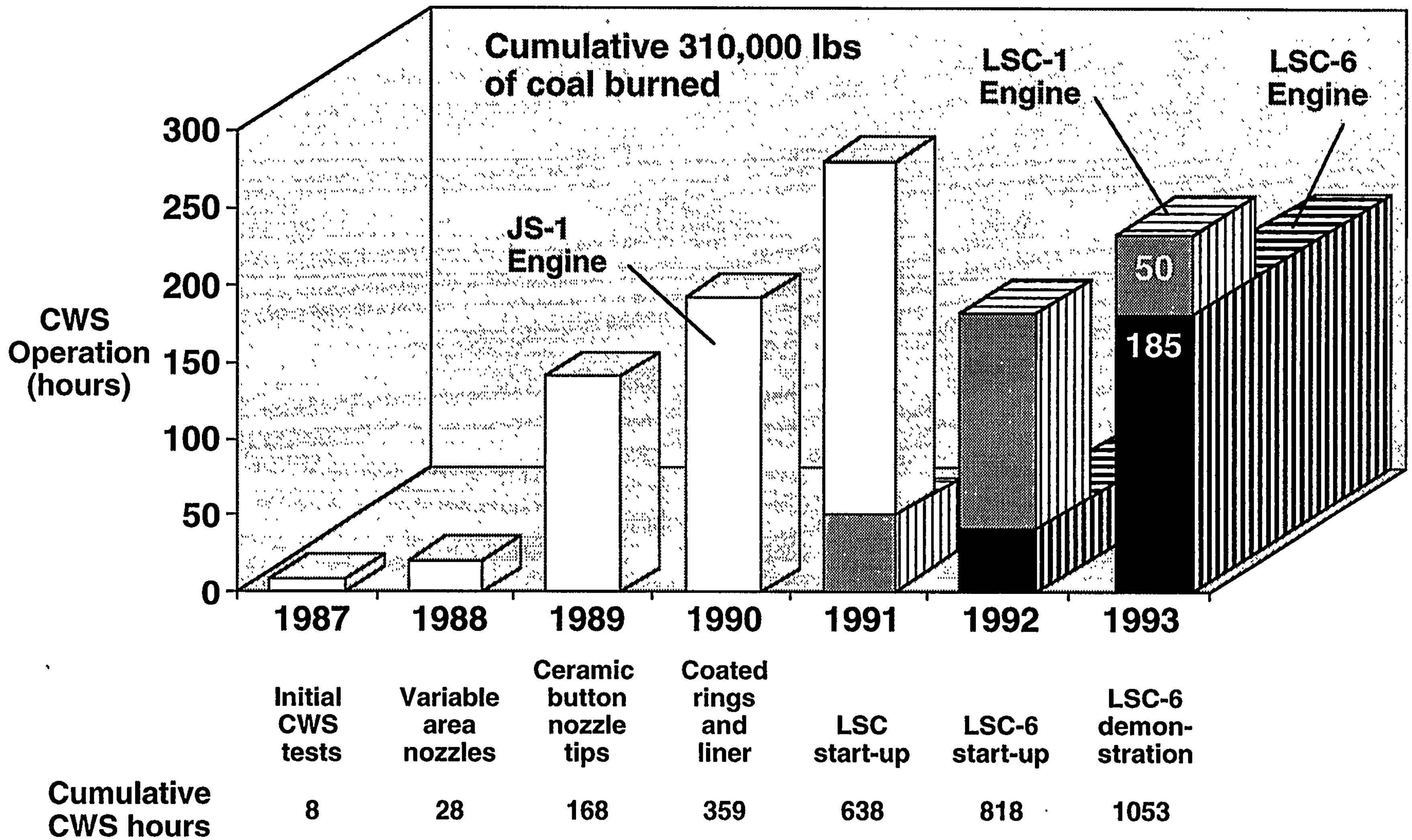


Figure III-1. History of Engine Test Experience

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The LSC coal engine combustion system (including injector, pilot, and chamber shape) has been developed with special emphasis on:

- CWS fuel spray development and fuel-air mixing;
- ignition and combustion of coal; and
- durable power cylinder components.

The current 1800 kW prototype LSC system as implemented results in a CWS combustion process which is in many ways comparable to diesel combustion. High-speed visualization studies of slurry sprays, coupled with empirical evidence from engine experiments, suggest that the CWS fuel spray entrains sufficient air and coal volatiles to yield a combustible air/fuel mixture. Ignition is positive and repeatable using DF2 pilot injection in combination with a 10.6:1 compression ratio and 260°F intake air temperature. Modelling efforts by Texas A&M University and Ricardo-ITI, as well as empirical data from Cooper-Bessemer's sub-scale JS engine and full-scale LSC-1 engine tests have verified rapid coal combustion rates which rival DF2 or natural gas combustion rates (in a 400 rev/min engine). For example, Figure III-2 shows a cylinder pressure trace from the LSC engine while operating on CWS at full speed (400 rev/min) and full load (200 psi bmep). Note that the combustion event is essentially complete after only 30 degrees crank angle duration. New nozzle tip designs and durable coatings/materials have successfully been used to extend the useful life of critical in-cylinder components, thereby allowing hundreds of hours of successful engine operation. Monthly engine test results for all CWS testing on the LSC engine are included in Appendix B.

1. LSC-1 Single Cylinder Development Tests

Cooper-Bessemer first adapted their six-cylinder LS series engine to operate one cylinder on coal (with the other five on diesel fuel) before attempting to run all six cylinders on coal. The research results and component designs from the single cylinder JS-1 engine tests were scaled-up and translated to the full-scale LS engine. The first CWS runs on the full scale cylinder (15.5 in bore, 22 in. stroke) were accomplished in July 1991 and full load operation was achieved in September 1991. Approximately 125 hours of CWS testing on the LSC-1 engine were completed and the results were used to develop and refine the full-scale injection and combustion system before switching all six cylinders to coal operation.

2. LSC-6 Multi-Cylinder Development Tests

Cooper-Bessemer then undertook the task of converting the LSC engine to six cylinder coal operation. A number of major engine components were redesigned to accommodate the coal-fueled engine requirements. A full complement of larger jerk pumps were installed because larger fuel flow rates are required with CWS compared to DF2. A new cam shaft was installed to obtain the desired fuel injection volume per stroke and to carry the increased load from the larger jerk pumps.

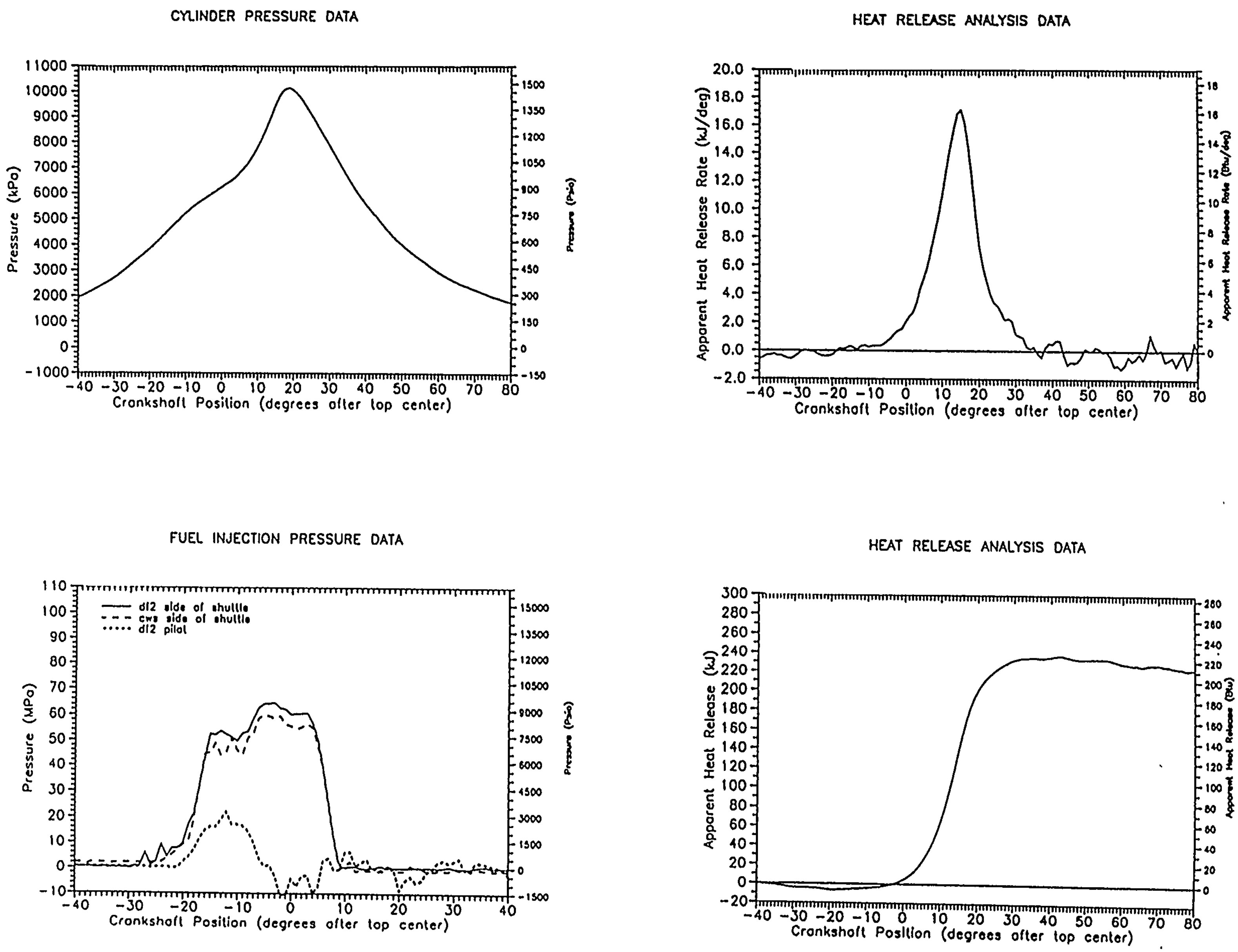


Figure III-2. LSC Coal-Fueled Diesel Test Data Demonstrates Excellent Performance

The cylinder block was also redesigned to accommodate the jerk pumps and larger cam shaft size.

Based on the LSC-1 engine test findings, the initial LSC-6 engine configuration for coal consisted of:

- injection cam: LSC-16-1C "fast rate" cam
- injection jerk pump: L'Orange 36mm plunger and barrel
- nozzle tip: 19 x 0.633mm diameter holes with sapphire inserts
- nozzle spray angle: 140 degree
- injection timing: 23 degrees BTC port closure
- ignition aid: twin diesel pilot injectors at 120 mm^3/stroke/injector (approximately 4 to 5% of the total fuel energy).
- piston rings: tungsten carbide coating
- cylinder liner: tungsten carbide coating

The LSC-6 engine underwent an extended break-in procedure and was then successfully tested on coal-water slurry fuel in September 1992. Full load testing was accomplished in November 1992.

Transition from diesel operation (the engine is started and warmed-up on diesel) to CWS operating was demonstrated to be extremely smooth and rapid. All six cylinders can be switched to coal fuel at once with the engine at full speed (400 rev/min) and 75% load (150 psi bmep). Transition from diesel fuel to CWS fuel took place in less than 15 seconds once coal reached the injectors. Other than a slight audible change in engine sound, the only outward indications the engine is running on CWS are the substantial reductions in NO_x and CO emission levels.

Table III-2 summarizes the test results of initial LSC-6 engine tests. All data was taken at full speed (400 rev/min) over a range of loads from 150 psi bmep to 200 psi. Peak firing pressure and exhaust temperature were well within design limits. The measured engine efficiency (6700 Btu/Bhp·hr) was considered to be excellent, exceeding the program goal. It is important to note that initial combustion optimization was conducted on the LSC-1 engine and that additional optimization was be undertaken on the LSC-6 engine. Table III-2 also lists DF2 fuel performance data taken under the same operating conditions using the CWS injection system. Diesel performance with the engine configuration reoptimized for DF2 (nozzle hole size, injection timing, etc.) would be different.

3. LSC-6 CWS Proof-of-Concept "Endurance" Test

In August 1993, the team of Cooper-Bessemer, Arthur D. Little, AMBAC, and Physical Sciences, Inc. successfully completed the world's first 100-hour "endurance" test with coal water slurry (CWS) on a full-scale 1.8 MW diesel engine with

Table III-2. Initial LSC-6 Engine Performance Results

	DF2 (Baseline)*	CWS (Initial Results)		
Speed (rev/min)	400	400	400	400
bmep (psi)	150	150	175	200
Power (bhp)	1,890	1,890	2,200	2,520
Peak Firing Pressure (psi)	1,320	900	1,400	1,600
Cylinder Exhaust Temp (°F)	930	870	890	920
Specific Fuel Consumption (Btu/bhp·hr)	7,300 (LHV)	7,500 (LHV) 7,800 (HHV)	6,750 (LHV) 7,050 (HHV)	6,700 (LHV) 7,000 (HHV)
NO_x (ppm) prior to SCR	1,180	594	852	1,020
CO (ppm)	2,100	300	-	-
CO_2 (%)	5.8	7.5	-	-
O_2 (%)	12.3	11.5	10.8	10.6

*Using CWS injection system (i.e. not optimized for diesel fuel)

integrated emission control system. The test was started Monday, August 23, 1993, and the engine was run around-the-clock until the test was completed Saturday, August 28, 1993. Engine operation and performance were remarkably steady throughout the test. Approximately 200,000 lbs (22,000 gal) of CWS was consumed.

The engine and fuel system configuration for the endurance test was identical to that used during engine development testing of the full-scale LSC-6. The 100-hour test was run without the cyclone. Running without the cyclone provided the opportunity to evaluate the turbocharger under "worst-case" accelerated wear conditions. The test was run at 400 rev/min and 175 psi bmep, and engine operating conditions were held constant during the 100-hour test. Baseline CWS fuel prepared by CQ was used throughout this test.

Overall engine performance results for this test matched previous full-scale LSC-6 test results and changed only slightly during the 100-hour test duration. Fuel consumption was excellent and was in the range of 6800 to 7000 Btu/bhp·hr (LHV). Cylinder exhaust temperature was normal and in the range of 900 to 910°F (average exhaust temperature of 6 cylinders). Peak cylinder pressure was typically 1200 to 1250 psi and occasionally was as high as 1300 to 1350 psi when individual cylinders were not operating at optimum conditions. Emission rates were also as expected, with NO_x ranging from 700 to 740 ppm at 11% O_2 (450 ppm at 15% O_2) during most of the test (this NO_x level is upstream of the SCR; actual NO_x stack emissions were much lower). Table III-3 summarizes key operating conditions and performance results obtained from this test. Figures III-3 and III-4 show the brake specific fuel consumption and exhaust temperature, respectively, as a function of time during the course of the 100-hour endurance run. This data demonstrates that engine operation and performance were remarkably steady during the test. Appendix C contains photographs of engine and turbocharger components after the 100-hour test.

Table III-3. LSC-6 Engine Performance During 100-Hour Proof-of-Concept Endurance Test

Parameters	Range
Fuel	Baseline CWS
Injection Configuration	Sapphire insert nozzle tips (new); 19x0.633 mm dia holes; 140 degree spray angle; 36mm injection pump; fast rate cam (LSC-16-1C)
Injection Timing	Nominal 18 BTC port closure
Pilot Configuration	Two DF2 pilot injectors per cylinder ($120mm^3$/stroke/injector)
Cyclone	Not present
Speed	400 rev/min
Load	175 psi bmep
Power	2200 bhp
Peak Cylinder Pressure	1200 to 1300 psi
Cylinder Exhaust Temperature	900 to 910°F
BSFC	6800 to 7000 Btu/bhp·hr (LHV)
NO_x (Engine out; prior to SCR) CO (Engine out) O_2 (Engine out)	700 to 740 ppm 180 to 200 ppm 11.3 to 11.4%
Emission control system: NO_x and SO_x* Particulate Stack plume	 70 to 90% NO_x and SO_x reduction >99.99% reduction Invisible

*Does not include NO_x and SO_x removal in coal cleaning step.

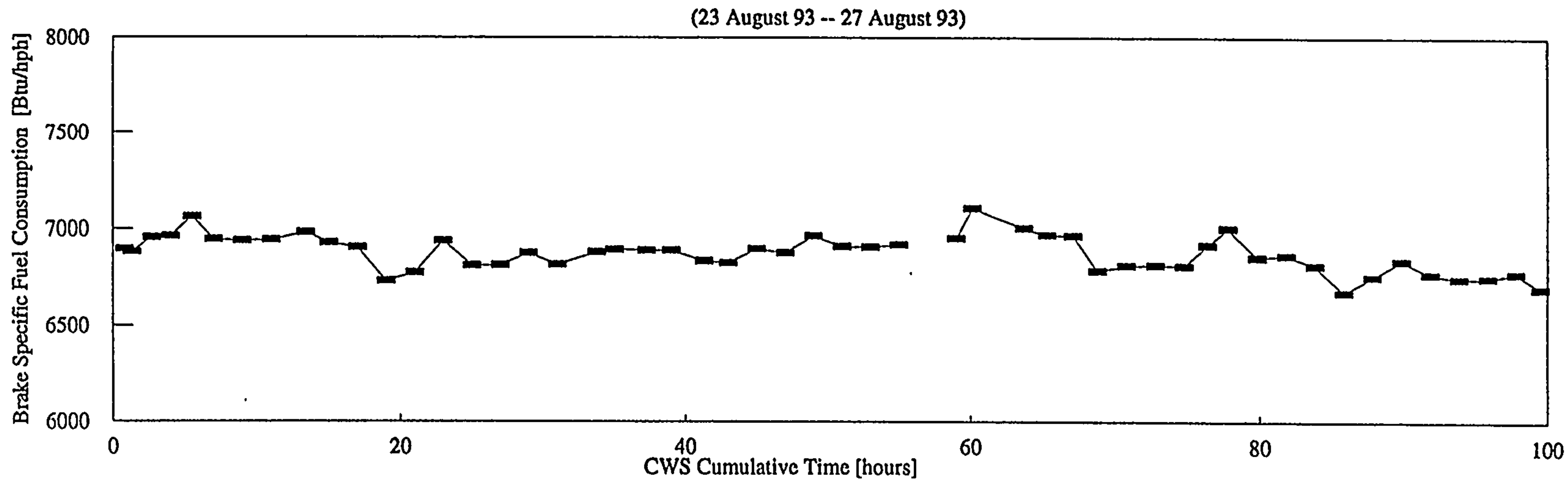


Figure III-3. 100-Hour LSC-6 CWS Endurance Test

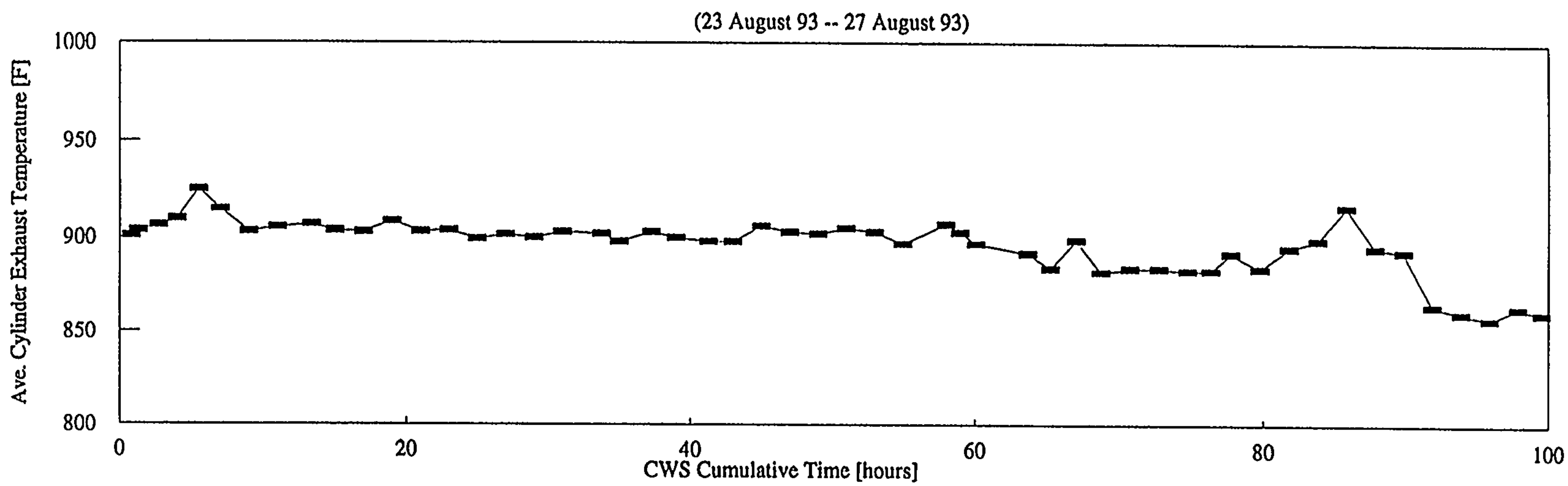


Figure III-4. 100-Hour LSC-6 CWS Endurance Test

B. Engine Performance

1. Major Findings During Full Scale Testing

Effect of Nozzle Tip Hole Size and Number of Holes

Fuel injection nozzle tip hole size and number of injection holes have a significant impact on the fuel spray and fuel-air mixing characteristics. Previous work as part of this project using CWS spray visualization chambers, the MIT rapid compression machine, and sub-scale JS engine tests concentrated mainly on small nozzle hole sizes ranging in diameter from 0.25 to 0.385 mm. Scale-up to the LS engine and fuel quantity required a significantly larger nozzle total open area (hole area x number of holes).

Accordingly, in keeping with this past experience, the initial LSC engine tests were conducted with sapphire insert nozzle tips having 37 x 0.35 mm holes. This resulted in reasonable combustion performance. Reducing the number of holes in an insert style nozzle tip could substantially reduce the cost of injector tips and simultaneously increase nozzle tip reliability. However, the spray and combustion performance impact of drastically changing the nozzle geometry was unknown for combustion chambers as large as those in the LSC-6 engine.

A number of engine tests were conducted using a range of nozzle tips with fewer larger holes to examine the effect of hole size on CWS combustion in the LSC engine (Kimberly, et al., 1993). Nozzle tip hole size ranged from 0.35 mm to 0.633 mm diameter with the number of holes reduced from 37 to as few as 18 holes. These tests demonstrated that nozzle tips with relatively large holes (large relative to most diesel experience) can result in good combustion performance. Table III-4 summarizes the performance of the nozzles tested.

Tests using a 19 x 0.633 mm hole nozzle tip yielded similar CWS injection and combustion performance to that obtained previously when using a 37 x 0.385 mm hole tip. The total open area of each nozzle was similar (4.2 vs. 4.3 mm^2) and injection pressure was therefore also similar (10,500 vs. 10,200 psi). Fuel consumption, exhaust temperature, and firing pressure were also similar for the two nozzle tips. It is believed that larger spray holes result in a greater proportion of larger fuel droplets (drop size distribution shifted to larger drop sizes). The larger average drop size would increase the slurry evaporation and burn time, assuming no fragmentation. If this is the case for CWS under these engine and injection conditions, it apparently does not have a dramatic impact on key performance parameters. This finding was implicitly confirmed by the excellent combustion performance (eg., low bsfc, high heat release rate, etc.) during the 100 hour LSC-6 proof-of-concept endurance run which used 18 x 0.633 mm diameter holes. The 18-hole injectors were identical to the 19-hole injectors, except that the center hole was not used.

Table III-4. Performance Summary of Injector Nozzle Tip Configurations and Comments

# Holes	Holes Size, MM	Spray Angle	Pump Size, MM	Inj. Timing, °BTC	Comments
37	0.385				Initial tests gave satisfactory performance.
19	0.633	120 140 155	36	23	Achieved 200 psi bmep w/good bsfc (6400 to 6700), peak pressure relatively high at this injection timing (1300-1400 psi).
19	0.633	120 140 155	32	23	Could not achieve 200 psi bmep w/32mm pump without exceeding peak pressure limit or pump capacity. <u>Implication</u>: Must use larger 36 mm pump size with the injectors.
25	0.533	155	32	23	
25	0.533	155 120	32	21	Achieved 200 psi bmep with good bsfc (6500 to 7000 but/bhp•hr) but usually with peak pressure very close or above peak pressure limit. Preliminary results indicate that peak pressure can be controlled using injection timing retard with only a small (or negligible) fuel consumption penalty.
19	0.533	155 120	32	21	
31	0.533	155 120	32	21	
18	**0.633**	**140**	**36**	**18**	**"Best" configuration selected for 100-hour test**

<u>Effect of Nozzle Spray Angle</u>

The shape of the fuel spray generated by the nozzle tip is known to be important in diesel combustion. The optimum shape is influence by the combustion chamber shape, in-cylinder air motion, injection rate, and fuel spray/droplet behavior. Based on previous experience, one of the objectives for the LSC engine combustion chamber/fuel spray optimization was to minimize fuel impingement on the cylinder head fire deck, piston, and cylinder liner while achieving maximum air utilization and rapid fuel-air mixing rates.

During the LSC engine development testing, nozzle tips with spray included angle ranging from 120 to 155 degrees were evaluated. Within this range, all nozzle tips gave similar results for the LSC engine combustion chamber. The 140 degree spray angle appeared to have a slightly shorter ignition delay resulting in higher peak firing pressure (depending on injection timing). Preliminary evidence suggested the 155 degree tip deposited some CWS on the fire deck and may not be acceptable for long duration operation. Table III-5 shows typical results for a test series evaluating fuel spray angle while other parameters were held constant. No large combustion performance impact was observed when spray angle was changed within this range. The 140 degree spray angle was selected for most of the development tests including the 100 hour proof-of-concept CWS "endurance" run.

Table III-5. Effect of Spray Angle on CWS Combustion Performance (19x0.633mm Nozzle Tip)

Spray angle, degrees	120	140	155
Peak Firing Pressure (psi) @ crankangle (deg. ATC)	1470 @ 15	1540 @ 12	1500 @ 15
Approximate bsfc (Btu/bhp·hr), LHV	6700	6700	6400

Effect of Injection Timing

CWS fuel injection timing affected a number of important engine performance parameters including fuel consumption, peak cylinder firing pressure, and NO_x emission rate. The optimum injection timing for the LSC engine on coal fuel was about 18°BTC at 400 RPM, full load. This timing was a compromise between best bsfc and lowest NO_x without exceeding the peak cylinder pressure constraint. Test experience showed that the optimum timing was affected by operating parameters such as engine speed and load, manifold air temperature and pressure, fuel composition, and injection system configuration.

The effect of CWS injection timing was explored throughout the development program. For the LSC engine, a relatively wide range of injection timing resulted in acceptable engine performance. However, timing became very sensitive for certain combinations of injection system configurations for which the peak firing pressure was near or exceeded the design limit. One of the most important findings was that, near the optimum injection timing, small changes in timing (2 to 5 degrees) dramatically impacted the peak firing pressure with only a small or negligible change in fuel consumption.

To illustrate this sensitivity to timing, during a specific engine test series, injection timing was advanced 3 degrees (25 degrees BTC port closure from 22 degrees BTC) so that start of injection was approximately 21 degrees BTC (vs. 18 degrees BTC). However, at this setting, engine operation was unsatisfactory because the peak cylinder pressure limit was exceeded before full load was achieved. Peak firing pressure was significantly higher for the advanced timing case (1480 vs. 1310 psi at the 380 ihp/cylinder condition) but fuel consumption was similar (7500 vs. 7700 results illustrate that two to five degrees injection retard (from best timing) reduced peak firing pressure 50 to 150 psi, increased exhaust temperature approximately 100°F, but had a negligible affect on fuel consumption. Therefore, injection timing retard can be used effectively to control peak firing pressure without a bsfc penalty.

Effect of Jerk Pump Plunger Size

The large capacity L'Orange diesel jerk pumps used on the LSC engine can be fitted with a range of plunger diameters. In general, for a given injection cam profile, the larger diameter plungers are able to deliver more fuel as well as inject it at a faster rate. During the LSC engine development tests, it was observed that jerk pump plunger size influenced the maximum power, heat release rate, and peak firing pressure.

In one test series, the injection rate was increased by replacing the 32mm jerk pump with a 36mm jerk pump. Nozzle hole size was increased from 0.533 mm to 0.633 mm (40% increase in total open area) to keep injection pressure at a nominal 10,000 psi. The faster injection rate (larger plunger diameter) produced higher peak heat release rates (16 Btu/degree vs. 12 to 13 Btu/degree) and higher peak firing pressure (1470 vs. 1310 psi). However, fuel consumption and exhaust temperature were similar for the two injection rates.

Given the benefits of using nozzle tips with the larger hole size (i.e., 0.633mm holes), the 36mm plunger diameter was selected for most of the LSC-6 development work and the 100 hour proof-of-concept endurance run.

Effect of Injection Cam Rate Profile

During the LSC development tests, the "slow rate" injection cam was evaluated using a variety of nozzle tips (37, 19, 25 holes; 140 and 155 degree spray angle). All tests were run with the 36mm jerk pump with timing set to either 21 or 23 degree BTC port closure.

The "fast rate" cam with the 36 mm jerk pump yielded injection durations of 25+/-2 degrees (depending on nozzle tip), resulting in timely burning of CWS. The "slow rate" cam yielded longer injection durations (approximately 40 degrees; 60% longer) than the original CWS injection cam. The longer injection duration resulted in lower injection pressure for a given nozzle tip. The longer duration and lower injection pressure resulted in lower peak cylinder pressure (1300 to 1400 psi vs. 1400 to 1600 psi, depending on nozzle tip), but significantly worse specific fuel consumption (7% to 20% worse). Two degree injection timing advance did not improve bsfc indicating that fuel consumption is relatively flat around the optimum timing (as discussed earlier).

As a result of these tests, the "fast rate" cam (CB LSC-16-1C) was used for the 100 hour proof-of-concept endurance run.

Effect of Ignition Pilot

Experience from JS engine tests showed that a single DF2 pilot injector (6 x 0.27mm holes) was an effective ignition aid with pilot fuel quantities as low as 3% of the total fuel energy. A series of tests were conducted on the LS engine to explore the sensitivity of DF2 pilot fuel quantity and pilot injection configuration on CWS performance. Tests were run with pilot quantity ranging from 2.3 to 7.9% of the total energy using single or twin pilot injectors. In general, twin pilot injectors had lower cycle to cycle variation in peak firing pressure than a single pilot injector (standard deviation of 15 to 19 versus 33 to 36 psi) indicating better combustion stability. Pilot quantity did not have a significant effect except for the test which had 7.9% pilot. This test showed lower bsfc (7200 vs. 7600 Btu/bhp•hr) but had higher peak firing pressure 1530 psi). Twin pilot injectors delivering approximately 4 to 5% of the total fuel energy (120mm^3 DF2 each injector) were used for further tests.

The analysis of the testing indicates that, as mentioned above, **two direct diesel pilot sprays injected across the chamber give the best ignition delay.** Ignition delay with these two pilots running at 220mm^3 is under 4 ms., which is equivalent to the delay observed with the JS engine. Roughly 4% of the energy going into the cylinder is through the pilot. An unusually rapid rise of the combustion pressure was observed after light-off. This drives the firing pressures above the usable limit. In order to slow down the pressure rise rate, the 32mm pump was used, which gives an injection rate equivalent to the slower cam that was procured. Also, the timing was shifted from 22° to 31-½° before top center port closing. In addition, the diesel torch pilot was modified to two direct diesel spray pilots over a range of fuel flow from 50mm^3 to 440mm^3; and the manifold air temperature was increased to as high as 300°F. The combustion trace consistently showed that once light-off occurs, there is a very rapid combustion. This was noticed on the JS engine when the 140° included-angle nozzles were tested.

IV. Durable Components

A. Introduction

It was recognized early in this program that a considerable challenge lies in developing engine components that could tolerate the abrasive environment of the coal water slurry (CWS) and/or the coal ash residue for reasonable periods of operation. Initial CWS tests in the JS engine and those conducted by other researchers showed that components made with standard steel materials permitted operation for less than two hours. Operating cost calculations suggested that, even though required component lives could be somewhat reduced for the coal-fueled diesel engine, average operating times of thousands of hours were still required.

Nevertheless, a host of candidate solutions was available. Hard materials appeared to offer the best possibility, and this was the primary approach taken by the Arthur D. Little/Cooper-Bessemer team. This approach also had been the most successful for the German companies who had pursued coal-fueled diesel technology in the 1930-1944 time period (Soehngen, 1976). Prior German work also suggested a range of novel, engineering approaches to the wear problem and many of these were again pursued during the period 1988-93 under other DOE funded heat-engine projects. These included schemes to inject coal fuel under low pressure (Badgley, 1992) and to wash the abrasive from the cylinder wall during part of the cycle (Raymond, et al.; 1991). Some new ideas were also pursued, including use of the coal fuel as a dry lubricant on the cylinder wall (Heshmat, 1990) and the scribing of a surface geometry to direct the abrasive away from the piston ring faces (Schwalb and Ryan, 1991). All of these engineering approaches were considered but ultimately rejected as impractical in the course of this program.

Ultimately, and almost without exception, hard materials proved to be the best approach to achieving the desired life goals for engine parts. Table IV-1 lists the materials that provided the greatest life in this program together with an estimate of the projected total lives compared to the current requirements for economical cost of operation. In a few cases - oil control rings and turbocharger rotor blades - durable component solutions have not actually been tested in the coal-fueled diesel engine. Projected lives of these components are based on laboratory test results rather than actual engine data. There are also a few components for which other approaches deserve further development. In particular, monolithic ceramic or improved hard coatings may be necessary for the exhaust valve seats and guides, and that some ceramic/metal composite is needed for the top compression ring to meet the life targets.

This chapter describes the mechanisms of wear, the approaches pursued and test results for each of the durable components listed in Table IV-1.

Table IV-1: Durable Components Developed for the Coal-Fueled Diesel Engine

Component	Best Solution Tested	Maximum Test Time on a Single Component (hours)	Projected Life* (hours)	Target Life (hours)
Injection nozzle tip orifices	Sapphire inserts	142	>500	500
Injection nozzle valve	Titanium nitrided steel; monolithic tungsten carbide tip	142	>500	500
Injection nozzle valve seat	Monolithic tungsten carbide	142	>500	500
Injection nozzle shuttle	Titanium nitrided steel	400	>500	500
Cylinder liner	Tungsten carbide plasma coating	232	>5000	12,000
Top Compression rings	Tungsten carbide Detonation Gun coating	232	>2000	12,000
Other compression rings	Tungsten carbide Detonation Gun coating	182	>5000	12,000
Oil control rings	Chrome plate	0	>5000	12,000
Exhaust valves	Tungsten carbide Detonation Gun coating	181	>500	12,000
Exhaust valve seats	Tungsten carbide Detonation Gun coating	100	>1000	12,000
Turbocharger rotor blades	Cyclone plus chrome carbide Detonation Gun coating	0	>10,000	25,000
Crankcase bearings	Centrifuge	450	Indefinite	25,000

*Note: Projected life can be improved by further hard materials applications development.

B. Wear Observations and Solution Approaches

Durability problems in the coal-fueled diesel engine are associated with wear; cases in which strength must be considered in durability assessments are those for which the material type has been changed substantially from that used for standard diesels. Most of the wear problems are caused by the coal slurry fuel in its raw form or the ash which remains after the fuel has been combusted. Some wear is caused by the mating of new, hard materials such as the piston rings against the cylinder liner.

1. Abrasive Characteristics of the Fuel

The coal water slurry and the resulting exhaust particulate derive their abrasiveness primarily from the ash content of the original fuel. The effort placed on optimizing the ash content/fuel cost - determined primarily by the cost of cleaning - is described in Section II.A. Furthermore, it is the silica in the ash that is the primary abrasive particle. In the previous program (Mayville, 1990), we determined several characteristics of the abrasive coal fuel and exhaust particulate that could be used to advantage to reduce wear:

- The hardness of the abrasive is approximately 500 kg/mm^2
- Wear rate increases linearly with ash content
- The silica content increases by one order of magnitude as the coal is converted from raw fuel to exhaust particulate
- Wear rate in ring/liner sliding increases linearly with the concentration of the abrasive in the lubricating oil.

The implications of these observations are that fabrication of components from materials harder than 500 kg/mm^2 will decrease wear dramatically (this is discussed in more detail below), that component life can be increased in direct proportion to the decrease in ash content and that methods to dilute the abrasive in the contact zone between ring and liner could have significant benefit.

The first two of these observations provided methods to substantially reduce wear, while the third observation had little practical value, primarily because the concentration of abrasive between ring and liner is evidently very low for the combustion process in our engine. In fact, tests in which the engine was intentionally stopped suddenly while burning CWS fail to show the presence of any coal fuel or ash residue on the cylinder liner.

2. Engine Wear

Components in the coal-fueled diesel engine that wear are determined simply by the path of the fuel through the engine, see Figure IV-1. Table IV-2 lists by component the mechanism of wear and the observed life when components of standard materials are used; the priority with which each component was addressed was determined by the extent to which its wear limited operation.

This table shows that two components, the injection nozzle tip orifices and the piston rings, were the components that most limited continuous operation. For this reason, they received the most attention in this and other DOE-funded programs. Potential solutions were derived for each of the components in Table IV-2 (see Table IV-1) except for the oil control rings and the turbocharger rotor blades (these components were not tested in this coal-fueled diesel engine in this program). All other components were modified with hard materials and tested in the prototype coal-fueled diesel engine.

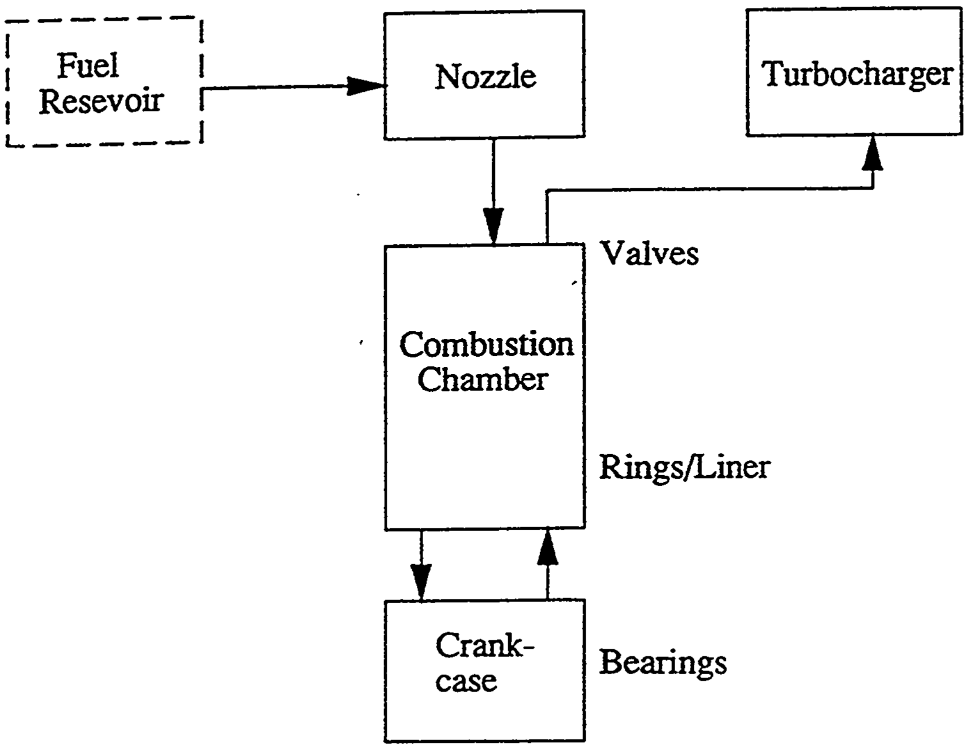


Figure IV-1. Schematic Illustration of the Abrasive Coal Fuel Flow Path and the Engine Components Affected

Table IV-2: Mechanisms of Wear for Components in the Coal-Fueled Diesel Engine

Component	Wear Mechanism(s)	Life on CWS for Standard Materials (hours)	Material
Injection Nozzle			Carbonitrided steel
Orifices	Solid particle erosion and Cavitation	1.5	
Valve	Solid particle erosion	150	
Valve Seat	Solid particle erosion	150	
Cylinder Liner	Three body abrasion	>500	Chrome plate
Piston Rings	Three body abrasion	70	Ductile iron
Exhaust valves	Solid particle erosion	300	Inconel 718
Turbocharger rotor blades and vanes	Solid particle erosion	300-500*	Inconel 718
Crankcase bearings	Abrasion	>1000	Various bearing materials

*Tested without protective cyclone upstream.

3. General Approach to Developing Durable Components

The results of prior work in Germany and the observation that the coal abrasive is relatively soft provided a strong incentive to focus on the use of hard materials to obtain the required component lives. As noted previously, a decade of work in Germany culminated in the conclusion that hard materials - at that time, very hard cast irons - resulted in the lowest wear rates. Bench top experiments also showed the dramatic decrease in wear rates that could be obtained with hard materials. Figure IV-2 shows the results of reciprocating wear tests in which a slurry of oil and silica was introduced between specimens meant to simulate the action of a piston ring on a cylinder liner. The figure plots wear rate as a function of the ring specimen hardness and shows that a reduction of wear - which translates directly to an equal increase in life - of two orders of magnitude can be achieved by application of ceramic materials in place of the conventional ductile iron. The approach used was to find methods to apply the hardest materials possible, while maintaining performance.

This was accomplished by testing the materials in apparatus of increasing mechanical complexity, finally being tested in the coal-fueled diesel engine itself. Bench top apparatus were used to test materials for the piston rings, cylinder liners, exhaust valves and turbocharger rotor blades. For example, small coupons were coated and tested in a high temperature, erosion-corrosion apparatus to simulate the engine exhaust environment. In some cases, sub-scale components were fabricated and tested. A silica/oil slurry-injected, gas-fired, two-stroke diesel engine was used to test various combinations of ring and liner materials. A single-cylinder JS engine was used to test most of the components in an actual coal-fueled environment. A larger multi-cylinder LS engine provided the final 100-hour proof-of-concept demonstration test of the system.

C. Component Development

1. Injection Nozzle

The purpose of the injection nozzle is to accurately meter fuel into the cylinder for each combustion cycle. This is accomplished in conventional diesels and in the coal-fueled diesel engine with a high pressure, multi-orifice nozzle. Figure IV-3 shows a cross section through part of a durable injection nozzle. The parts of the nozzle that require protection for the coal-fueled diesel engine application are: at the tip, the orifices, the valve, valve seat and, further upstream, the fuel metering shuttle and the inlet check valves. Wear rates are not available for these components because, to date, the material loss has been too slight to measure. Eight injection nozzles with durable components for each of these parts have each been tested for over 100 hours on CWS with negligible wear.

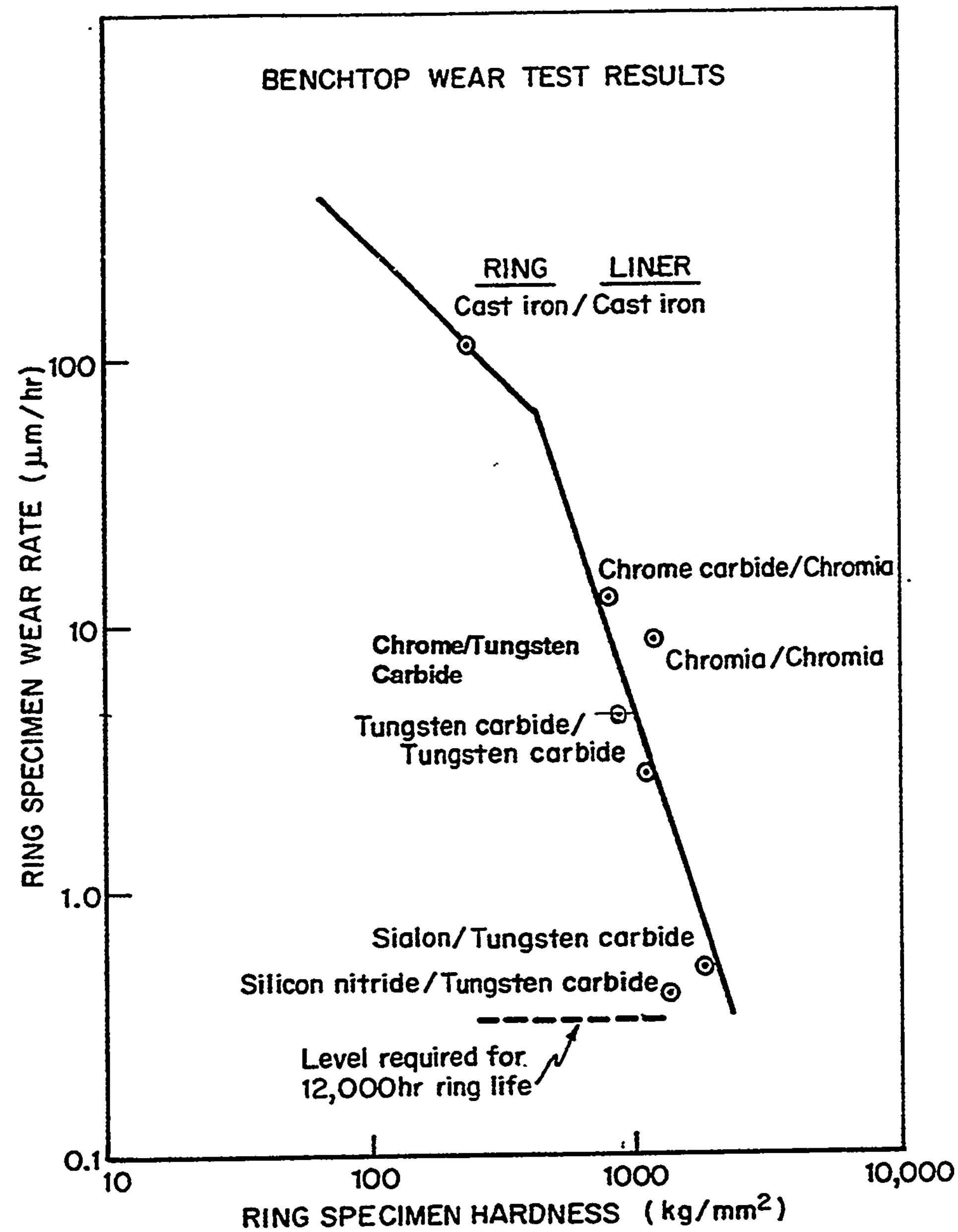


Figure IV-2. **Wear Rate as a Function of Hardness for Various Materials Used and Considered for the Coal-Fueled Diesel Engine**

Orifices

The nozzle tip includes approximately 20 orifices, 0.018 inches in diameter, that can tolerate very little wear without degradation in performance. For example, a common limit on orifice diameter increase for conventional orifice nozzles is 0.001 inches.

The durable nozzle design utilizes sapphire tube inserts to mitigate orifice wear, see Figure IV-3. The inserts are fitted into a 410 stainless steel cylindrical piece which is in turn electron beam welded into countersunk holes in a 410 stainless steel nut.

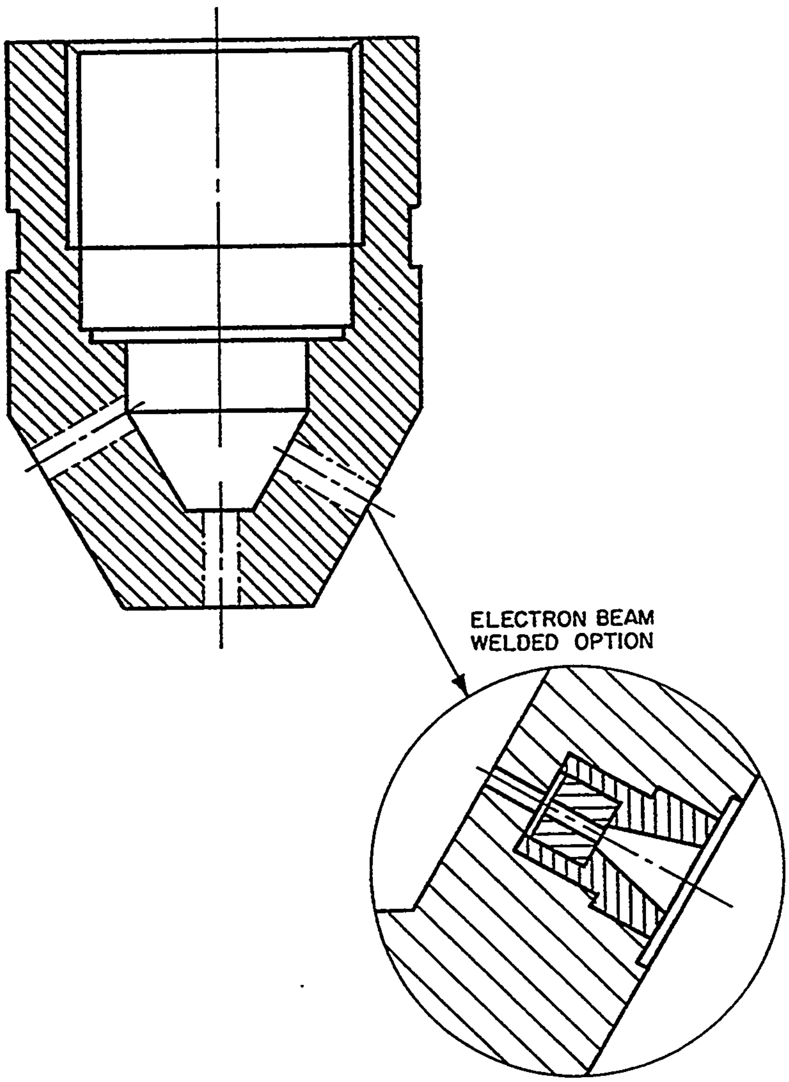


Figure IV-3. **A Sketch of the Sapphire Insert, Durable Injection Nozzle Tip Used Successfully in the 100 Hour Demonstration Test**

This nut is threaded onto the nozzle body which contains the valve, shuttle and other parts.

Over 100 hours of operation on coal fuel with eight nozzles have been accumulated, and in each case the orifices show negligible wear. Wear causes the circular hole to become slightly ellipsoidal, but not for every insert. This is likely due to the fact that the sapphire is softer in some of its crystallographic planes. In the course of the program, there were isolated instances where a few of the sapphire inserts were found to be absent from some of the nozzles after testing in the engine. This problem was traced to too large a tolerance between the sapphire insert O.D. and the cylindrical metal insert I.D. and was easily corrected.

An alternate, button nozzle tip design was also demonstrated to provide the required life for the coal-fueled diesel engine. The button, shown schematically in Figure IV-4, is a monolithic, tungsten carbide cermet piece that is fastened to the nozzle body with a tapered nut. Two such nozzle tips endured over 100 hours of operation on coal-fuel without wear. This design was not pursued because of the difficulty in fitting the required number of orifices into the small button and because of the current, long lead times required to fabricate buttons.

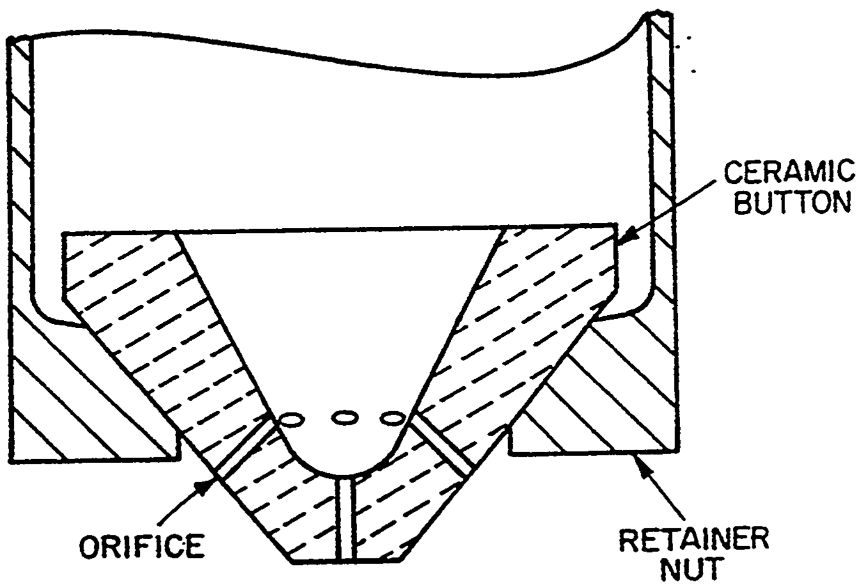


Figure IV-4. An Illustration of a Monolithic Tungsten Carbide Injection Nozzle Tip Button Design

Valve and Valve Seat

Fuel is admitted to the nozzle tip by the opening and closing of a needle valve. The current design for valve consists of a titanium nitride coated A2 steel valve stem and a monolithic tungsten carbide tip. The valve seat is also a monolithic tungsten carbide piece brazed into the nozzle body. Six sets of these components have survived over 100 hours of CWS operation without significant wear.

Shuttle

The shuttle provides the pumping action and the interface between the pressurizing oil and CWS. Although the tolerance between the O.D. of this part and the I.D. of the part into which it fits is quite small - ~ $2x10^{-6}$ inches - wear from the coal can still occur. Therefore a titanium nitride coated shuttle was used in a hardened steel guide and this combination has successfully endured over 400 hours on CWS in seven nozzles.

Check Valves

The check valves admit CWS into the chamber of the nozzle during the injection cycle. Tapered seat check valves have been implemented and operated for over 100 hours with monolithic tungsten carbide valves and seats.

2. Piston Rings and Cylinder Liner

The piston rings and cylinder liner provide many functions. Most important of these is the formation of a seal against the gases in the combustion chamber. However, the cylinder liner must also provide support against the thrust loads of the piston and a chemical and thermal barrier against the source for and products of combustion.

Durable piston rings and cylinder liners have been achieved by the application of tungsten carbide coatings. Compression rings are provided with a groove on their face, into which is sprayed a coating using Praxair's Super D-Gun coating SDG 2047. The groove, illustrated in Figure IV-5 protects the edge of the coating from chipping. Liners are coated with Praxair's LW11B. This latter coating is plasma sprayed because the inside diameter and length of the cylinder liner preclude use of the D-gun or Super D-gun application technique. Durable oil control rings were not developed during this program because of time and priority constraints. However, the proposed component is a chrome plated ring that has been tested successfully in the lab.

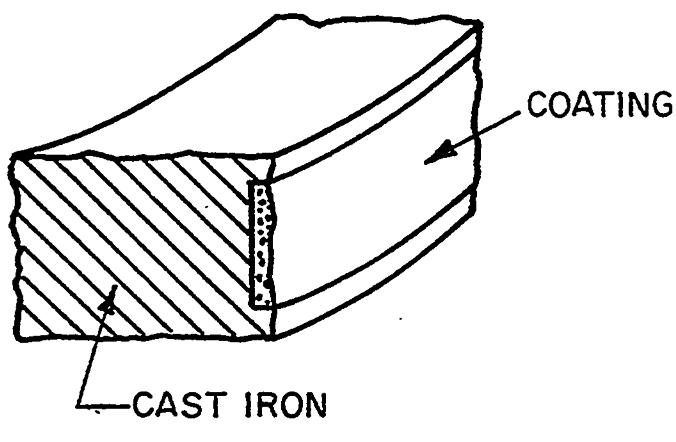


Figure IV-5. The Piston Ring Face Geometry Used to Accept Hard Coatings for Coal-Fueled Diesel Engine Service

Compression Rings

The loss of seal between the piston rings and the cylinder liner - specifically the growth in ring end gap - is the limiting durability mode in the coal-fueled diesel engine. Cooper-Bessemer generally recommends ring replacement when the end gap change exceeds approximately 0.100 inches. Acceptable performance has been observed with end gap changes of over 0.150 inches. Use of the second figure as a design goal for operation of 12,000 hours requires that the combined allowable radial wear of ring and liner be:

$$0.150 = [\Delta r \text{ (ring)} + \Delta r \text{ (cylinder)}] x \ (2\pi)$$

Now each ring will experience far more sliding than any given segment of the liner because the ring must traverse the entire liner with each stroke while a segment of the liner will experience rubbing only when the rings are in its vicinity. In fact, the sliding distance experienced by each ring for the LS engine is:

$$x(\text{ring})/\text{stroke} = 20 \text{ inches}$$

while a segment of the liner experiences a sliding equal to:

$$\begin{aligned} x\ (\text{liner})/\text{stroke} &= (\text{face width of ring})x(\text{number of rings}) \\ &= (0.25)x(6) = 1.5 \text{ inches.} \end{aligned}$$

If the wear rate were the same per unit load and sliding distance for the ring and liner materials - it is to a first order approximation - then we would expect the radial change on the rings to be over ten times that of the liner, making the rings the most critical component. Thus, it is reasonable to design and assess durable ring/liner component life primarily on the basis of ring life and to design the liners to last as long as the rings. The required maximum wear rate for the rings then becomes:

$$\begin{aligned} \Delta\text{end gap} < 0.150/12000 = 1.2x10^{-5} \text{ inches/hour} \\ (1.2x10^{-3} \text{ inches/100 hours.)} \end{aligned}$$

Table IV-3 lists the range of compression ring wear rates for rings that did not fail because of processing related problems. This table shows that, in general, all but the top compression ring have wear rates near the required value. Additional, longer term testing is required to determine whether the ring wear rates will decrease or increase with running time and whether further development will be required for the top or other compression rings.

Table IV-3: Durable Compression Ring Wear Rates on CWS

Ring I.D.	Average End Gap Wear (10^{-3} inches/100 hr)	Longest Time on a Single Ring (hrs)
Top	11.4	232
Second	3.8	182
Third	3.7	182
Fourth	2.2	103

Some effort was expended on more durable compression rings. Bench top wear testing (see Figure IV-2) indicated that monolithic ceramics could provide the needed wear resistance and research was conducted to determine methods by which ceramics could be practically implemented into piston rings. The two methods considered were: (1) the fabrication of split, all monolithic ceramic rings; and (2) the brazing of ceramic inserts into a metal substrate ring. Only the latter concept was pursued because it could provide the toughness required for the mechanically harsh environment of installing and operating rings in diesel engines and would permit finish grinding by the same procedures used for coated rings. Unfortunately, a suitable method for fabricating such rings could not be developed within the time of this program.

Coating technology continued to improve during the course of this program, providing coatings that behave more and more like monolithic materials. These advanced coatings will be used in an effort to meet the top compression ring life goal of 12,000 hours.

In some of the tests, failure or accelerated wear of the ring coatings due to failure modes other than the expected abrasive wear has been noted. For example, in some cases, coatings were not adequately ground leaving very thin coating material on the edges of the rails (see Figure IV-6). This material was easily chipped during operation and caused substantial scratching in the rings and, to a lesser extent, the coated liners. If the coating is too thick - say 0.020 inches - that cracks will propagate from the surface of the coating to the bottom of the groove. As a result, the coating specification has been tightened to ensure proper grinding and the use of coatings less that 0.015 inches thick.

At the required maximum average end gap wear rate of 1.2×10^{-3} inches/100 hours, the coating thickness must be at least 0.025 inches to achieve a 12,000 hour life. As a result, there is still some development required to apply ring coatings this thick that will not fail by cracking.

Oil Control Rings

Throughout the entire program only oil control rings of standard ductile iron were used in the coal-fueled diesel engine tests. These rings with their tapered rails do not permit the application of plasma or D-gun sprayed coatings into grooves because of their narrow face geometry.

Instead, the use of chrome plated rings for this component has been investigated. Figure IV-2, above, showed how the chrome plate on tungsten carbide provides about one-half the wear resistance of the tungsten carbide coatings. Because wear rates of ductile iron oil control rings are less than one-half that of ductile iron compression rings when run on CWS and because larger end gap changes are allowable, it is expected that the chrome plate can provide the required 12,000 hour life. Actual engine testing of such rings will be required to verify this. If this is not successful, the next solution will be application of the newer Super D-gun type coatings, the unprotected edges of which Praxair believes may be able to run without chipping.

Cylinder Liners

Liners coated with LW11B have shown very little wear and almost no scratching or chipping problems. Wear is generally below the measurement threshold, which is about 0.001 inches on the diameter. This is only an indication that other degradation modes have not arisen and that the relationship described earlier seems to apply; that is, the wear rate of the liner is about 10 times less than that of the rings. Thus, the LW11B coating is currently the preferred approach to achieving durable liners.

3. Exhaust Valves and Seats

The exhaust valves of the coal-fueled diesel engine provide the path for the exhaust gases, uncombusted coal and the ash to be emitted from the combustion chamber. Their environment is quite severe. Temperatures of the exhaust are in the range of 850 - 900°C (1560 - 1650°F) and the gas velocity is estimated to be 70 m/sec (220 ft/sec) during part of the exhaust stroke. Exhaust particulate consists of 20 to 40% ash making the exhaust particularly abrasive. And the volume of material that flows by the exhaust valves is enormous: in a 20 cylinder engine running for 10,000 hours, each valve will pass 7500 kg (17,000 lb) of abrasive ash.

Wear on the exhaust valves tends to be concentrated in the area at which the stem meets the head, as illustrated in Figure IV-7. The reason for this is that the entrained particles are impinging at angles that range from 20-40° to the valve surface tangent. These are the impingement angles at which wear by a ductile removal process is greatest.

Valves

To date, a satisfactory durable exhaust valve has not been demonstrated. Hard coatings applied to the surface below the guide area increase the life by a factor of

Figure IV-7. A Photograph of an Inconel 718 Exhaust Valve Run for 250 Hours in the Coal-Fueled Diesel Engine

two over valves of standard material, but the maximum life is still projected to be about 500 hours on CWS. There were some data in the beginning of the program to suggest that material loss on the valves was due to erosion-corrosion, in which case coatings would have provided a substantially greater improvement. However, the 100 hour test on the LS engine has demonstrated that the wear mechanism is primarily solid particle erosion for which the thin coatings - approximately 0.010 inches thick - impart minimal increase in life.

The current recommendation is to pursue monolithic ceramic valves or inserts. Norton-TRW has provided such exhaust valves to a number of industries, and it appears that monolithic SiAlON ceramic valves can be implemented in the LS engine with only slight modification to the locking groove.

Table IV-4 lists the exhaust valve wear data obtained from operation of the LS engine on CWS. The increase in life due to application of the coatings is consistent with bench top erosion studies carried out early in the program. These laboratory experiments did not include the testing of SiAlON or other ceramics, but it is expected that the erosion resistance of the monolithic ceramics at high temperatures will be superior to the coatings. Combine this property with a monolithic valve, rather than thin coatings, and it is expected that the life of the exhaust valves can be increased to approximately 10,000 hours.

Wear on the seats of the exhaust valves has been minor. The slight seat wear observed with the standard Inconel 718 valves is reduced to negligible wear when coatings are applied. This is likely due to the very low impingement angle of the entrained particles as they pass the seat area.

Table IV-4: Exhaust Valve Stem Wear Rates Observed in the Coal-Fueled Diesel Engine Tests

Comments	Material	Change in Diameter (10^{-3} inches/100 hours)
Uniform Wear	Inconel (no coating)	110
Localized Wear	Tungsten carbide coating	65-130
Localized Wear	Triballoy coating	65-130
Uniform Wear	Chrome carbide coating	50

4. Valve Seat Inserts

The exhaust valve seat inserts experience an environment that is very similar to the valves themselves. The primary difference is in the angle of impingement of the exhaust gas and the entrained particles: it is very low in the seating area as described above. For this reason, very little wear occurred on the seat inserts even for standard materials. The current choice for the seat insert material is a tungsten carbide coating.

5. Turbocharger

The purpose of the turbocharger is to utilize the exhaust to boost the pressure in the cylinders during the compression stroke of the engine cycle. It is the next significant component in line after the exhaust valves. Thus, without some intermediate step or process the turbocharger would experience the same high temperature abrasive conditions as the exhaust valves. Figure IV-8 shows a photo of severe guide vane wear in the radial flow turbocharger used without protection in the 100 hour proof-of-concept test on the LS engine. The turbocharger recommended for use in the LS engine is an axial flow CB13 model, for which only the rotor blades would be subjected to wear from the CWS.

The approach to providing durable turbocharger components relies on two lines of defense, neither of which were demonstrated during the course of this program. The first defense, and one commonly used in protecting rotating turbomachinery in abrasive environments, is to use inertial separation to remove abrasive particles from the exhaust stream. General practice is to remove all particles over 3 μm from the air that passes through the turbomachinery. The cyclone developed for this program, its specifications and performance are described in Chapter V of this report.

The second approach to mitigating turbocharger wear is to coat the rotor blades with hard coatings. This protection is provided because the cyclone is not 100% effective in removing particles over 3 μm. The preferred coating, based on the bench top erosion studies described in the previous section, is Praxair's SDG2207, a chrome carbide Super D-Gun coating.

6. Crankcase

The crankcase contains the lubricating oil for several components including the crankshaft and connecting rod bearings and the piston rings. Contamination of the crankcase with burned or unburned fuel could possibly lead to wear of bearings or blockage of important oil passages. Analyses on the LS engine show that coal-related particulate enters the crankcase at a rate of about 340 grams (0.75 lb)/ hour of six cylinder operation on CWS. This indicates that approximately 0.05% of the fuel intake - which is about 2000 lb/hour - manages to get past the rings. Chemical and spectrographic analysis of this particulate indicates that a significant percentage of it is unburned coal.

To date, no damage to the bearing components has been observed in either the JS engine, whose bearings have been inspected, or the LS engine. However, without intervention, the mass of particulate would continue to accumulate with operating time and clogging would eventually occur.

The approach to mitigating any adverse effects from this particulate is to use a centrifugal separator on the oil in the crankcase sump. A 5,000 gallon/hour system was purchased and used on the LS engine during all of its operation. A computer model was developed and verified to show that with this centrifuge an equilibrium concentration of particulate in the sump will reach 0.5%. No bearing damage was observed in the JS at such a concentration. However, the high frequency with which the centrifuge must be cleaned and the high steady state mass of particulate that will result (200 lb) suggests that a larger, self cleaning centrifuge should be applied for long-term operation of multicylinder coal-fueled diesel engines.

V. Emission Control System

A. Emission Control System Design

Effective controls for NO_x, SO_x, and particulate emissions are essential for successful commercialization of stationary, coal-fueled diesel engines. A major goal in the program was to establish the optimum emission control system from performance and cost perspectives and then to demonstrate the ability of this system to reduce pollutants to levels which will be required at 5 to 50 MW cogeneration and independent power production sites in the year 2000 to 2030 timeframe.

As part of this effort, PSI Technology Company (PSIT) and Arthur D. Little (ADL) developed, installed, and tested an integrated engine Emission Control System (ECS) capable of treating the 1.8 MW engine's full exhaust flow (7700 scfm).

1. Engine Emissions and Control Targets

Emission measurements conducted during single cylinder engine testing combined with coal-water slurry (CWS) properties provided a sound basis for initially defining uncontrolled emission levels from full scale coal-fueled diesel engines. The emissions characteristics of the ECS were designed to be superior to those of larger, advanced, coal-power options. The projected levels and ECS performance targets were as follows:

Particulates: Commercially-viable, engine-grade, CWS is expected to contain 1 to 2 wt% ash (dry basis). Although this is much lower than the parent coal, particulate control devices are still necessary. With the demonstrated high engine combustion efficiency (99 to 99.5% carbon burnout), uncontrolled particulate emissions have been measured at about 1 to 3 lb/MMBtu. Achieving the coal-fired boiler New Source Performance Standard (NSPS) level of 0.05 lb/MMBtu requires a reduction of about 95 to 98%. In addition to air pollution considerations, particulate control is needed to protect the engine turbocharger from potentially severe wear.

SO_2: Engine-grade CWS has a sulfur content of about 0.7 to 1.5 wt% (dry basis), which yields SO_2 levels in the untreated engine exhaust gas of about 210 to 450 ppm at 11% O_2 (1.0 to 2.1 lb/MMBtu). We are conservatively using the NSPS for coal-fired utility boilers as a guideline and the overall required NSPS reduction for SO_2 is currently 90 or 70%, depending on the uncontrolled emission level. Considering the low sulfur content of engine-grade CWS and the relatively small powerplant capacity of expected engine installations, 70% reduction of SO_2 in the exhaust gas has been chosen as a reasonable target.

NO_x: Measured emissions of NO_x from the coal-fueled engine are about half those of conventional diesel engines, due in part to the flame temperature suppression effect of water in the slurry. Measured coal-fueled diesel NO_x emission levels of 600 $\pm$200 ppm at 11% O_2 (1.8 $\pm$ 0.6 lb/MMBtu) must be significantly reduced to make the coal-fueled diesel engine commercially viable. For example, a reduction of 50 to

75% would be necessary to meet the NSPS coal-fired utility boiler standard of 0.6 lb/MMBtu. However, recognizing that state and local regulations are often more stringent, and that future NSPS may tighten to the level of low-NO_x burners (0.3 lb/MMBtu), the NO_x control system was designed to achieve 85% reduction (to 0.25 lb/MMBtu). Furthermore, it incorporated Selective Catalytic Reduction (SCR), a control method considered Best Available Control Technology (BACT) by many regulatory agencies.

CO and Unburned Hydrocarbon Emissions: The combustion characteristics of the CWS fuel in the Cooper-Bessemer engine have been excellent. Carbon monoxide and unburned hydrocarbon emissions are low, in the ranges of 100-300 ppm and 20-200 ppm, respectively. As a result, control methods for these pollutants are not necessary.

2. Overview of Emission Control Activities

Phase I

The initial phase of this program was devoted to characterizing engine emissions and evaluating the economic and technical merit of a wide range of emission control options. Generally, two criteria were used to screen candidate technologies: probability for technical success and potential for minimizing life-cycle cost.

Phase II

The second phase of this program involved the final selection of the emission control technologies to be incorporated into the 1.8 MW_e coal-fueled engine demonstration, conceptual and detailed design of the ECS, system construction, start-up, and component testing. During this time the ECS components were thoroughly tested individually and while operating as a system.

Phase III

The final phase of this program demonstrated sustained operation of the coal-fueled diesel and ECS over a 100-hr run. While the focus of this test phase was engine operability, the ability of the ECS to meet emission performance goals during a sustained 100-hr run was proven. While this period of time was not sufficient to thoroughly evaluate the long term performance of all components (e.g., SCR catalyst life) it provided the opportunity to evaluate the controllability and effectiveness of all of these technologies when used as a system with the coal-fueled diesel.

3. Selection of Methods for Controlling Emissions

Method of Controlling Particulate Matter: Particulate matter in the coal diesel exhaust gas is of concern for two reasons. First, relatively large particles (over 10 micron in diameter) can erode the engine's turbocharger, causing performance and maintenance problems. Therefore, a cyclone separator was located upstream of the turbocharger. Second, particulate matter controls must be implemented to attain accepted emissions standards. Here a conventional bag filter was selected as the final particulate removal device because it is capable of high collection efficiencies (over 99%) and can be used in combination with duct injection SO_2 control technologies.

Method of Controlling NO_x: The NO_x control approach employs a combination of in-cylinder combustion modification (25 to 50% reduction to 0.8 to 1.2 lb/MMBtu) and post-combustion treatment (80 to 90% reduction to 0.08 to 0.25 lb/MMBtu). The post-combustion NO_x control methods evaluated in detail during Phase I of this program are summarized in Table V-1. SCR is a commercially available method that satisfies performance targets. Three technologies, reburning, Selective Non-Catalytic Reduction (SNR) at 750 to 850°F, and NO reduction on engine exhaust particulate, were evaluated further in laboratory research programs because they offered the potential for much lower cost than SCR, but none proved readily feasible.

Table V-1. NO_x Control Options (Post-Combustion Treatment)

Process	Percent Removal	Potential Advantages	Assessment
1. SCR (Selected)	80-90	• Commercially available	• Technically feasible • Selected for C-B coal engine • Low cost designs emphasized
2. Reburning	less than 60	• Inexpensive compared to SCR • Can recover heat	• Tests showed not feasible • Temperature and stoichiometry result in high fuel consumption
3. SNR Process at 750 to 850°F	Not demonstrated	• No catalyst needed • Inexpensive compared to SCR	• Tests showed not feasible • SO_2 interference • By-products not fully determined
4. NO Reduction on Engine Particulate	Not demonstrated	• No catalyst, no NH_3 • Consume particulate • Inexpensive compared to SCR	• Tests showed not feasible as single method. • Low NO_x reduction at expected particulate loading

Method of Controlling SO_2: Table V-2 lists those SO_2 control technologies that advanced for detailed evaluation: spray drying with hydrated lime sorbent, duct injection of calcium-based sorbents, and duct injection of sodium-based sorbent. To compare the economic impact of the SO_x control options, the operating costs of systems as applied to the coal-fueled diesel engine were estimated. Table V-2 compares estimates of levelized busbar and initial capital costs of FGD processes for a 12 MW_e engine system. Clearly, duct injection options offer significant economic benefits over spray driers.

Dry sodium injection systems have several advantages relative to calcium duct injection systems for application to the coal-fueled diesel engine. First, the capital cost of the sodium-based systems is lower. Second, injection of sodium sorbents is a more mature technology. Commercial installations are now in the field whereas the

Table V-2. SO_2 Control Options

Process	Percent Removal	Levelized Busbar Cost (mils/kW-hr)	Initial Capital Costs ($/kW)	Assessment
1. $NaHCO_3$ injection at 300°F (selected)	50 - 80	8	100	• Proven technology that meets control target • Disposal of residue more costly than others
2. Calcium-based sorbent injection at 300°F	40 - 70	7	111	• Humidification required • Difficult to implement on small systems (5 to 20 MW)
3. Spray dryer at 150°F	over 70	18	378	• Proven technology • High capital cost

first large-scale demonstrations of calcium sorbent duct injection are currently taking place. These demonstrations have indicated that the reliability of calcium injection/humidification systems is not high, especially when injecting into smaller ducts while operating near the dewpoint of the flue gas, as is required to approach 70% SO_2 reduction. Finally, injection of sodium sorbents has been shown to remove 10 to 20% of NO_x in addition to SO_2.

Spent calcium and sodium based sorbents are both non-hazardous and can be disposed of in landfills. The spent sodium sorbent, however, is highly leachable. Liners and leachate collection may be required for landfills used for spent sodium sorbent, making disposal of this material more expensive than for spent calcium sorbent. In the long run, sale of the spent sorbent as a by-product may be implemented.

Based on the analysis as described above, dry sodium injection was selected as a low cost method for the control of SO_2 on the coal-fueled diesel engine. The low operating cost and relative maturity of this technology will ensure that high levels of SO_2 reduction will be demonstrated within the timeframe of engine commercialization.

4. Integrated Coal Diesel Emissions Control System (ECS)

The ECS designed for Cooper-Bessemer's 1.8 MW_e coal-fueled engine is comprised of the following eight subsystems: in-cylinder NO_x reduction, cyclone, SCR reactor, heat exchanger, sorbent injection, baghouse, induced draft (ID) fan, and flue gas sample conditioning and analysis. Figure V-1 provides a layout of the ECS, while Figure V-2 shows the installed system at Cooper-Bessemer's Mt. Vernon, Ohio engine laboratory. In operation, exhaust gas from the engine first enters the cyclone where relatively large particulate matter is removed. Gas exiting the cyclone goes to the engine turbocharger where the temperature and pressure are reduced to about 800-850°F and 20 in. w.c., respectively. At this point the flue gas can be directed either to the ECS bypass stack or to the ECS. The first subsystem in the ECS is the SCR reactor where NO_x is reduced by about 80%. Then the gas enters a water-cooled heat exchanger which reduces the gas temperature from 800-850 to 350°F, simulating a heat recovery steam generator. After the heat exchanger, sorbent is injected into the flue gas in a mixing venturi, reducing SO_2 by about 70%. The exhaust gas and sorbent mixture enters the baghouse where the sorbent is removed from the flue gas. After the baghouse the clean exhaust gas flows through the ID fan and to the stack. The ECS control room is located central to the major components of the ECS and contains the flue gas analysis system, data-logger and control panels for the ECS subsystems. From this room operators can control and monitor the performance of all of the subsystems in the ECS.

Major components of the ECS are discussed below.

Cyclone. The cyclone is designed to remove about 90% of particles having diameters of 20 μm and about 50% of the 5 μm diameter particles while operating the engine with all cylinders firing CWS fuel. The low pressure loss (about 6 in.w.c.) across the cyclone ensures minimal impact on turbocharger and engine performance. Cleaned gases exit from the top of the cyclone to the turbocharger, while the solids exit the bottom of the cyclone through a rotary valve.

Selective Catalytic NO_x Reduction System. The SCR system (Figure V-3) reduces the concentration of NO and NO_2 in the exhaust gas by reaction with ammonia over a ceramic zeolite catalyst. Anhydrous ammonia for the system was stored on site as a liquid in a 500 gal tank at 50 to 175 psig. Ammonia vapor was drawn off the tank, reduced in pressure, and then injected into the exhaust gas just upstream of the SCR catalyst. The mixture of ammonia and flue gas entered the reactor from the top at 800° to 850°F, flowed down through the catalyst and exited at the bottom. Catalyst space velocity was about 6800 hr^{-1} at full engine load. With an inlet NO_x concentration of 500 ppm, about 11 lb/hr (3.9 scfm) of ammonia was required for 80% NO_x reduction at full engine load.

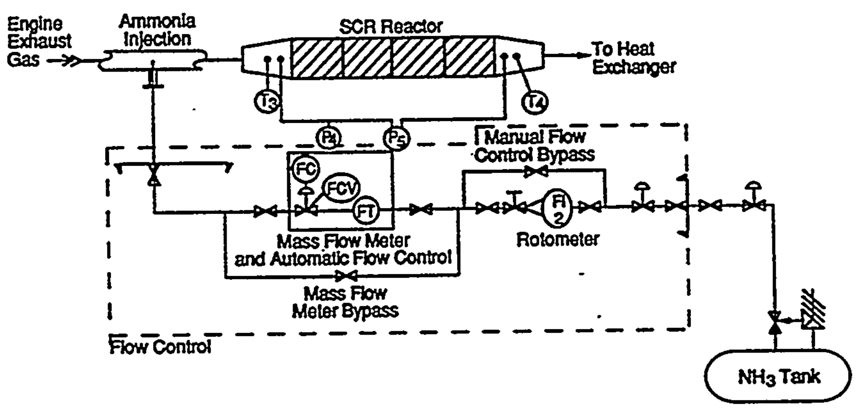


Figure V-3. Selective Catalytic Reduction Subsystem

Sorbent Injection System. The sorbent injection system (Figure V-4) reduces the concentration of SO_2 in the flue gas by reacting SO_2 with sodium bicarbonate ($NaHCO_3$) particles. Sodium bicarbonate sorbent, supplied from the bag dump hopper through a rotary valve, was entrained in air supplied by a separate blower and carried to the mixing venturi. Exhaust gas at about 350°F entered the venturi where the flow converges into the throat creating a high velocity mixing zone. Sorbent was injected into the exhaust gas in this region. The flow then expands and enters the baghouse.

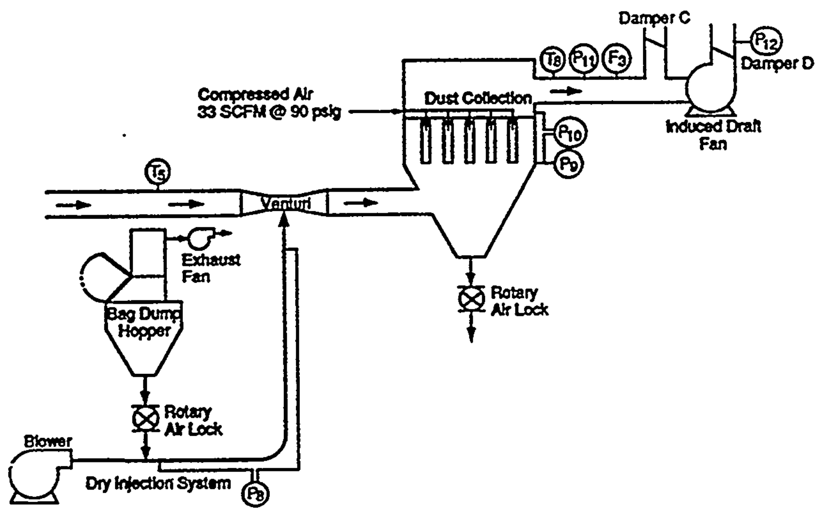


Figure V-4. Sorbent Injection and Baghouse Subsystems

Baghouse. The baghouse (Figure V-4) separates ash and sorbent particles from the exhaust gas, and provides additional contacting time for removal of SO_2 from the flue gas by the sorbent. Exhaust gas entered the baghouse plenum beneath the filter bags. Gas flowed upward, through the bags, and into the outlet plenum. As the gas flowed through the filter bags, particulate was collected on the outside of the bags. The bags were periodically pulsed with air, causing the particles fall into a hopper below the bags. Ash was continuously withdrawn from the hopper through a rotary valve and discharged into a drum for disposal.

5. Gas Analysis, Data Collection, and ECS Controls.

The ECS was equipped with a gas conditioning and analysis system for continuously monitoring O_2, SO_2, and NO_x. A data logger produces a local record of all ECS measurements and sends the data to the main data acquisition system. EPA

Method-5 equipment and protocol were used to measure particulates loadings at key positions in the ECS system.

The ECS was also equipped with a programmable logic controller to control normal system operation and to protect the equipment from damage in the event of abnormal operations. A key design criteria for this system was to allow the engine to operate independently of the ECS in the event of an ECS failure and emergency shutdown.

Figure V-5 shows a process and instrumentation diagram for the facility. The facility was instrumented so that the pressure drop and temperature change across every major piece of equipment could be measured. Gas samples could be withdrawn from three locations in the system, so that the NO_x and SO_2 reductions could be measured for each of the emission control devices independently. The first sample point was upstream of the SCR reactor and before NH_3 injection. The second sample point was just upstream of sorbent injection. The final sample point is after the baghouse. All of the sample lines ran back to a common sample conditioning and analysis system.

B. Emission Control System Performance

This section presents the performance of the ECS installed on the prototype CFD engine, while processing flue gas from the engine operating on coal water slurry (CWS). A total of about 160 hours of operation were conducted on CWS. While CWS operation occurred in a series of tests, for clarity, this report presents the performance results as a single effort.

The performance of each emission reduction subsystem is presented in separate sections bearing the name of the subsystem, while the composite system performance is presented in the summary section. Each subsystem section presents the design performance of the subsystem, final diesel oil fired performance (where possible), and the performance during CWS operation. Performance during CWS operation includes the effects of cumulative operation.

1. Overview of ECS Test Series (as part of LSC-6 Engine Testing)

The system shakedown was conducted with the engine firing diesel oil. At the end of this shakedown period, the SCR system was achieving up to 90% NO_x reduction, and the sorbent injection system achieved up to 80% SO_2 capture. Baghouse performance was not measured during shakedown, and the cyclone was not installed during shakedown.

During the winter of 1993, the LSC-6 engine was converted to coal firing. Coal firing began on March 30, 1993. Coal firing was conducted in four test series as follows: a 24-hour test on March 30 through April 1, a 12-hour test on May 26, a 12-hour test on June 17, and a 100-hour test on August 23 through 28 August.

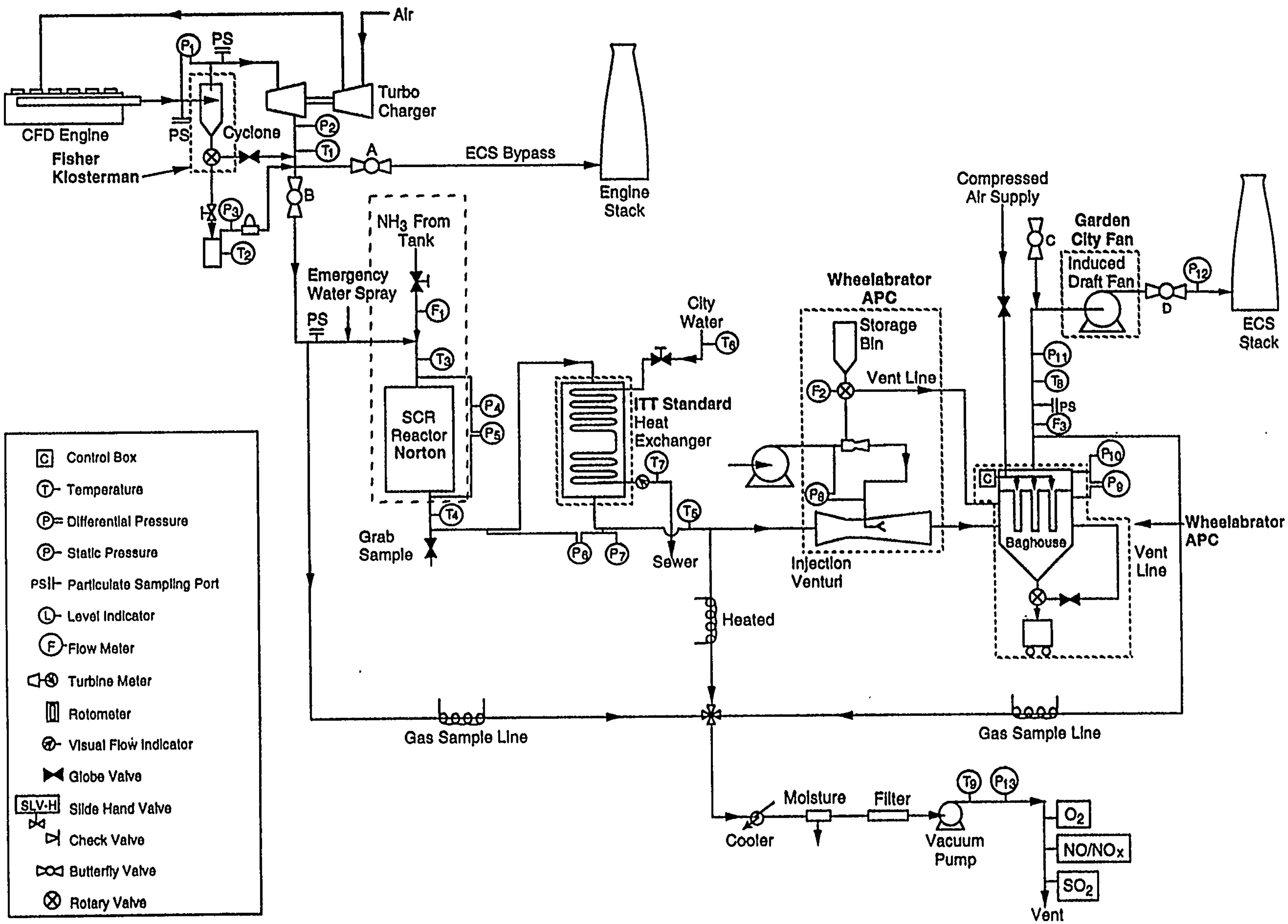


Figure V-5. Emissions Control System Process and Instrument Diagram

Initially, the SCR was able to achieve up to 90% NO_x reduction. However, performance dropped to a steady value of 75 to 80% NO_x reduction. The sorbent-injection performance improved steadily over the course of testing, achieving up to 95% SO_2 capture. The baghouse was able to capture 99.90 to 99.98% of the incoming particulate matter. The final steady state emissions from the system were as follows:

Stack Emission	ECS Performance Vs. Goal (NSPS Std.)
0.35 lb/MBtu NO_x	Actual 90% vs. 80% goal
0.08 lb/MBtu SO_2	Actual 80% vs. 70% goal
0.003 lb/MBtu Particulate matter	Actual 99.9-99.98% vs. 99.5% goal

Both the sorbent injection and baghouse systems exceeded their performance goals by substantial margins. The SCR system exceeded goals initially; but interactions with effluents from coal combustion and operation below the design temperature resulted in below design performance. SCR performance can be achieved by catalyst reformulation, but this identifies the need to accurately know the engine exhaust temperature over the load range. The cyclone is the only system which did not perform adequately. Unfortunately, the cyclone was unable to capture any significant amount of particulate matter, presumably due to the pulsing nature of the flow. It is apparent that a simple cyclone will not suffice for environmental compliance but should be used to protect the turbocharger.

2. Cyclone Performance

In order to protect the turbocharger from larger particles exiting the engine which might erode the turbine blades, a cyclone was installed between the exhaust manifold and the turbocharger inlet. The cyclone was provided by Fisher-Klosterman Inc. This section presents the expected performance of the cyclone and compares this to the measured results in terms of the quantity and nature of the particulate leaving the turbocharger both with and without the cyclone in place.

Cyclone Selection and Expected Performance: The cyclone system is comprised of the cyclone itself and the ash handling equipment. The cyclone was designed to remove 90% of the particles greater than 20 μm diameter, and 50% of the particles greater than 5 μm diameter. Table V-3 shows the expected capture efficiency as a function of particle size. These removal efficiencies assume a steady flow of about 9300 ACFM exhaust gas at 1020°F and 20 psig, and spherical particles with a density of 156 lb/ft^3 (25% ash and 75% carbon). Figure V-6 shows the particle size distribution of particulate matter obtained from pilot-scale coal diesel tests (Benedek, et al., 1989). This is the particle size distribution exiting the exhaust manifold and entering the cyclone. From this distribution, one would expect the particle capture efficiency to be about 33% of the total mass entering the cyclone. Total particulate

Table V-3. Design Cyclone Capture Efficiency (Expected Performance)

Stokes Equivalent Diameter, µm	Expected Collection Efficiency, Weight %
1.0	5.17
2.0	18.08
3.0	30.71
4.0	41.52
5.0	50.42
7.0	63.66
12.0	81.20
20.0	90.12
50.0	97.48
145.0	99.89

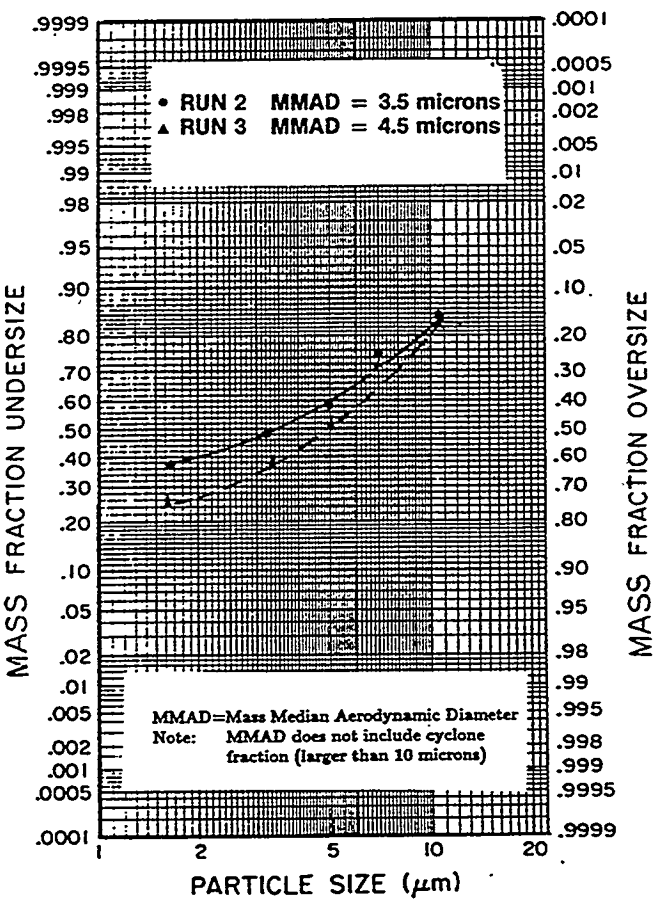


Figure V-6. Particulate Size Distribution loading from the engine is expected to be between 1 and 3 lb/MBtu (Wilson, et al., 1991). Therefore, since the heat input of the engine is about 18 MBtu/h at full load, the mass capture rate of ash by the cyclone should be between 5.3 and 16 lb/h.

Measured Cyclone Performance: During CWS operation, the system was operated both with and without the cyclone in place. The duct work on both the inlet and outlet of the cyclone was prone to fatigue failure of the welds. Despite attempts to make the duct design more rugged, it was not possible to operate with the cyclone in place for extended periods. Fortunately, the turbocharger was a radial inlet design, allowing operation without the cyclone. As a result of operating both with and without the cyclone, it was possible to obtain particulate matter samples from after the turbocharger which were representative of both raw engine particulate matter, and of particulate matter after the larger engine particles had been removed by the cyclone. Table V-4 shows the mass loading and size distribution of the raw engine exhaust. Table V-5 shows the mass loading and size distribution of the engine exhaust with the cyclone in place.

The data on particulate loading and size distribution indicate that the size and quantity of particulate entering the turbocharger was more a function of engine operation than the presence of the cyclone. This is consistent with observations of solids collected in the cyclone. There was never sufficient solids collected in the cyclone to obtain a weight collected or collection rate, indicating a very low efficiency. On 31 March 1993, the particle size distribution of material in the collection drum had a mass mean particle diameter of 74.0 μm, compared to 3.8 μm for the particulate entering the turbocharger. Without the cyclone in place, the mass mean particle diameter entering the cyclone was 2.9 to 3.5 μm.

Clearly, the cyclone did not achieve the design performance. There are two possible reasons for the poor performance. First, the particle characteristics (shape and density) may have been different than anticipated. Second, the pulsing nature of the gas flow may have prevented the formation of the flow patterns in the cyclone necessary for particle separation. There is a pressure pulse with every exhaust stroke, which occurs at a frequency of 20 Hz. Studies of the pressure pulsations from the LSB-6 engine have previously shown pressure spikes of 5 to 10 psi. Subsequent conversations with Fisher-Klosterman confirmed that this level of pulsation can significantly impair cyclone performance. The use of a silencer upstream of the cyclone to remove/smooth the pulsation should be considered for future activity (assuming its size and heat loss can be tolerated).

Configuring the exhaust gas ductwork for the cyclone also proved to be difficult. The pulsing nature of the gas flow combined with the vibration of the engine resulted in the repeated failure of ductwork. The problem can be corrected through improved location of expansion joints and duct supports. However, the particle separation

Table V-4. Engine Particulate Without the Cyclone In Place

	Sample								
	5/5/93 22:25 150 BMEP	5/26/93 10:13 175 BMEP	5/26/93 11:35 200 BMEP	5/26/93 15:15 175 BMEP	6/17/93 175 BMEP	8/23/93 175 BMEP	8/24/93 175 BMEP	8/23/93 175 BMEP	8/24/93 175 BMEP
Mass Loading lb/DSCF	6.34E-5	3.24E-5	2.80E-5	2.93E-5	3.24E-5	2.67E-5	3.88E-5	7.53E-5	10.2E-5
% > 1.2 µm	84.7	86.7	84.7	84.6	-	81.9	77.4	90.0	89.9
% > 2.3 µm	62.3	65.0	59.7	59.3	-	59.9	54.5	77.9	80.1
% > 3.3 µm	50.1	52.0	45.8	45.2	-	45.7	45.2	71.5	74.4
% > 4.6 µm	38.1	39.3	32.7	31.4	-	32.1	32.3	64.6	68.2
% > 5.5 µm	32.4	33.4	26.6	25.2	-	26.1	27.2	60.7	64.6
% > 7.8 µm	21.6	23.1	16.1	14.6	-	16.1	17.9	52.1	56.4
% > 11.0 µm	12.6	14.9	8.1	7.2	-	8.5	10.4	42.0	46.4
% > 22.0 µm	3.1	4.8	0.5	0.6	-	1.7	2.9	20.7	23.5
% > 37.0 µm	0.6	1.0	0.0	0.0	-	0.3	0.6	12.9	9.5

Table V-5. Engine Particulate with the Cyclone In Place

	Sample	
	3/31/93* 23:45 175 BMEP	6/17/93 175 BMEP
Mass Loading lb/DSCF	7.62E-5	3.11E-5
% > 1.2 µm	88.4	-
% > 2.3 µm	68.7	-
% > 3.3 µm	55.6	-
% > 4.6 µm	41.7	-
% > 5.5 µm	34.2	-
% > 7.8 µm	21.8	-
% > 11.0 µm	11.7	-
% > 22.0 µm	2.7	-
% > 37.0 µm	1.0	-

*Note: Injector malfunction on this day produced above-normal carbon emissions.

system remains one of the more challenging engineering design areas of the coal-fueled diesel system.

3. Selective Catalytic NO_x Reduction Reactor Performance

The CFD engine was originally expected to produce about 360 ppm NO_x (1.8 lb/MBtu) (Wilson, et al., 1991).[a] Therefore, post combustion NO_x reductions of 80% would be necessary to achieve NO_x emissions of 0.36 lb/MBtu, a level that is competitive with coal fired boilers equipped with low-NO_x burners. About 89% NO_x reduction is required to achieve 0.2 lb/MBtu NO_x, which is the bench mark for the next generation of coal-fired power systems, such as integrated gasification combined cycle and the systems being developed under DOE's Combustion 2000 program. As a result of the Phase I and Phase II development studies, SCR was selected as the only technology which could reliably achieve >80% NO_x reduction at the temperature conditions available after the exhaust header. This section describes the design performance of the SCR, the performance of the SCR with the engine-firing diesel oil and the performance of the SCR with the engine firing CWS.

SCR Expected Performance: Norton Company supplied an SCR system which utilizes NC-300 catalyst. The system was originally supplied with three rows of 18 in. tall honeycomb catalyst, with a total volume of 68 ft^3. The system was designed to process 34,860 lb/h (19,100 ft^3/min) of gas at 800°F (650°F min, 960°F maximum), resulting in a space velocity of about 16,ho^{-1}. The specified performance was to achieve an 80% reduction in NO_x emissions (from 536 to 107 ppm) at a NH_3:NO_x, mole ratio of about 0.83. The catalyst formulation supplied was designed to achieve >80% reduction between 800° and 900°F. The ammonia slip was expected to be less than 10 ppm. Pressure drop through the reactor was expected to be 3.3 in. w.c. with three catalyst layers, or 4.4 in. w.c. with four catalyst layers.

Measured SCR Performance (Oil Firing): As a result of SCR performance limitations discovered during the shakedown tests, an additional 12 in. tall row of catalyst was added at the top of the reactor. This change increased the catalyst volume to 83.1 ft^3. Gas flows were somewhat lower than anticipated, 26,000 to 31,000 lb/h. Therefore, the space velocity of the gases in the reactor was between 10,300 and 12,300 h^{-1}.

Initial SCR testing was conducted with the engine running on diesel oil, which resulted in engine exhaust NO_x of about 820 to 880 ppm. In order to simulate coal operation, tertiary dibutyl sulfide was added to the oil to produce SO_2 in the exhaust gas at concentrations expected for CWS operation. Figure V-7 shows NO_x reduction

[a]All emissions reported on a dry basis, and corrected to 15% O_2.

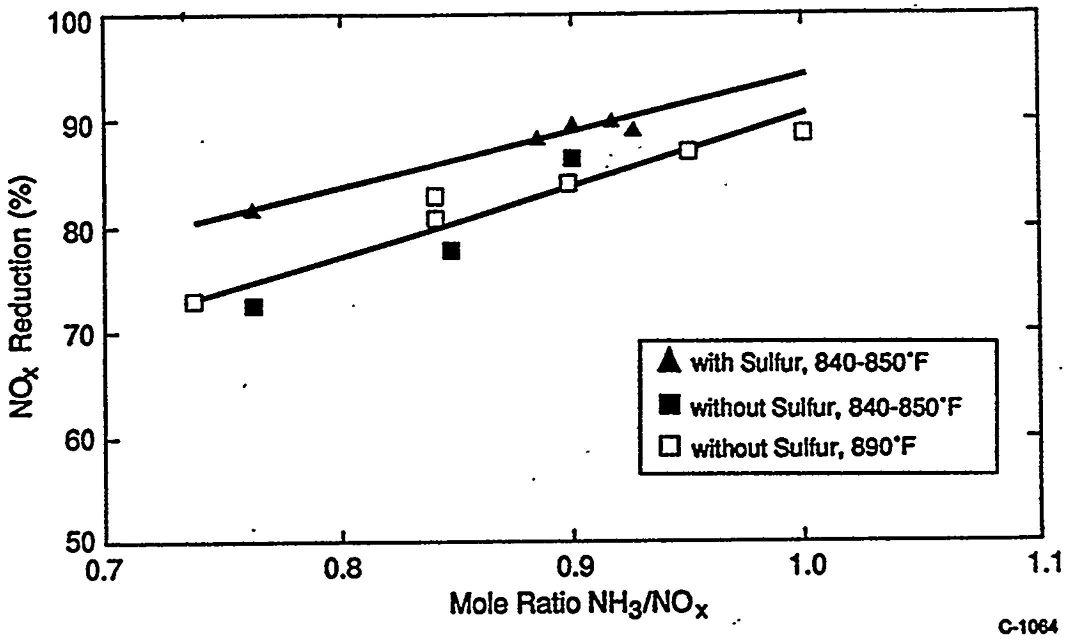


Figure V-7: NO_x Reduction as a Function of NH_3/NO_x Mole Ratio

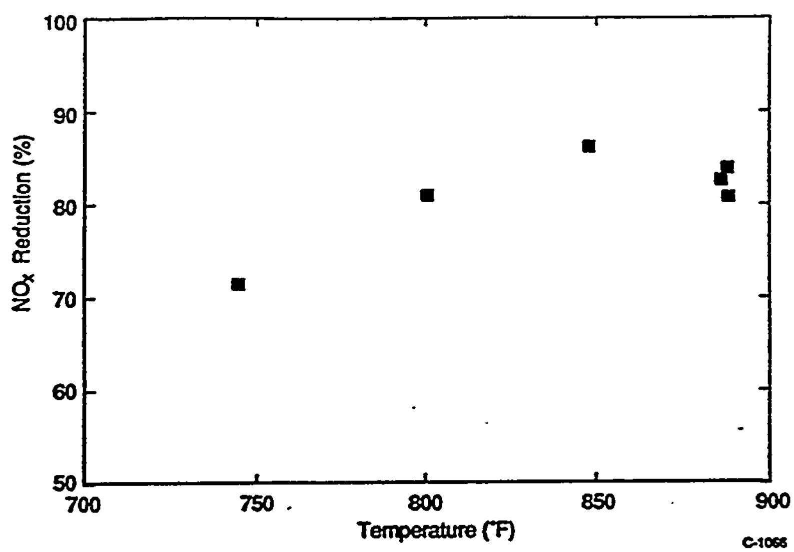


Figure V-8: NO_x Reduction as a Function of SCR Inlet Temperature Without SO_2 in the Flue Gas as a function of NH_3/NO_x mole ratio under these conditions. Acceptable SCR performance (greater than 80% NO_x reduction) was achieved at mole ratios greater than 0.84. NH_3 utilization during these tests was 0.89 to 0.95. The effect of temperature on NO_x reduction is shown in Figure V-8. The temperature range over which greater than 80% NO_x reduction could be achieved was 800° to 900°F.

There was an unexpected benefit due to the presence of SO_2 in the exhaust gas as shown in Figure V-7. About 10% of the engine exhaust NO_x was NO_2. Without SO_2 in the exhaust gas, the NO_2 appeared to pass through the SCR reactor unchanged. With SO_2 in the exhaust gas, the NO_2 at the outlet of the SCR reactor was near zero, and NO_x reduction increased by about 6%.

Performance of the SCR with Coal Fuel Operation: During coal operation, the performance of the SCR reactor can be affected both by the presence of SO_2 and ash in the exhaust gas. Therefore, it is critical to compare the effects of SCR operating variables before and during coal operation. For constancy, the evaluation of the effects of operating variables were made for tests with the engine load at 175 BMEP to keep gas flow rate and inlet NO_x concentration comparable.

Figure V-9 shows the effect of SCR inlet temperature on NO_x reduction and NH_3 utilization at NH_3/NO_x ratios between 0.87 and 0.93. Comparing this data to the data in Figure V-8, the low temperature threshold for effective NO_x reduction is about 15°F higher during coal operation than during oil operation. This could be due to catalyst aging, or interaction with the SO_2 or ash.

Figure V-10 shows the effect of NH_3/NO_x mole ratio on NO_x reduction, limiting the database to tests when the SCR inlet temperature was greater than 800°F. Figure V-11 shows the same data presented as NH_3 utilization. The results are keyed to identify the test period in which the data was collected. It appears that cleaning the SCR after the June tests may have caused a marginal and temporary increase in NH_3 utilization. Unfortunately, during the second test series (16 May and 17 May 1993) and much of the fifth test series (23 August through 28 August 1993), the engine was run with the turbine bypass closed, so that the SCR inlet temperature fell below the design inlet temperature. NO_x reduction was clearly better during the first test series than during the last test series. Performance appears similar for the latter three test series.

NH_3 slip measurements were made for three SCR performance tests. These measurements were made with SCR inlet temperatures of 769° to 777°F, which is below the SCR operation window. NH_3 slip was 11 ppm at NH_3/NO_x mole ratio equal to 0.88, 18 ppm at NH_3/NO_x mole ratio equal to 0.92, and 30 ppm at NH_3/NO_x mole ratio equal to 1.04. The last point is clearly outside the operating range of the SCR reactor. However, for the one point within the anticipated range of NH_3/NO_x mole ratio (0.8 to 0.9), the NH_3 slip was close to the design limit of 10 ppm. Since

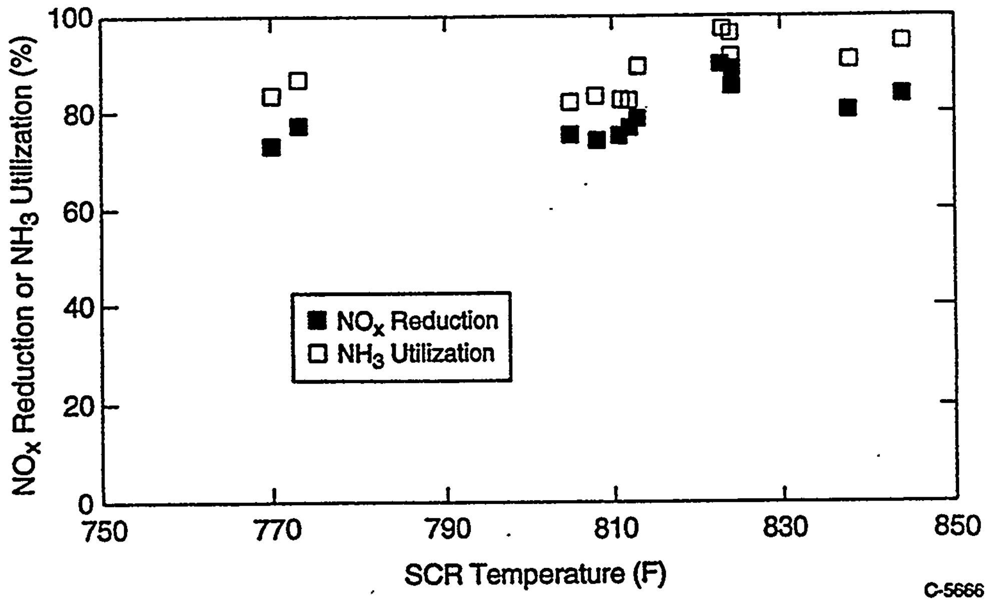


Figure V-9: **SCR Performance as a Function of SCR Inlet Temperature During Coal Operation. Gas Flow 26,700 to 29,170 lb/h, NH_3/NO_x 0.87 and 0.93**

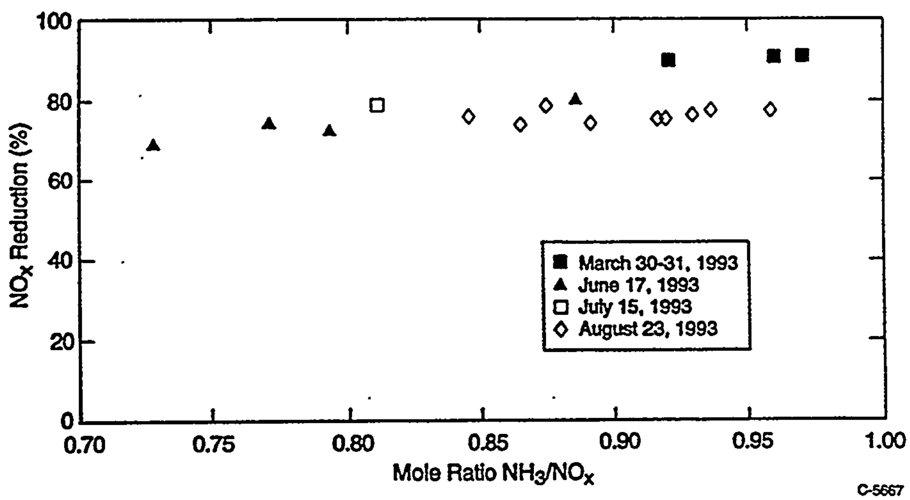


Figure V-10: **SCR NO_x Reduction as a Function of NH_3/NO_x Mole Ratios at 175 BMEP, SCR Inlet Temperature 800°F**

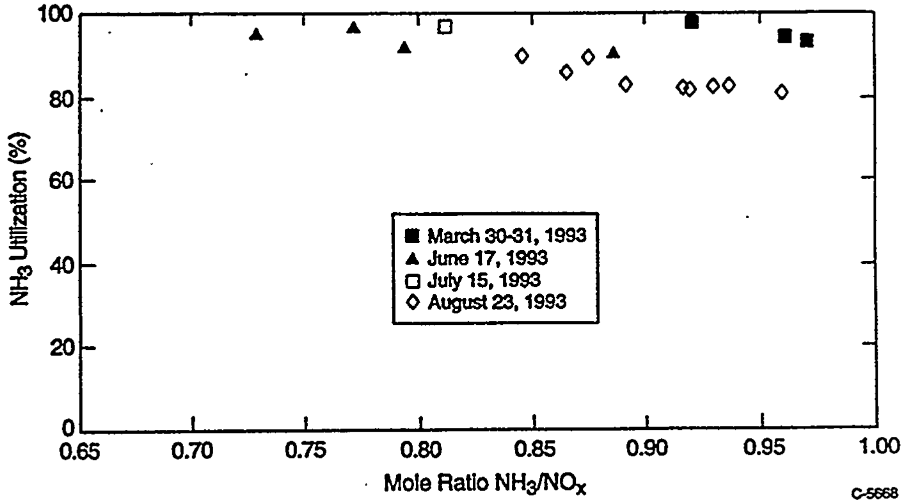


Figure V-11: SCR Ammonia Utilization as a Function of NHc/NOx Mole Ratio 175 BMEP. SCR Inlet Temperature 800° to 830°F

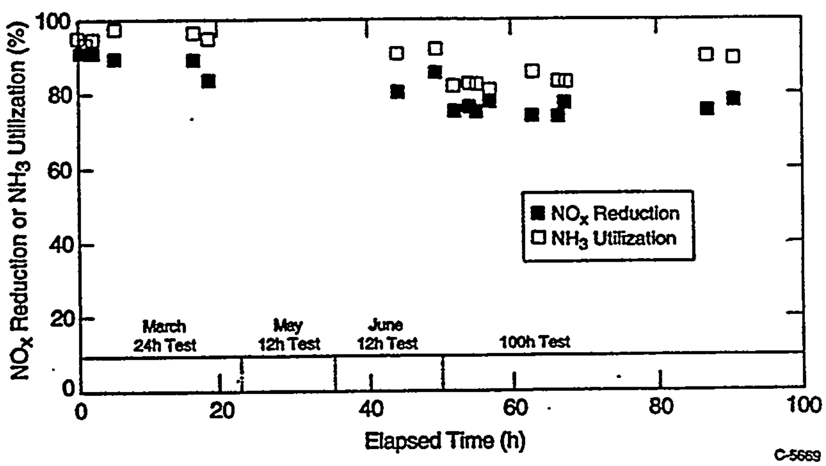


Figure V-12: SCR Performance During Coal Operation. Gas Flow 26,700 to 29,170 lb/h SCR Inlet Temperature 800° to 844°F, NH_3/NO 0.85 to 0.95

NH_3 utilization increases from about 83% at 770°F to about 95% at 820°F, NH_3 slip below 10 ppm can be expected when the SCR is operated within the design limits.

Figure V-12 shows SCR performance for a narrow range of conditions as a function of elapsed time processing exhaust gas from CWS operation. The range of conditions is limited to SCR inlet temperature between 800° and 844°F, NH_3/NO_x mole ratio between 0.85 to 0.97, and exhaust gas flow rate between 26,700 and 29170 lb/h. According to Norton Company, inlet NO_x has only a small effect on performance above 50 ppm. This allows the use of data from different engine loads which have similar gas flow rates (space velocity), but different NO_x concentrations. This broadens the database in terms of time frame to include data in the 0 to 20-hour period. Figure V-12 shows that NO_x reduction and NH_3 utilization decreased steadily with time until about 50 hours of operation. This degradation in performance could be due to the presence of a stable layer of ash on the catalyst surface or poisoning of some of the catalyst. After 50 hours of operation, performance appears to have stabilized.

Based on the available data, long-term operation of the SCR should provide 75 to 80% NO_x reduction depending on the specific conditions. Additional catalyst surface could bring the system up to design performance. Norton is also able to produce a catalyst able to perform more effectively in the 750° to 850°F range.

4. Sorbent Injection Performance

The CFD engine is expected to have SO_2 emissions of 130 to 270 ppm, based on sulfur concentrations in the coal of 0.7 to 1.5%. Dry sorbent injection of $NaHCO_3$ was selected as the SO_2 control technology which had the highest probability achieving the goal of 70% SO_2 reduction. The sorbent injection and mixing system was supplied by Wheelabrator Environmental. The injection system consists of a sorbent hopper, a blower for pneumatic transport, a rotary air lock between the hopper and transport air for isolation and metering, and a venturi where the sorbent is injected and mixed with the exhaust gas.

Expected Performance of Sorbent Injection System: The system was designed to treat 11,850 ACFM of exhaust gas at 350°F containing 280 ppm SO_2 with 65 lb/h of $NaHCO_3$. Therefore, to achieve >70% SO_2 reduction, the Na_2/SO_2 mole ratio is 1.1.

Measured Performance of Sorbent Injection System (Oil Firing): The initial operation of the ECS was with the engine burning diesel oil. To shakedown the SO_2 capture system, tertiary dibutyl sulfide was added to the fuel for periods of 2 to 4 h. The shakedown tests were all conducted with the SCR in operation. SO_2 and NO_x concentrations entering the sorbent-injection venturi were 280 to 340 ppm and 60 to 160 ppm, respectively.

Figure V-13 shows SO_2 reduction as a function of Na_2/SO_2 at two temperatures. As expected, SO_2 reduction increases with Na_2/SO_2 mole ratio. It also appears that, at low to moderate stoichiometric ratios, SO_2 reduction decreases with increasing temperature.

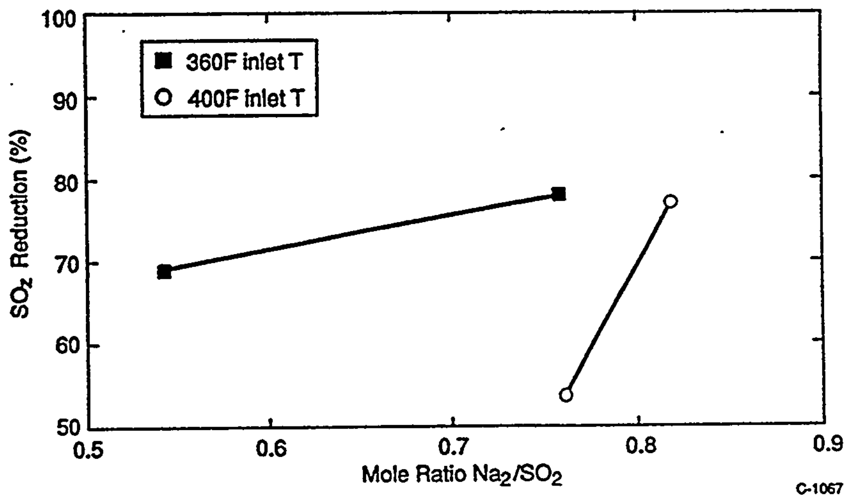


Figure V-13: SO_2 Reduction as a Function of Na_2/SO_2 Mole Ratio at 360° and 400°F

As in previous sorbent-injection studies, 20 to 35% NO_x reduction by $NaHCO_3$ was observed (Coughlin, et al., 1990). Because the adsorption of NO_x uses some of the sorbent, the combined reduction of SO_2 and NO_x was evaluated as a function of combined mole ratio, $Na_2/(SO_2+NO_x)$. Figure V-14 shows combined reduction as a function of combined mole ratio. The behavior is very similar to SO_2 reduction alone.

Measured Performance of Sorbent Injection System (Coal Slurry Tests): During the coal fueled tests, the effectiveness of sorbent injection to control SO_2 and NO_x was affected by the baghouse-inlet temperature and the $Na_2/(SO_2+NO_x)$ mole ratio. Therefore, the effects of each of these operating variables were assessed by limiting the range of the variable not being evaluated.

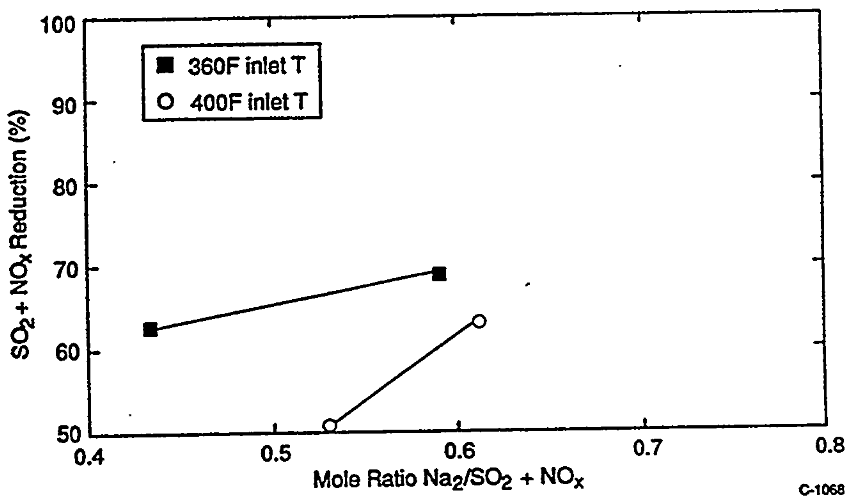


Figure V-14: Combined SO_2 and NO_x Reduction as a Function of $Na_2/(SO_2 + NO_x)$ Mole Ratio

The effect of injection temperature was evaluated over two ranges of $Na_2/(SO_2+NO_x)$ mole ratio. Figure V-15 shows the effect of temperature for $Na_2/(SO_2+NO_x)$ mole ratios between 0.82 and 0.90, while Figure V-16 shows the effect of temperature for $Na_2/(SO_2+NO_x)$ mole ratios between 1.00 and 1.20. SO_2 reduction appears to fall off at temperatures less than about 370°F. This reduced performance is more pronounced for the lower range of mole ratios. NO_x reduction appears to increase with increasing temperature for low mole ratios, and decrease with increasing temperature at higher mole ratios.

The effect of $Na_2/(SO_2+NO_x)$ mole ratio on SO_2 and NO_x reduction was evaluated for injection temperatures between 380° and 430°F. This temperature range was selected as being well above the minimum temperature criteria and below the operating limits of the baghouse. Figure V-17 shows the effect of $Na_2/(SO_2+NO_x)$ mole ratio on SO_2 and NO_x reduction. There is a slight increase in SO_2 reduction with increasing $Na_2/(SO_2+NO_x)$ mole ratio, though even at the lower mole ratios, SO_2 reduction is typically above 80%. There is no apparent effect of $Na_2/(SO_2+NO_x)$ mole ratio on NO_x reduction.

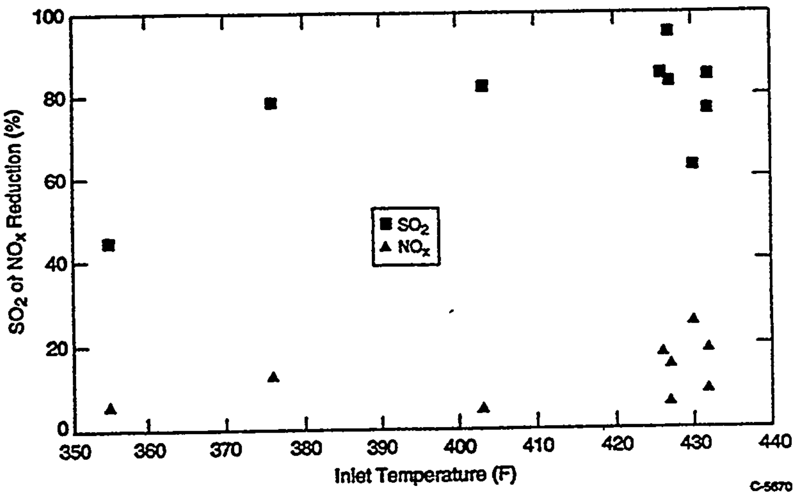


Figure V-15: Sorbent-Injection Performance as a Function of Sorbent-Injection Temperature. Gas Flow 26,700 to 29,170 lb/h, $Na_2/(SO_2+NO_x)$ 0.82 to 0.90

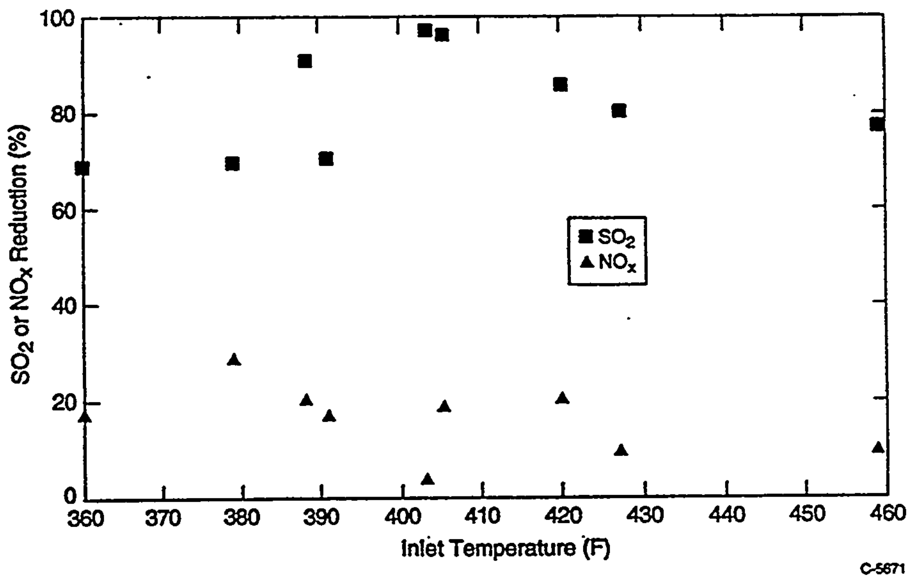


Figure V-16: Sorbent-Injection Performance as a Function of Sorbent-Injection Temperature. Gas Flow 26,700 to 29,170 lb/h, $Na_2/(SO_2+NO_x)$ 1.00 to 1.20

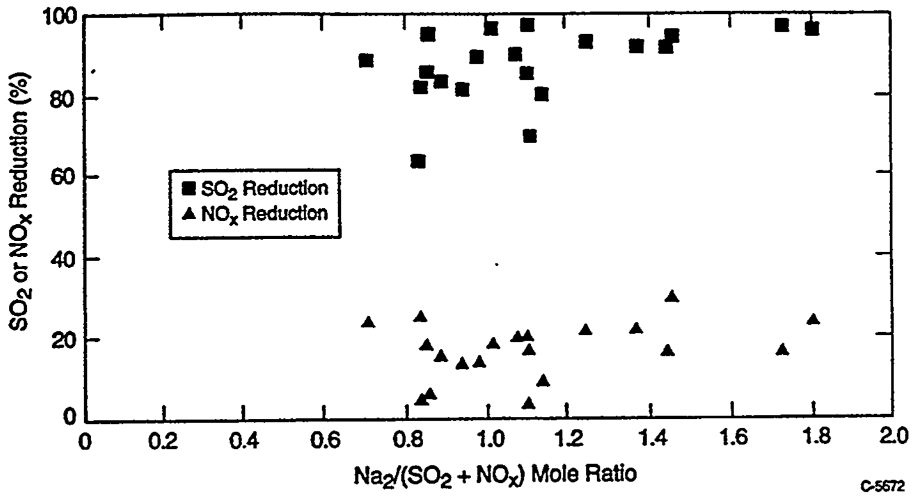


Figure V-17: Sorbent-Injection Performance as a Function of $Na_2/(SO_2+NO_x)$ Mole Ratio. Gas Flow 26,700 to 29,170 lb/h, Sorbent-Injection Temperature 380°to 430°F

Combined SO_2 and NO_x reduction by $NaHCO_3$ is a complex and poorly understood process. NO_x reduction is dependent on SO_2 reduction, and will not occur in the absence of SO_2. Both adsorption processes occur simultaneously with calcination of the sorbent, which is why little reaction occurs below 350°F; and then the level of reduction decreases at high temperatures (>500°F) as the calcination time becomes shorter than the reaction time.

Figure V-18 shows the emissions-reduction performance of sorbent injection over the duration of coal operation. The range of injection temperature was limited to between 380° and 430°F, and $Na_2/(SO_2+NO_x)$ mole ratio to between 1.0 and 1.4. SO_2 reduction seems to have improved early in the program and then leveled off from about 30 h onward. NO_x reduction varied from about 10 to 30%, and seems to have been unaffected by the length of operation.

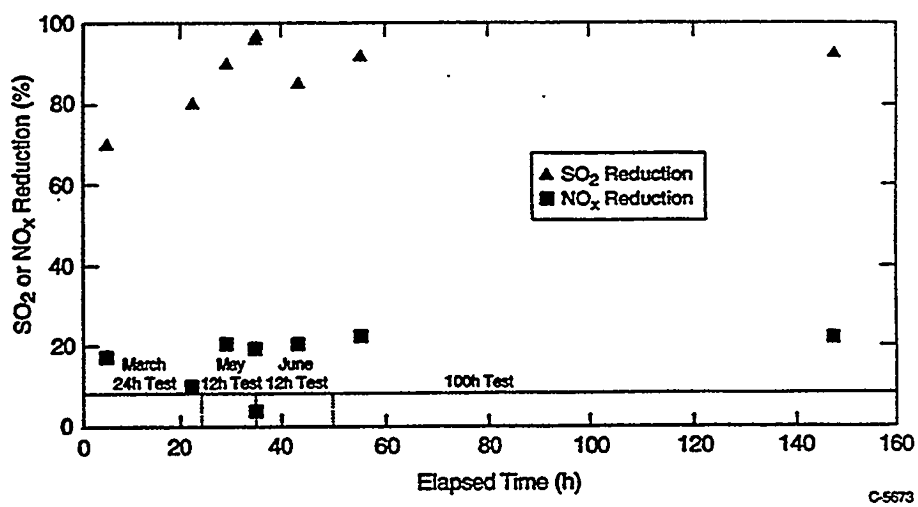


Figure V-18: Sorbent-Injection Performance During Coal Operations. Sorbent-Injection Temperature 380° to 400°F, $Na_2/(SO_2+NO_x)$ 1.00 to 1.40

5. Baghouse Performance

The particulate entering the final particulate clean-up was expected to be between 45 and 66 lb/h depending on the amount generated by the engine. With a gas-flow rate of 7750 SCFM, the loading would be 9.7E-5 to 2.4E-4 lb/DSCF or 2.8 to 4.1 lb/MBtu. Therefore, to meet the requirements of the 1990 Clean Air Act (CAA) of 0.05 lb/MBtu would require 98.2 to 98.7 reduction in particulate matter emissions. A baghouse was determined to be the technology which could most reliably achieve this level of reduction. Furthermore, a baghouse provides additional SO_2 adsorption time as the sorbent resides on the bag surface and the exhaust gas passes through the filter cake.

Expected Baghouse Performance: The baghouse was supplied by Wheelabrator Environmental along with the sorbent-injection system. The baghouse was designed to process 11,850 ACFM of exhaust gas at 350°F, with a maximum operating temperature of 500°F. There are 15 rows of 15 bags; each bag is 6 in. in diameter and 12 ft long. The total cloth area is 3,179 ft^2, providing an air to cloth ratio of 3.73. The baghouse was designed to operate with a pressure drop of 4 to 6 in. w.c.

Measured Baghouse Performance with Coal Fuel Operation: Three Method 5 samples were taken from the stack during the course of testing. In addition, there were three stack particulate matter samples gathered over the course of successive ammonia measurements during the 100-hour test.

Particulate matter entering the baghouse was comprised of engine particulate plus the spent and unreacted $NaH(CO_3)$. The particle loading of the exhaust gas from the engine was about 3.3E-5 lb/DSCF. The sorbent flow rate was about 50 lb/h, which with a median gas flow of 28,000 lb/h results in a sorbent concentration of 1.4E-4 lb/DSCF. Therefore, the total particle loading entering the baghouse was about 1.7E-4 lb/DSCF.

The Method 5 particulate sample results showed particle concentrations of 3.2E-8, 3.3E-8, and 10.7E-8 lb/DSCF in the stack. However, in all cases, though the samples were collected over the course of 1 to 2.4 h, the actual amount of particulate matter collected was negligible (0.001 and 0.0016g). These particle loadings result in a collection efficiency of 99.94% to 99.98% for the baghouse.

The particle samples collected in concert with the ammonia samples were not conducted in a manner suitable for EPA certified testing because the samples were taken from a single point, not a traverse grid. However, the total sample times were 4.6 to 7 h, and measurable amounts of particulate matter were collected (0.0072 to 0.0121g). The particulate loadings for these tests were 7.1E-8, 1.4E-7, and 1.7E-7 lb/DSCF, which correlate to collection efficiencies of 99.96%, 99.92%, and 99.90%. The baghouse clearly achieved the particulate matter capture goals of the program.

6. Conclusions and Recommendations - Emission Control System

During the coal-fueled testing, the system was able to meet all of the emission performance goals. Figure V-19 shows the NO_x reduction for both the SCR and sorbent-injection systems, and the total NO_x reduction by the system during coal operation. The NO_x reduction by the SCR reactor, which provides the majority of the NO_x reduction, definitely diminished through about 50 h of operation. After 50 h of operation, the performance of the SCR reactor steadied out, just below the performance goals of the system. However, NO_x reduction by the sorbent-injection system was relatively constant during the entire coal-fired operation. Within the operating range of the SCR reactor and sorbent-injection systems, the total NO_x reduction was greater than the goal of 80% for all but two test points.

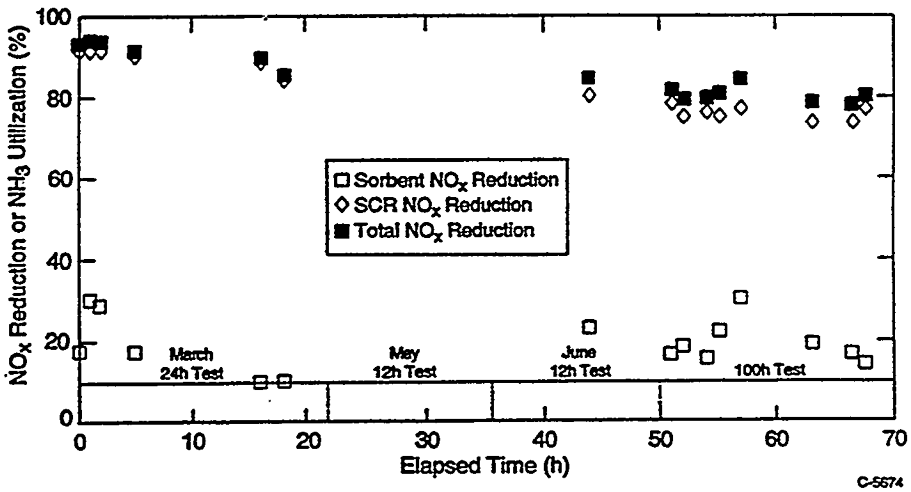


Figure V-19: Emission Control system NOx Reduction During Coal Operation. Gas Flow 26,700 to 29,170 lb/h, SCR Inlet Temperature 800° to 844°F, NH_3/NO_x 0.85 to 0.97, Sorbent-Injection Temperature 390° to 430°F, $Na_2/(SO_2+NO_x)$ 1.00 to 1.40

Figure V-20 shows both NO_x and SO_2 reduction for the system over time for the points where both the SCR and sorbent-injection systems were within their operating limits: SCR inlet temperature 800° to 900 °F and sorbent-injection temperature 370° to 450°F. NO_x reduction started high and gradually fell to just about the lower limit of acceptable operation (80% NO_x reduction). For the design conditions, SO_2 reduction started just below the lower limit (70% SO_2 reduction) and improved to over 90% SO_2 reduction during the course of operation.

During the majority of testing, the engine was run at 175 BMEP. At this load, the heat input was about 15.2 MBtu/h of CWS and 1 MBtu/h of diesel oil for the pilot. The total system NO_x reduction of 79 to 85% NO_x (82% avg.) reduction translates to NO_x emissions of 0.35 lb/MBtu which compares favorably to current pulverized coal technology. Two tests were conducted early in the test series in which the SCR reactor achieved 90 and 91% NO_x reduction. For these two tests, the total system NO_x reductions were 92 and 94%, resulting in NO_x emissions of about 0.14 lb/MBtu, which is below the goal of the DOE-sponsored Combustion 2000 Program (0.2 lb/MBtu). The proposed NO_x emission limit for the NESCAUM region is 0.33 lb/MBtu for sources producing more than 25 tons/yr.

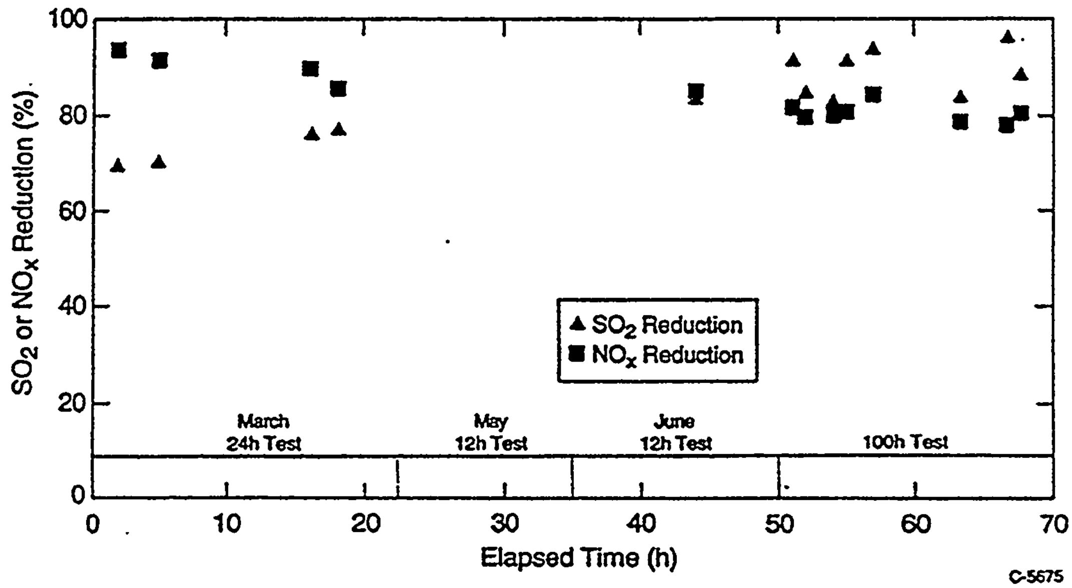


Figure V-20: Overall Emission Control System Performance During Coal Operation. Gas flow 36,700 to 29,170 lb/h, SCR Inlet Temperature 800° to 844°F, NH_3/NO_x 0.85 to 0.97,

The SO_2 reduction for the 175 BMEP tests at design conditions ranged from 70% initially, to between 85 and 95% during the 100-hour test. The initial SO_2 emissions of 0.3 lb/MBtu dropped to a steady state level of 0.08 lb/MBtu. This level of SO_2 emissions was more than an order of magnitude below the 1990 CAAA, Title IV requirement of 1.2 lb/MBtu.

Particulate emissions ranged from 6.5E-4 lb/MBtu during the first 24-hour test to 2.3E-3 and 3.0E-3 lb/MBtu during the 100-hour test. These levels of emissions were about one order of magnitude below the 1990 CAAA goals of 0.05 lb/MBtu.

Several deficiencies associated with operation of the ECS arose primarily due to the intermittent operation of the facility. These include the high temperature ductwork design and the ID fan outlet damper. Both items need to be replaced prior to renewed operation of the pilot facility. In addition, the heat exchanger and baghouse ash removal valve need to be replaced. The heat exchanger was designed with 12 fins/in. on the tubes to extend the heat exchange surface. The fin spacing is appropriate only for relatively clean exhaust gases. Poor cyclone performance and apparent tar and moisture condensation resulted in particulate matter gradually plugging the heat exchanger. The current heat exchanger should be replaced with a bare tube design. Some cost may be saved by using a marginally finned heat exchanger; however, there is a risk that the low cost heat exchanger would have to be replaced with a bare tube heat exchanger. Ash was originally removed from the baghouse via a rotary air lock (RAL) which repeatedly seized. The RAL was eventually removed and not replaced. A manual knife gate valve should be installed to close this opening for personnel protection during ash removal.

There are two system performance deficiencies: the SCR and the cyclone. The gas temperature entering the SCR reactor was, under certain operating conditions, lower than expected, resulting in SCR NO_x reduction marginally below our target. This can be overcome by replacing the catalyst with a lower temperature formulation. Refurbishing cost can be minimized by progressively replacing one row of catalyst at a time, until the performance goals are achieved. The cyclone clearly did not perform well. If capture of relatively large particulate matter is needed to protect the turbocharger, reduction of flow pulsing by use of a silencer should be considered. Or, alternatively, the cyclone must be replaced with either a barrier filter, (e.g., a stainless steel baghouse or ceramic filter) or with a more aggressive aerodynamic separator, such as ADL's spin filter or LSR Technologies' Core Separator. In addition, the ductwork for the pre-turbocharger particle separation must be designed to withstand the system vibrations.

Finally, automatic control of ammonia and sorbent injection has not been demonstrated. While this should be a relatively straightforward implementation of commercial controls, an automatic control system should be installed, and the control algorithm verified.

VI. Commercialization Plan Based on Demonstration Test Results

The commercialization plan for the coal diesel technology (Wilson, et al., 1992) has been updated based on the results of the 100-hour system demonstration test at Cooper-Bessemer. The key practical implications of the tests were as follows:

(1) Test results show that the technology met both the efficiency target and the emissions target, and performance in these areas did not degrade during the 100-hour test. Therefore, efficiency and emissions improvements areas are not on the critical path; straightforward engineering effort can achieve scale-up of the engine and emission system to commercial plant sizes.

(2) Longer run times are needed to estimate useful lifetimes of certain engine components, particularly the useful life of piston rings and exhaust valves. This data on engine components is critical before commercial introduction of the technology. Engineering solutions and material selections are available for durable components, but these solutions must be optimized and demonstrated for several thousand hours, not several hundred hours, as has been accomplished so far.

(3) Thus, the next logical step toward commercialization is a field demonstration program with 5,000-10,000 hours of engine run time on coal fuel. Since this will require four years (a practical field demonstration program will include several lengthy test periods, rather than continuous operation), the implication is that commercial introduction (plant orders) can be targeted in the 2000-2005 timeframe assuming a successful field demonstration. The commercialization plan has been updated to include these steps.

(4) Coal slurry fuel is expected to become competitive in the U.S. with diesel oil and natural gas in the 2000-2005 timeframe, based on energy price projections made by DOE and others. This gives Cooper-Bessemer and other team members the necessary time to optimize and demonstrate the wear solutions for critical hard parts, through a field demonstration program of 5,000-10,000 hours.

(5) Field demonstration opportunities for small coal-diesel plants will be pursued in special situations where clean coal slurry holds a price advantage, such as:

- The DOE CCT-V Demonstration Program (up to 50% cost shared).
- Alaska rural electrification (where diesel oil costs \$4-\$12 per million Btu delivered to certain remote communities).
- China, which has both coal reserves and the need for rapid installation of non-grid power (such as diesels).
- Eastern Europe, which also has coal reserves and is undergoing rebuilding of the electric power infrastructure so as to greatly reduce emissions.

(6) Test experience has shown that the capital cost of the coal diesel plant will not be a barrier to commercialization. The cost of all equipment modules for the plant has been established, and the installed plant cost estimates appear to be competitive:

- $1600/kW for early demonstration plants
- $1300/kW for mature plants

These costs are well below the capital cost of other small coal plants, especially in the NUG market under 50 MW plant size.

(7) Test results have established the coal-water slurry specification, and have proved that a wide range of coals can be utilized to prepare engine-grade slurry. The cost of the slurry will be under $3.00/MMBtu once adequate slurry-demand exists in a given region. The commercialization plan incorporates a series of steps to build up an "infrastructure" for coal-water slurry production and distribution. This is recognized as critical.

In this chapter, the target applications are described and the coal diesel characteristics are compared to other technologies which will be competing in these markets. Then scenarios for penetrating each of the three target market segments are described.

A. Coal Diesel Applications and Basis of Competition

1. Commercial Coal Diesel Plant Configurations

The Clean Coal Diesel Plant of the future is targeted for the 10-100 MW non-utility generation (NUG) and small utility markets, including independent power producers (IPP) and cogeneration. A family of plant designs will be offered using the Cooper-Bessemer 3.8 MW and 6.3 MW Model LS engines as building blocks. In addition, larger plants will be configured using certain engine models in the 10-25 MW class (Cooper-Bessemer will license the technology to other large bore stationary engine manufacturers). For example:

Using Cooper-Bessemer Engines:

8 MW Plant	2 x 3.8 MW (12 cyl) plus 0.8 MW bottoming cycle (b.c.)
14 MW Plant	2 x 6.3 MW (20 cyl) plus 1.4 MW bottoming cycle
21 MW Plant	3 x 6.3 MW with b.c.
28 MW Plant	4 x 6.3 MW with b.c.
42 MW Plant	6 x 6.3 MW with b.c.

Using Larger 14 MW and 25 MW Engines (from a Licensee):

45 MW Plant 3 x 14 MW with b.c.
61 MW Plant 4 x 14 MW with b.c.
92 MW Plant 6 x 14 MW with b.c.
110 MW Plant 4 x 25 MW with b.c.
165 MW Plant 6 x 25 MW with b.c.

As shown in the list above, it is quite realistic to design and build larger clean coal diesel plants in the 50-165 MW capacity range. In fact, the 14-25 MW class diesel engines offer a fuel savings advantage over the smaller engines (typically 45% vs. 40% simple cycle efficiency (LHV)). While a plant can be built as small as 2 MW (based on the Cooper-Bessemer Model LS-6 engine), our cost projections indicate that an 8 MW plant is likely to be at the lower end of what is economically attractive. It should also be noted that the coal diesel plant also can be configured for cogeneration applications.

The reciprocating engine offers a remarkable degree of flexibility in selecting plant capacity. This flexibility exists because the engines are modular in every sense. Scale-up is accomplished simply by adding cylinders (e.g., 20 vs. 16) or by adding engines (4 vs. 3). There is no scale-up of the basic cylinder size. Thus, there is essentially no technical development needed to scale-up the Cooper-Bessemer Clean Coal Diesel Technology all the way from 2 MW (one 6-cylinder engine such as the LSC-6 engine which has been tested) to 48 MW (eight 20-cylinder engines), other than engineering adaptation of the turbocharger and other subsystems to match the engine.

The emissions control system for the commercial coal diesel plant will be very similar to that which has been tested successfully:

- Cyclone separators will remove large particulate upstream of the turbocharger.
- NO_x control will be achieved by combustion optimization, selective catalytic reduction and by reduction across the duct injection system.
- SO_2 control will be achieved by duct injection of sodium bicarbonate followed by sorbent separation in a fabric filter.
- Final particulate control will be achieved by use of a fabric filter.

Use of advanced diesel engines which operate at high brake mean effective pressures, suitably converted for coal-water slurry firing, in future commercial coal diesel plants can lead to combined cycle generating efficiencies of 50 percent.

2. *Key Performance Characteristics of the Clean Coal Diesel*

The Clean Coal Diesel will offer the following performance characteristics in its mature configuration beginning in the 2005-2010 timeframe:

- Installed cost $1300/kW (cost estimate for CCT-V: $1600/kW)
- Efficiency 48.2% (LHV) (demonstrated: 41% - LHV)
- NO_x emissions 0.11 lb/MMBtu (demonstrated: 0.18 lb/MMBtu)
- SO_x emissions 0.37 lb/MMBtu (equivalent to emissions from 0.3% sulfur diesel fuel oil)
- Particulate emissions 0.01 lb/MMBtu

3. *Basis of Competition*

The advantage of this 10-100 MW clean coal diesel technology is that it is targeted for non-utility generation (NUG) and small utility capacities, whereas all other clean coal technologies have been designed for the central station utility market (generally 200-500 MW):

• IGCC	200-500 MW
• PFBC	100-300 MW
• Fuel Cell with Integrated Gasification	200-500 MW

Fuel cell technology is under rapid development and, although initially more costly, fuel cell power plants using natural gas reformers will eventually compete in the 100 kW-10 MW range (below the target size range of coal-diesel plants). Figure VI-1 illustrates the unique market position of the clean coal diesel with respect to its competitors.

In the early market introduction (2000-2010), the clean coal diesel will compete against both natural gas technologies and small coal plants (10-100 MW) of the PC, AFB, and stoker variety.

a. *Clean Coal Diesels Will Be Competing Primarily With Gas Turbine and Reciprocating I.C. Engines In Small Utility, IPP, and Cogeneration Markets.*

In power generation applications, the new clean coal diesel technology must compete for markets with a number of established and emerging power generation technologies. In general, these technology options will include natural gas/diesel fuel-fired reciprocating engines (IC's), gas turbines, and advanced coal fluid-bed boilers with steam turbines. In many cases, the alternative power generation technologies are benefitting from substantial commitments of both industry and government technology development funding (for example, advanced gas turbines, fuel cells, and integrated gasification combined cycle power plants).

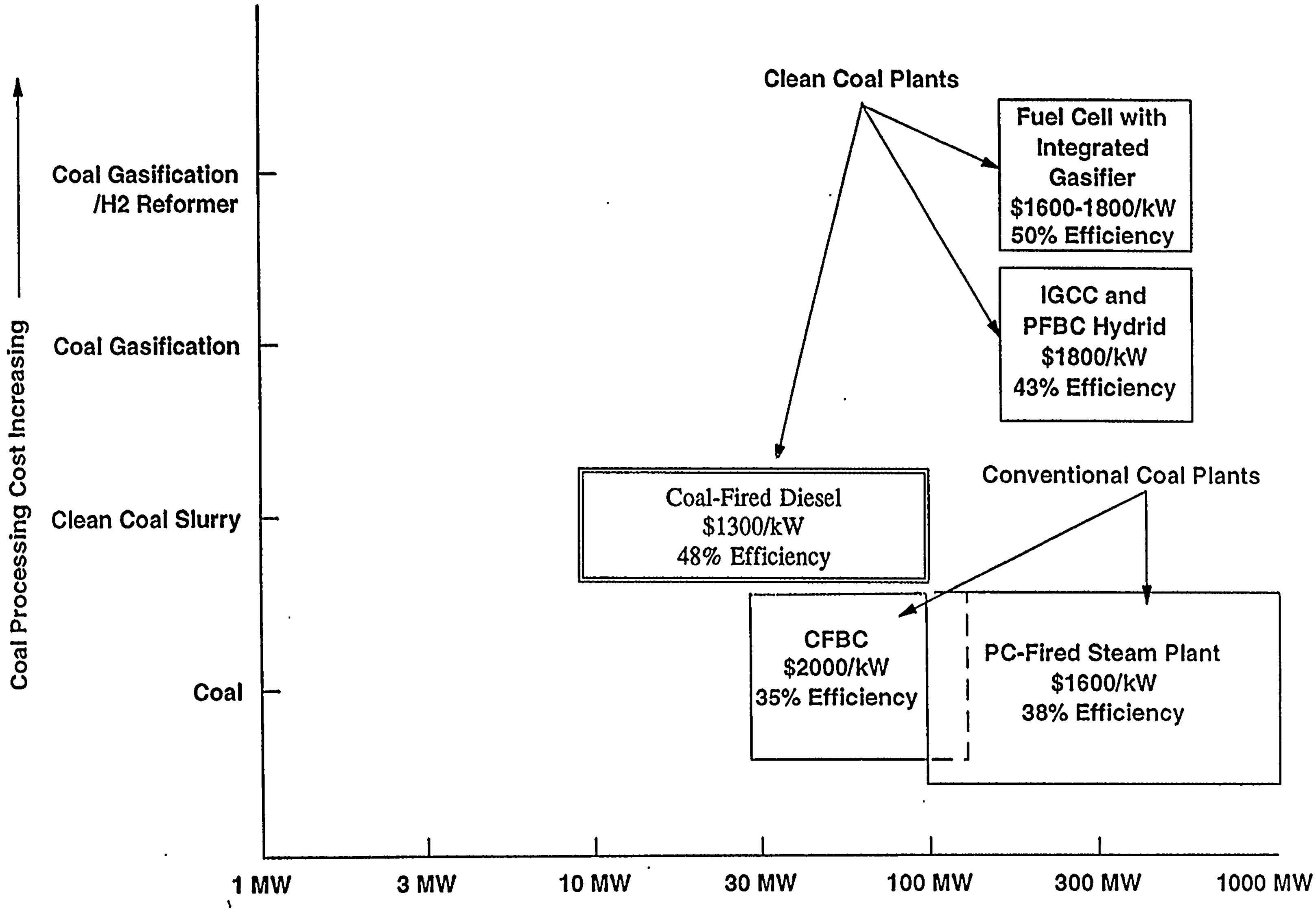


Figure VI-1. Coal-Fueled Diesel is an Attractive Option for a 10-100 MW Power Plant

In considering the potential market opportunity for clean coal diesels, it is necessary to recognize the large variations in application requirements. In particular, it is expected that system capacity will strongly influence technology selections.

b. N.U.G. and Smaller Utility Applications (10-100 MW)

Power generation needs will increasingly be met by cogeneration and distributed power (IPP) systems having typical capacities in the range 10 MW to 100 MW. The principal technologies against which clean coal diesels will compete in these applications over the coming decade will be natural gas reciprocating (IC) engines and gas turbines. Advanced fluid bed coal units (CFBC) will also compete in the 50-100 MW range. Also, natural gas fuel cells appear to be on a rapid development track and will emerge as a competitor in the 100 kW to 10 MW range until natural gas prices increase. As this section of the proposal will explain, the coal diesel primary target market is the 10-100 MW sector.

c. Large-Scale Utility Applications (100 MW to 500 MW)

In larger scale applications (greater than 100 MW), the clean coal diesel technology cannot effectively compete with gas turbine combined cycle (GTCC) plants and high efficiency coal-based technologies. The largest diesel engines assembled today (Sulzer RTA84M and MAN B&W K90ME) are in the 45 MW to 50 MW range, so 150 MW is near the upper limit of reasonable plant size.

In the near-to-medium term, GTCCs will be the technology of choice in the 100 MW and higher size range power plant when using natural gas as a fuel. This is because (a) natural gas price projections are favorable, (b) GTCC units can be expected to exhibit efficiency levels approaching 50% (HHV basis) by the mid-to-late 1990s, and (c) GTCC units have low emission levels (< 20 ppm NO_x). By late in the decade, if gas prices continue to increase, advanced coal technologies will become increasingly competitive for large base load power stations. These technologies include fuel cell with integrated gasification, pressurized fluid beds, integrated gasification combined cycle, and ultra-supercritical pulverized coal. Therefore, to avoid competing with these advanced clean coal technologies in the 100-500 MW market, we have targeted the 10-100 MW market for the clean coal diesel.

4. The Efficiency and Emission Characteristics of the Clean Coal Diesel Technology are Highly Favorable Compared to Alternatives in the Under 100 MW Power Capacity Range

As indicated in Figure VI-2, the efficiency of conventional power system technologies tends to decrease as the size of the power generation unit decreases. However, the diesel engine efficiency is relatively constant from 10 MW to 100 MW plant size. This effect is expected to enhance the competitive advantage for clean coal diesels in smaller applications. Clean coal diesel combined cycle plants with simple bottoming cycles are expected to have overall cycle efficiencies of 48-50%. Plants with advanced diesels and more sophisticated bottoming cycles are expected to have cycle efficiencies in the range of 53-55%.

As indicated in Table VI-1, the clean coal diesel technologies have remarkably low emissions of regulated emissions, including NO_x. For example, NO_x emissions from clean coal diesels are comparable to those from highly controlled gas turbines and are less than 40% of those from I.C. engines utilizing advanced emission control equipment.

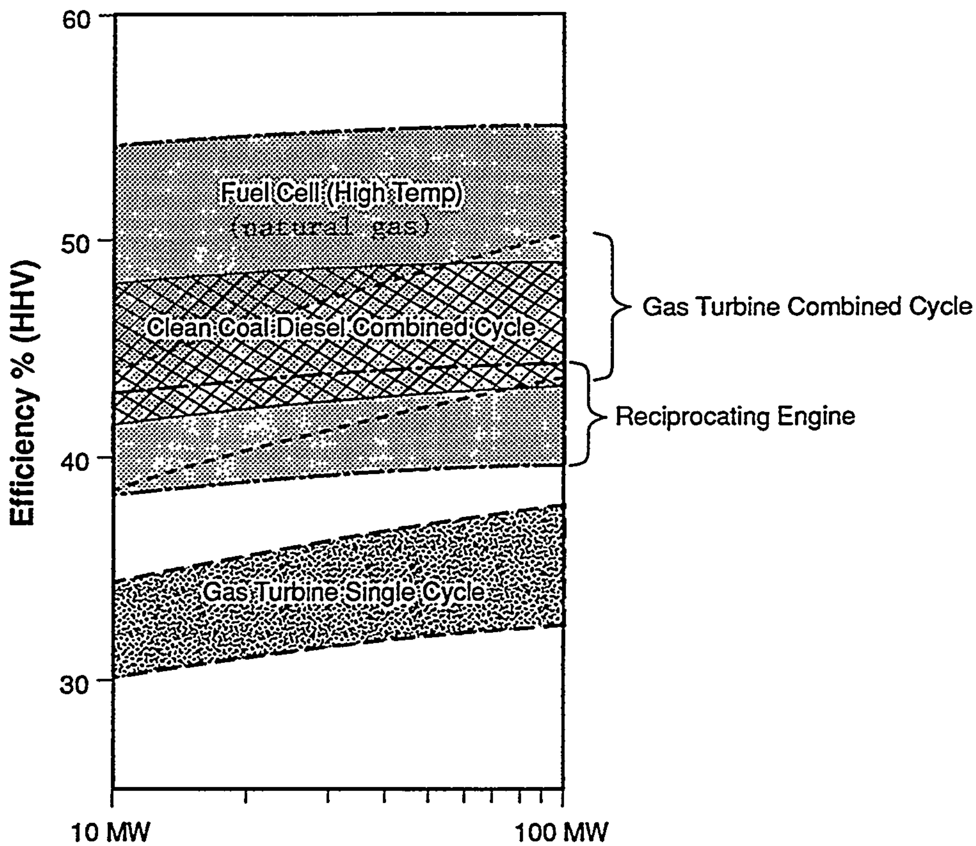
Table VI-1. Comparison of Coal Diesel NO_x Emissions with Conventional Alternatives

Technology	NO_x Emission @ 15% O_2		
	Uncontrolled	Controlled	Control Technologies
Clean Coal Diesel	200-400 ppm	20-40 ppm	SCR system
I.C. Engine (natural gas or 2 D fuel oil)	500-1500 ppm	50-150 ppm 50-150 ppm	Lean burn SCR system
Gas Turbines	120 ppm	30-60 ppm 20-40 ppm 10-20 ppm	Steam injection Lean premise SCR system
Fuel Cells (natural gas reformer)	5 ppm	5 ppm	None
CFBC	NA	100-200 ppm	Low fluid bed temperature

5. *The Capital Costs of Clean Coal Diesel Systems are Favorable Below 100 MW*

It is recognized that in the post-2000 period when gas prices rise to double the cost of coal (or more), the highly favorable efficiency and low NO_x attributes of clean coal technology will allow for some modest level of capital cost premium as compared to alternatives. However, large market penetration in mature configurations will require that clean coal diesel system installed cost, at least, be close to that of new plants based on natural gas-fired reciprocating engines or gas turbines.

The first demonstration coal diesel plants with CWS fuel preparation plants will be installed for an estimated $1600/kW, complete with building, stack auxiliary equipment, and emission control system. For comparison, conventional diesel plants are currently being installed at a total cost of $1100/kW. However, once modest production levels are achieved, the cost of clean coal diesels in mature configurations are projected to be about $1300/kW in the 10-100 MW plant capacity range, which is the focus of current commercialization efforts.



Note: Ranges reflect the impact of multiple technologies and stages of development.

Figure VI-2. Comparison of Clean Coal Diesel Efficiency Levels with Conventional Alternatives (10 MW to 100 MW Capacity Ranges)

The capital costs for various technologies which would compete with clean coal diesels are strongly dependent upon the configuration and the size of the power generation system. As the size of the equipment decreases, its cost per unit of capacity increases (see Figure VI-3). This economy of size provides part of the economic advantage of coal diesels, in that coal diesel costs that are less dependent on size than competing technologies. This is another key advantage of the modular design of clean coal diesels.

Although simple gas turbines are relatively inexpensive, the types of units that would be competing with clean coal diesels in the 10-100 MW power range (gas turbine cogeneration, STIGs, or gas turbine combined cycles - GTCC) are likely to have installed capital costs in the range of $900 to $1500 per kW (1991$). Provision of (SCR) to meet the stringent environmental requirements add an additional $50 to $120 per kW to the gas turbine cost, depending on size of the power generating equipment. The cost of competing technologies, when retrofitted with environmental controls, is typically over $1000 per kW in target application segments. In short, the clean coal diesel at $1300/kW can compete with natural gas power plants (GTCC) on an installed cost basis and is significantly cheaper than competing clean coal technologies.

6. Combination of High Efficiency and Moderate Costs Will Make Fuel Clean Coal Diesels Economically Attractive in a Range of Power and Cogen Applications

Multiple applications are being considered for clean coal diesels including electric power only and those utilizing both electric and thermal outputs (cogeneration). A key parameter indicating the relative economics of the technology options is their relative costs of electricity in a power-only mode, since electricity is, by far, the most highly valued system output.

Figure VI-4 indicates estimates of electricity costs calculated using the Arthur D. Little Power Generation Cost Model. This model uses standard life cycle costing procedures commonly used by electric utilities. Capital, operating and maintenance, and fuel and electricity costs are the principal cost variables one needs to consider in order to develop levelized electricity cost estimates for clean coal diesels and for conventional power generation technologies. For conventional technologies, the Arthur D. Little Power Generation Cost Database was used as a source of input data. DOE/EIA fuel price forecasts out to the year 2010 were used, and typical escalation factors were used for the period 2010 to 2030.

At natural gas price levels currently projected through about 2005, Figure VI-4 confirms that the clean coal diesel technology is not yet economically viable as compared to the other alternatives such as GTCC (natural gas) or advanced PC units. This is consistent with the current situation whereby clean coal diesels are being

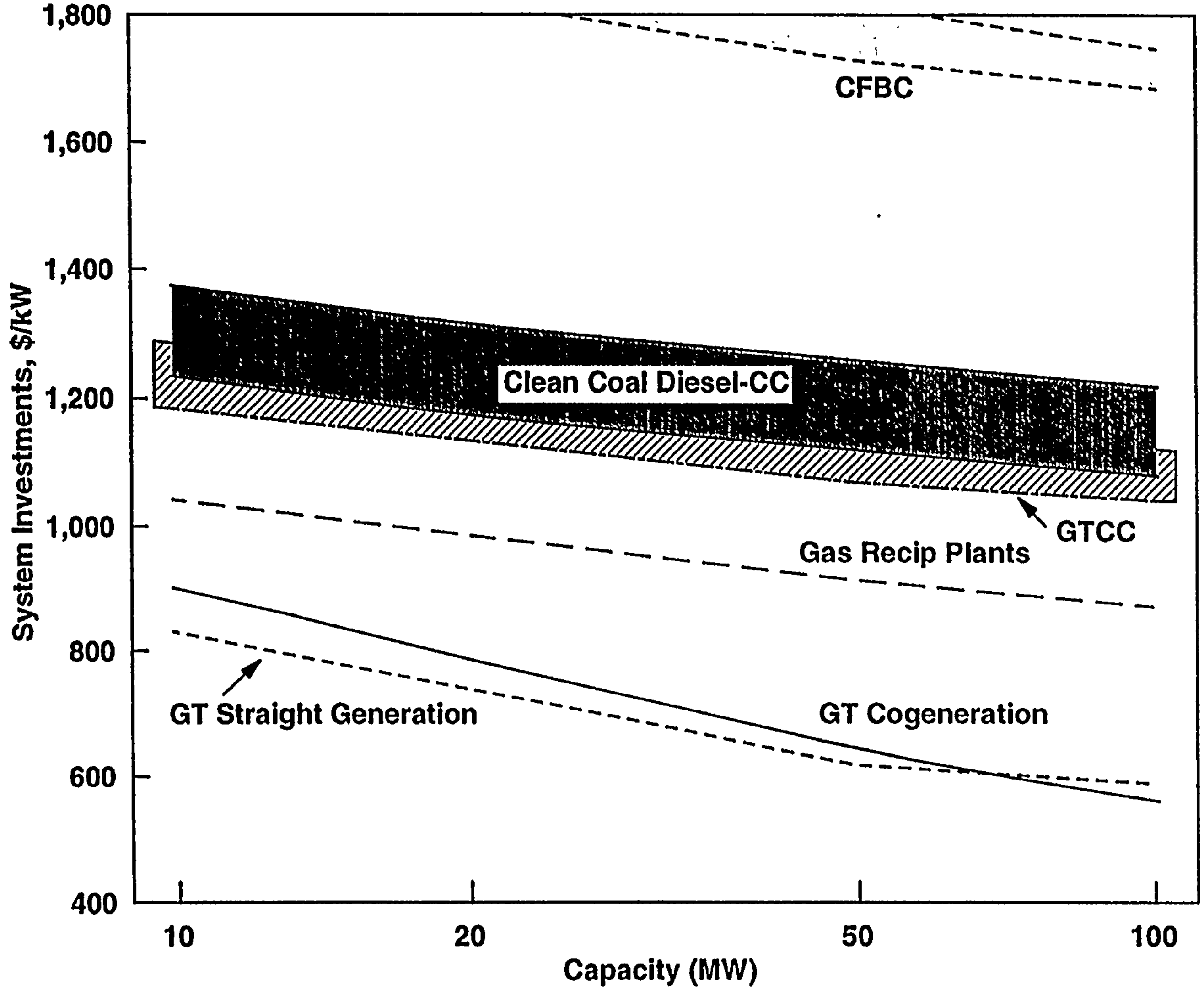


Figure VI-3. Clean Coal Diesel Capital Cost will be Competitive with Natural Gas GTCC Power Plants 10-100 MW Capacity Range assembled on a field test basis in preproduction quantities. The clean coal diesel is proposed as a CCT-V technology for the post-2000 period.

The analyses indicates that if and when natural gas prices rise to about the $4.50/MMBtu level, the economics of clean coal diesel technology become favorable as compared to the alternatives considered - particularly in the lower capacity range (below 50 MW), which is the primary focus for distributed power and cogeneration applications.

7. *Applications and Market Segments*

The markets for clean coal diesel technology can be divided into three application segments each with distinct technology and competitive characteristics (see Section VI-E for details). These are:

1. Non-Utility Distributed Power (10-100 MW), which includes

   a. Industrial Cogeneration
   b. "IPP," Independent Power Producers

2. Small utility plant expansion and repowering, Municipal Utilities and Co-Ops

3. Export CCT Technology for Distributed Power in Developing Countries (10-100 MW)

The segments include applications across a span of power capacities from 10 MW to 100 MW. Coal diesels have excellent performance potential across the range of applications and capacities which cover all three of these markets. This is due, in part, to the inherently modular construction of diesel engines, whereby large systems are typically constructed from multiples of a common engine model, and each engine model is constructed from multiples of a specific cylinder design.

Table VI-2 shows how the clean coal diesel meets the important market priorities in each potential market sector. Before analyzing these three market sectors in more detail, in the following three sections we project and substantiate the important characteristics of the CDCC technology in its mature form:

Section VI-B: Emissions Characteristics
Section VI-C: Energy Efficiency of CDCC
Section VI-D: Cost of Electricity of CDCC

Table VI-2. Major Potential Market Segments for Clean Coal Diesels

Application	Important Parameters	Clean Coal Diesel Attributes for this Application	Competitive Technologies (Examples)
Distributed Power (1 MW-100 MW)	• Reliability	• Modular capacity increments	• Fuel cells in urban applications (post year 2000)
	• Cost of power	• Low installed cost	• I.C. engines and gas turbines in rural applications (natural gas and fuel oil)
	• Small footprint	• Good part load efficiency	• Small AFBC units
	• Low installed cost	• Low emissions	
Cogeneration (1 MW-50 MW)	• Overall efficiency	• Ready access to waste heat	• Gas/diesel I.C. engines (gas and oil fired)
	• Quality of waste heat	• Low emissions	• Gas turbine plants (natural gas)
		• Modular construction	
		• High reliability/low maintenance	
Export CCT for Distributed Power in Developing Countries	• Installation ease	• Small footprint	• Gas turbine combined cycle (where natural gas is available)
	• Footprint (size)	• Modular capacity	• Diesel I.C. Engines (heavy fuel oil)
	• Lifecycle cost	• Feasible maintenance	
	• Ease of maintenance		

B. Environmental Performance Characteristics

1. Introduction

This section describes the environmental performance of the commercial embodiment of the Coal-Diesel Combined Cycle (CDCC). The environmental performance of the commercial CDCC plant will be superior to that of the Demonstration Facility at Cooper-Bessemer's Mt. Vernon Laboratory. This is because of the increased maturity of the CDCC technology and because of advances in current commercial emissions control technology. The emissions of key pollutants from the commercial CDCC plant when burning an engine grade coal water slurry prepared from a typical Eastern bituminous coal are summarized in Table VI-3.

The commercial CDCC plant has emissions of the three major criteria pollutants (NO_x, SO_x, and particulate) which are more than 90 percent lower than those for a typical PC plant. The commercial CDCC plant dry solid waste stream is some 45 percent lower than that for a typical PC plant.

Table VI-3. Emissions of Commercial Coal Diesel Plant and Typical PC Plant

Pollutant	Control Method Used	Emission Rate (Lb/MMBtu)	
		Coal Diesel Plant	Typical PC Plant
NO_x	Optimized CWS combustion, SCR, and duct injection	0.11	1.20
SO_x	Sodium bicarbonate duct injection plus baghouse	0.37	3.80
Particulate	Baghouse plus cyclone	0.01	0.10
Dry solid waste	Specialized landfill	7.28	13.20

In the commercial CDCC, NO_x is controlled by three approaches: in-cylinder combustion optimization; selective catalytic reduction; and reduction across the sodium bicarbonate sorbent injection system. Sulfur control is achieved by firing CWS prepared from coal cleaned to 0.5-1.5 percent sulfur and by duct injection of sodium bicarbonate. Particulate control is achieved by use of a cyclone separator between the exhaust manifold and the turbocharger and a fabric filter system downstream of the sorbent injection location. The fabric filter system is an integral part of the sulfur control system. The complete CDCC plant showing all components of the emissions control system is illustrated schematically in Figure VI-5. This illustration is for a 14 MW two-engine CDCC plant, but the configuration of the emissions control system components is similar to that for any commercial CDCC plant.

Air toxics are of increasing concern in coal-fired power generation systems. Based on preliminary test results, the air toxics control performance of the commercial CDCC is expected to be superior to most coal-fired technologies because approximately 90% of the heavy metals appear to be removed in the gravimetric coal cleaning process. In addition, there is mounting evidence the bulk of the remaining toxics are found on the surface of the very fine (sub-micron) particulate. Of the commercial particulate control technologies available, the fabric filter has the best collection performance for very small particulates, and consequently is expected to have the best air toxics collection performance. There is also a possibility that the short combustion time in the coal diesel cylinder will limit the amount of toxic vaporization that can occur, thereby ensuring superior control of toxic compounds in downstream particulate control equipment.

The sulfur control technology employed in the commercial CDCC is a dry system, and consequently generates no waste water stream. The cooling water circuit employs a dry cooling tower. Consequently, the only liquid waste stream generated is the small boiler blowdown water stream. Discussions of the emissions control

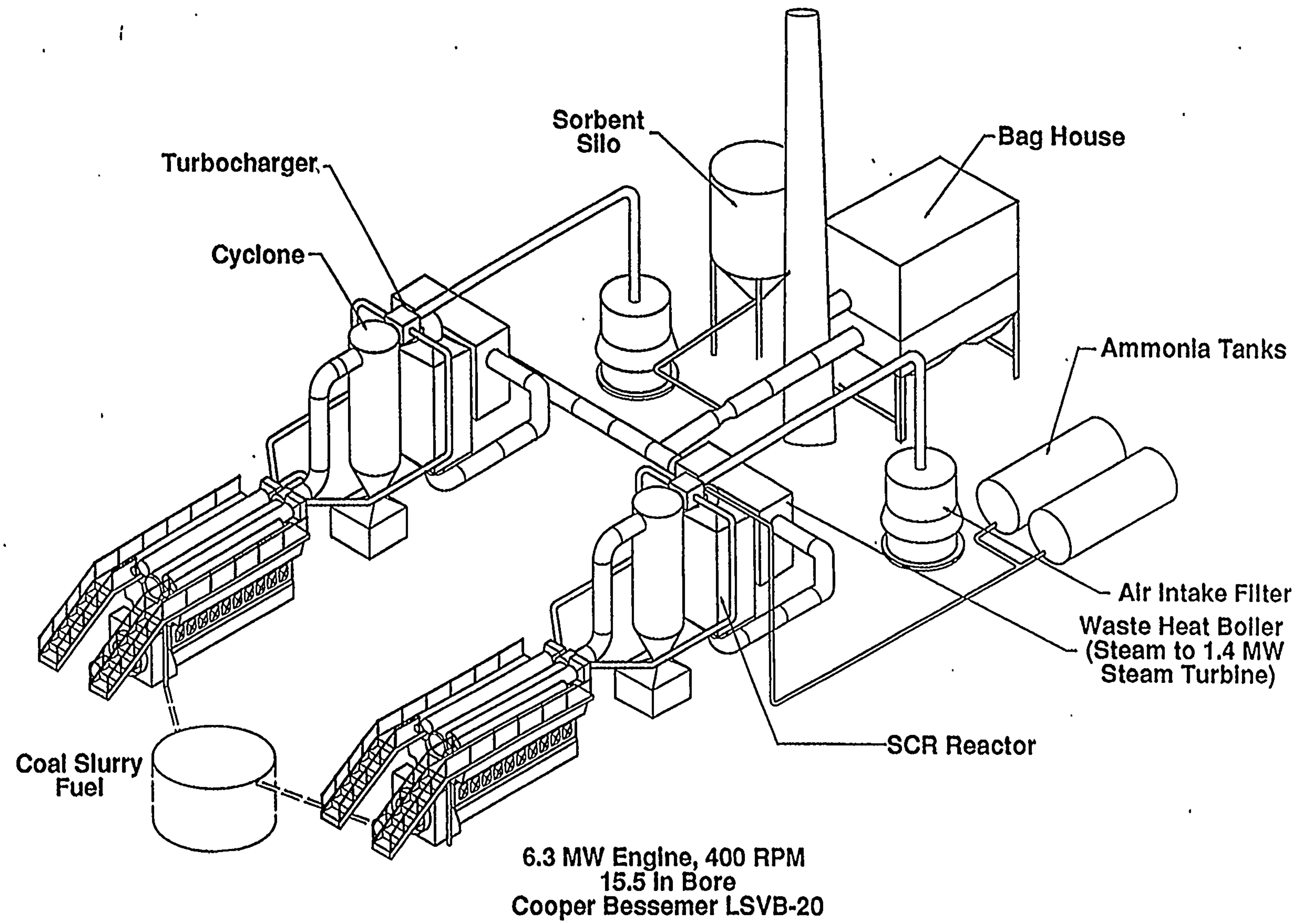


Figure VI-5. 14 MW Coal-Fueled Diesel Power Plant with Steam Turbine Bottoming Cycle technologies employed in the commercial CDCC are contained in the following sections.

2. NO_x Emissions

The CDCC fires engine grade coal-water slurry, which contains approximately 50 percent water. The presence of this quantity of water in the fuel suppresses the in-cylinder peak temperature and consequently the thermal NO_x formation rate. This, together with optimization of the CWS injection process, has yielded coal diesel NO_x exhaust manifold levels of 400-600 ppm at 11% O_2. Uncontrolled NO_x emissions from non-coal diesel engines are approximately twice this level, at more than 1200 ppm.

The in-cylinder NO_x control will be augmented by post-combustion NO_x control in order to meet current and future stringent emissions requirement. Selective catalytic reduction (SCR) will be used as the post-combustion control method as it is the only commercially available method which can meet the performance targets.

SCR units have been widely employed on coal-fired power plants. To date most of the experience with SCR in coal applications has been in Germany and Japan. SCR units have also been used widely on oil fueled reciprocating engines in Germany and to some extent in the U.S. SCR units have been demonstrated to achieve 90 percent NO_x reduction with very low ammonia slip rates. A number of manufacturers are now prepared to guarantee performance targets of greater than 90 percent reduction with less than 10 ppm ammonia slip. The only SCR unit tested on a coal-fired diesel is the zeolite SCR tested as part of the DOE/METC program. Performance to date has been as guaranteed by the manufacturer. The commercial CDCC plant is therefore assumed to be equipped with an SCR system which gives 90 percent NO_x reduction with less than 10 ppm ammonia slip. The zeolite catalyst employed has many advantages over conventional vanadia-titania catalysts, including an operating temperature window compatible with diesel post-turbocharger temperatures, resistance to poisoning by ash constituents and sulfur compounds, and non-hazardous disposal of the spent catalyst. The SCR catalyst must be replaced every three to six years, depending on the zeolite durability in the coal flue gases.

In addition to the SCR NO_x reduction, the sodium bicarbonate injected for sulfur control can cause a small amount of NO_x reduction. This typically amounts to 20 - 30 percent of the NO_x level at inlet to the sorbent injection/baghouse. Thus, an additional percentage of the engine-out NO_x will be reduced in the sulfur control system. By using these three NO_x control approaches, the commercial CDCC plant is estimated to emit NO_x at the rate of 0.11 lb/MMBtu when firing an engine grade CWS prepared from a typical Eastern bituminous coal (comparable levels were measured in the 100-hour test).

3. SO_2 Emissions

Engine-grade CWS has a sulfur content of approximately 0.5 to 1.5 percent. This sulfur content yields SO_2 in the untreated engine exhaust of about 150 to 450 ppm at 11% O_2. Dry duct injection sodium bicarbonate will be used in the commercial CDCC plant.

Duct injection of sodium bicarbonate is a commercial technology which has been shown to produce up to 90 percent SO_2 removal from coal flue gases. Use of a dry sodium bicarbonate injection system which obtains 85 percent SO_2 removal, together with the sulfur removal obtained in coal cleaning and slurry preparation, is therefore projected to yield SO_2 emissions for the commercial CDCC plant of 0.37 lb/MMBtu when firing an engine grade CWS prepared from a typical Eastern bituminous coal. This would be for a coal having a sulfur content of 2.5 percent sulfur, which is reduced to 1.7 percent by coal cleaning. The total sulfur reduction obtained in the commercial CDCC using a typical coal would be 90 percent.

Dry sorbent injection technologies have a clear cost advantage over conventional scrubber technologies. Levelized costs for dry injection systems have been estimated to be a factor of three lower than those for scrubbers for CDCC plants of the size of the commercial plant. In addition, dry sodium injection systems have several advantages relative to calcium duct injection systems for application to the coal-fueled diesel. First, the capital cost of the sodium-based system is lower. Second, injection of sodium sorbents is a more mature technology. Commercial installations are now in the field, where as the first large-scale demonstrations of calcium duct injection are only now currently taking place. These demonstrations have indicated that the reliability of calcium injection/humidification systems is not high, especially when injecting into small ducts while operating near the dew point of the flue gas, as is required to approach 75 percent SO_2 reduction. Finally, injection of sodium sorbents has been shown to remove 10 to 40 percent of NO_x in addition to SO_2. Thus, the low capital and operating costs and high maturity of sodium-based duct injection will ensure that high levels of SO_2 reduction will be achieved within the timeframe of engine commercialization.

4. Particulate Emissions

Larger particles in the coal diesel engine exhaust can erode the turbocharger, causing maintenance and performance problems, and must therefore be removed upstream of the turbocharger. This is accomplished using a conventional reverse-flow cyclone separator. The cyclone separator is designed to have a 50 percent cut diameter of 5 microns. Additional particulate control must be implemented to achieve acceptable emissions standards. In the commercial CDCC plant a fabric filter system with reverse pulse-jet cleaning would be used. This system is conventional and commercially proven and can achieve up to 99.9 percent capture. It is also an integral part of the SO_2 control system. Based on the coal-diesel tests, the projected particulate emissions from the commercial CDCC plant are 0.01 lb/MMBtu.

5. *Air Toxics*

Limited data exist on the emission of air toxics from coal diesels. As part of this program, data was obtained on the mercury, selenium, and arsenic content of source coal, cleaned coal, coal-diesel fly ash. The results indicate that approximately 90% of these heavy metals were removed by the gravimetric coal cleaning process. However, any emissions estimates are based on the limited data and understanding of the fate of trace elements in other coal combustion processes. Trace elements in coal may be classified into three groups depending on their volatility and behavior during coal combustion. The compositions of these groups is illustrated in Figure VI-6. Group 1 elements are concentrated in larger particulates or bottom ashes and slags. Group 2 elements are volatilized in combustion but condense downstream and are concentrated in fine particulate. Group 3 elements are the most volatile and are depleted in all solid phases. There is considerable overlap between the groups, depending on the element and the operating condition.

Combustion temperature and residence time are important parameters governing trace element behavior. The combustion timescale is short in the coal diesel, which should lead to lower volatilization of trace elements and reduced toxic emissions. Based on the limited amount of data available from coal-fired power plants, it appears that 70-80 percent of the mercury remaining in the clean coal fuel remains in the gas phase downstream of the particulate control systems, with small amounts of B and Se (20 - 30 percent) and very small quantities of other trace elements.

The total emission of all trace elements is estimated at less than 0.002 lb/MMBtu. Thus, based on the above assumptions and at a plant annual capacity factor of 90 percent, the commercial 45 MW CDCC plant would emit less than 2.5 tons per year of all toxics and would be exempt from regulations under Title III of the 1990 Clean Air Act Amendments. Title III requires reductions at plants which emit more than 10 tons per year of any one toxic compound or more than 25 tons per year of any combination of toxic compounds.

6. *Solid Wastes*

There are two sources of dry solid wastes in commercial CDCC plants: the cyclone hopper and the baghouse hopper. Solids collected in the cyclone consist primarily of coal ash and unburned carbon. The baghouse waste consists of reacted sorbent (sodium sulfate), unreacted sorbent (sodium bicarbonate), ash, and carbon. For a 45 MW CDCC plant, the total dry solids waste production rate is projected to be approximately 1 ton per hour or approximately 7 lb/MMBtu. The composition of the total solids waste stream by mass is given in Table VI-4.

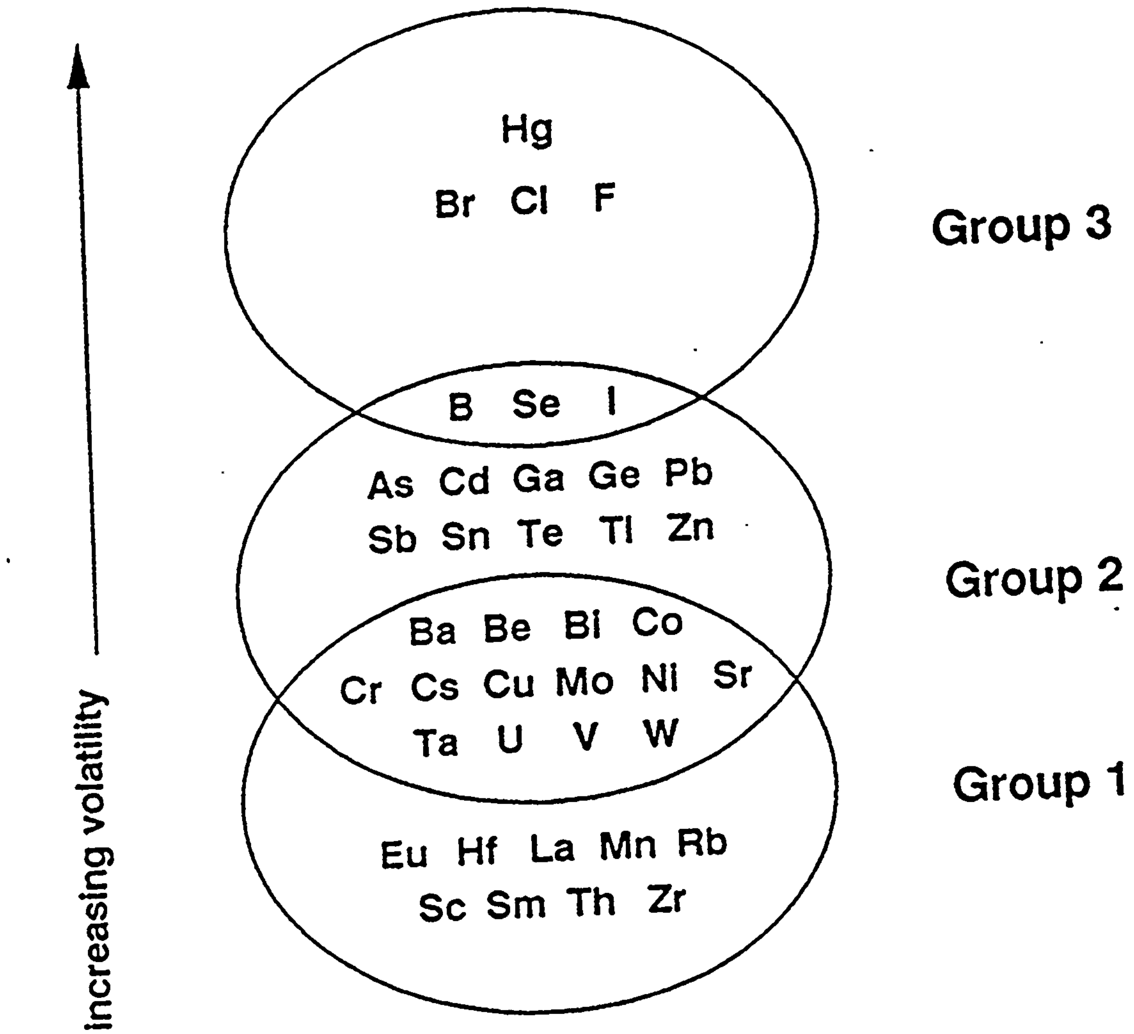


Figure VI-6. Classification of Trace Elements from Coal Combustion

Thus, approximately 30 percent of the waste is ash and carbon (aqueous-insolubles) and approximately 70 percent is soluble sodium compounds. Because of the high levels of soluble constituents, disposal of such untreated waste can be expected to be subjected to disposal restrictions involving impoundments or landfills designed for minimization and containment of leachate. This will usually consist of a double-barrier lining system with runoff provisions and leachate monitoring/treatment similar to that specified for RCRA-designated hazardous waste (as well as for flue gas desulfurization (FGD) sludges in many states). In addition to soluble salts, other concerns are chemical oxygen demand from soluble sulfite species and the potential for solubilization of heavy metals.

Table VI-4: Estimated Composition of Waste Stream from Commercial CDCC Plant (45 MW)

Constituent	Production Rate lb/MMBtu
Na_2SO_4	4.43
$NaHCO_3$	0.65
Ash	1.47
Carbon	0.73

The cost of disposal in double-lined impoundments is not expected to be significantly different from that for sludge produced from other sodium-based and many calcium-based FGD technologies. There also exist potential methods for stabilization and recycling of the spent sorbent.

7. Range of Source Coals

The excellent environmental performance of the commercial CDCC plant relies both on preparation of engine-grade CWS from coals that can be cleaned to approximately 2 percent ash, as well as on the performance of the coal-diesel engine and its post-combustion emissions control devices. It is therefore essential to the commercial success of the CDCC technology that sufficient reserves of suitable coals exist.

Over 10 billion tons of suitable coal have been identified in the U.S. This was done by surveying the washabilities of US coals and determining which seams are cleanable to 2 percent ash with a heavy media cyclone. Some of these coal seams, such as Upper Elkhorn #3, Brookville, Blue Gem, Imboden, and Dorchester can be cleaned with high enough cleaning efficiencies that plants could be constructed to produce a premium coal product exclusively. However, the majority of the coals surveyed could easily be separated into a premium product with less than 2 percent ash and a middlings product that is appropriate for conventional boiler use (10 - 12 percent ash). Significant coal seams appropriate for this coal cleaning strategy include: Indiana #7, Brazil Block, hazard #5, Upper Elkhorn #1 and 2, Lower and Middle Kittaning, Lower Freeport, Pittsburgh, Splash Dam, Upper and Lower War

Eagle, Pocahontas #3 and Winifrede. All these coal seams have reserves of 100 to 500 million tons each.

C. Energy Efficiency of Commercial CDCC Plant

The smaller coal-diesel combined cycle (CDCC) plants, (up to about 40 MW) will be built around Cooper-Bessemer LSVB-20 engines. These engines have a simple cycle efficiency of approximately 40 percent and a combined cycle efficiency of approximately 45 percent. The larger commercial CDCC plants (40-150 MW) will be based around modern high performance diesel engines which will have a simple cycle efficiency of 45 percent and combined cycle efficiency of approximately 49 percent.

A typical 45 MW commercial CDCC plant is illustrated schematically in Figure VI-7. The commercial CDCC plant is expected to range from 14 MW (two 6 MW engines plus steam turbine) to 160 MW (six 25 MW engines plus steam turbine), with a typical size being 45 MW (three 14 MW engines plus steam turbine), as illustrated. Key performance parameters for this hypothetical 45 MW commercial CDCC plant are given below:

Coal heat input:	318.5 MMBtu/hr
Diesel engine electrical output:	42.0 MW
Diesel engine heat rate (LHV):	7582 Btu/kWh
Steam turbine electrical output:	3.0 MW
Total plant electrical output:	45.0 MW
Plant heat rate (LHV):	7078 Btu/kWh
Overall generating efficiency (LHV):	48.2%

This plant is based on modern diesel engines which operate at 45 percent simple-cycle efficiency. Such engines are currently in commercial production for liquid and gaseous fuel operation and can be converted for coal-water slurry operation. Table VI-5 lists examples of high efficiency diesel manufacturers and engine models.

One example of a high efficiency diesel engine is the MAN-B&W 9K80MC-S two-stroke engine. This engine operates at 100 rpm, generates 24.5 MW, and has a simple cycle efficiency of 50.2 percent (LHV) on fuel oil. A 50 MW combined cycle power plant at Coloane in Macau (see Figure VI-8), based on two such engines, has an overall generation efficiency of 52.5 percent (LHV). Future diesel engines are expected to achieve simple cycle efficiencies of up to 53 percent and combined cycle efficiencies of up to 55 percent. The fact that actual existing diesel plants operate at

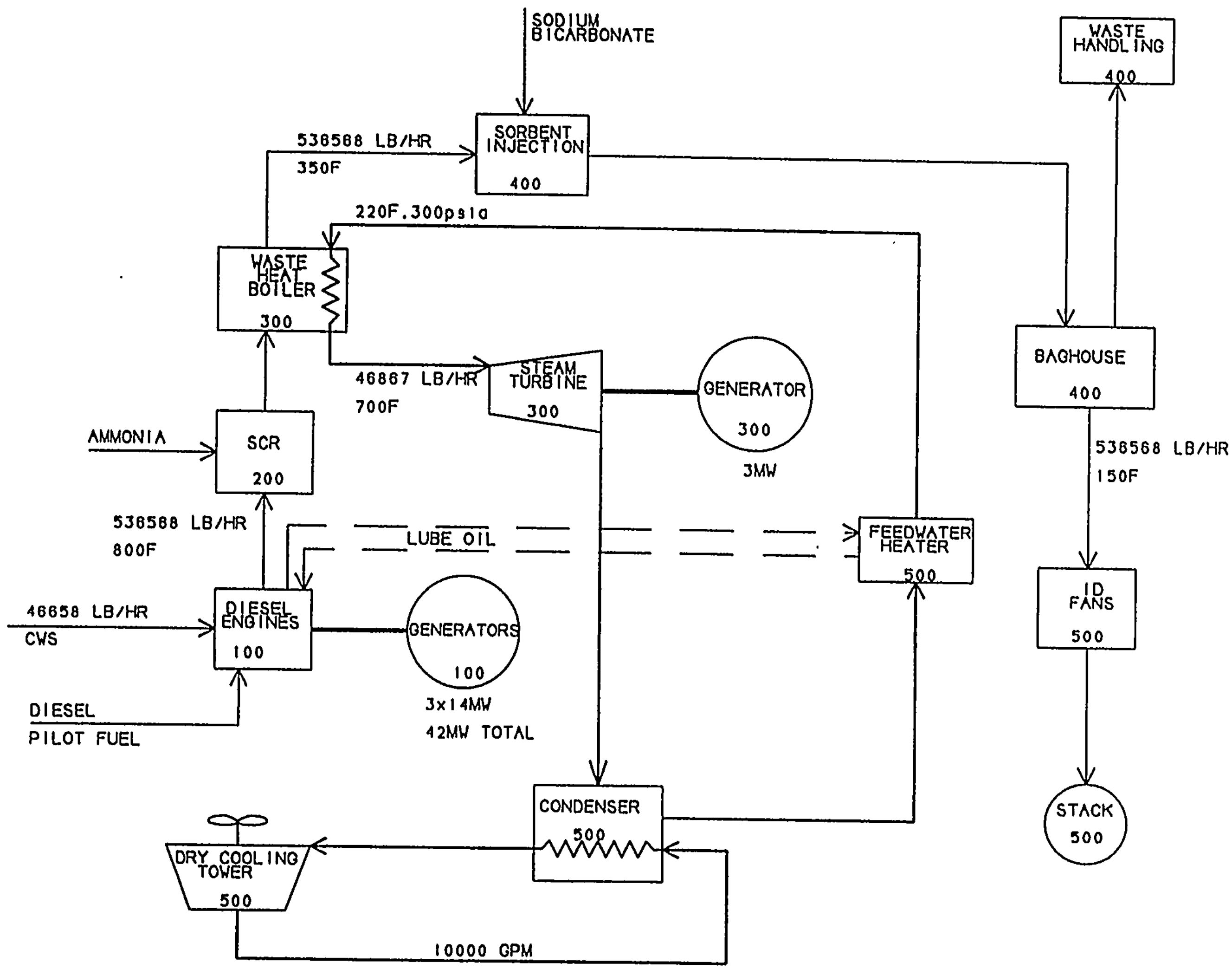
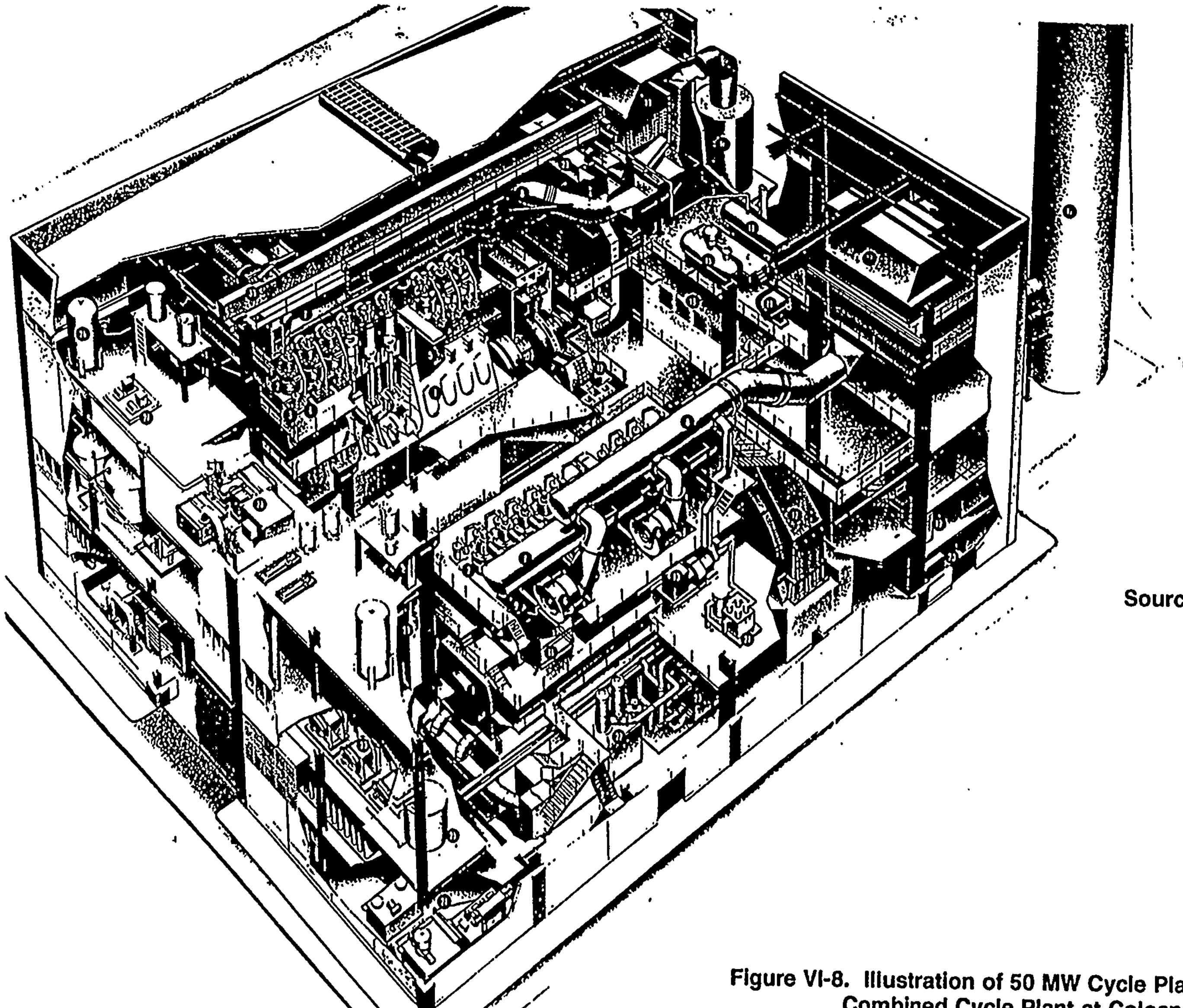


Figure VI-7. Flow Diagram for Commercial, 45 MW Coal-Fueled Diesel Combined Cycle Plant



Source: Modern Power Systems

Figure VI-8. Illustration of 50 MW Cycle Plant Oil-Fired Diesel Combined Cycle Plant at Coloane Macau

52.5% (LHV) supports the expectation that the coal diesel will readily reach these performance levels when offered to the NUG and small utility market.

Table VI-5: Proven High Efficiency for Selected Diesel Engine Models (Over 10 MW Output)

Manufacturer	Model	kW/cyl	Rating (MW)	Simple Cycle Efficiency (% LHV)
MAN B&W	K 80 MC-C	3410	40.9	49.2
Pielstick (Coltec - Fairbanks Morse Div.)	PC 4.2	1210	21.8	43.4
Mitsubishi	UEC 85 LSII	5250	42.0	53.0
New Sulzer	RTA 94C	3820	30.6	50.8

In the commercial CDCC plant illustrated in Figure VI-7, each of three coal-fired diesel engines drives a 14 MW generator. The engines fire 50 percent solids coal-water slurry prepared from Eastern bituminous coal cleaned to 2 percent ash at the mine site. The performance parameters for all three engines are summarized below:

CWS lower heating value:	6825 Btu/lb
CWS solids content:	50%
CWS flow rate:	46,658 lb/hr
Fuel heat input:	318.5 MMBtu/hr
Electrical output:	42.0 MW
Exhaust gas mass flow rate:	536,568 lb/hr
Exhaust gas pressure:	10" H_2O
Exhaust gas temperature:	800°F (prior to exhaust gas treatment)

Exhaust gas from the diesel produces additional power using a waste heat boiler, as follows: The waste heat boiler generates steam at 700°F and 300 psia which is expanded through a steam turbine. The steam turbine drives a generator which produces 3.0 MW. The boiler feedwater is heated using the heat rejected to the diesel engine lubrication oil, which amounts to some 3-4 percent of the diesel engine heat input. The steam bottoming cycle is based on conventional waste heat recovery and steam system components. The bottoming cycle performance parameters are summarized below:

Steam side boiler pressure:	300 psia
Stem flow rate:	46,867 lb/hr
Generator electrical output:	3.0 MW
Gas exit temperature:	350°F

Feedwater temperature:	220°F
Superheated steam temperature:	700°F
Turbine exhaust pressure:	3" HgA
Condenser temperature:	116°F

The commercial CDCC plant shown in Figure VI-7 has a simple and conservative bottoming cycle. It is possible to improve the overall generation efficiency by using more sophisticated bottoming cycles. For example, approximately 10 percent of the heat input to the coal diesel engine is rejected to the engine cooling jacket. A low pressure loop in the Rankine bottoming cycle is needed to recover this low temperature energy. A more efficient method is to use ebullient cooling, in which saturated steam generated by the boiling of water in the engine water jacket is admitted directly to a low pressure stage of the steam turbine. This approach, though adding complexity to the system, has the potential for providing approximately twice the steam turbine power output of the simple bottoming cycle in the commercial CDCC. This approach, together with optimization of boiler pressures, approach temperature differences, and feedwater temperatures, and the use of future advanced diesel engines, could lead to overall cycle efficiencies as high as 53 - 55 percent.

The energy efficiency of the commercial 45 MW CDCC plant is compared to that of a typical 306 MW pulverized coal (PC) plant in Table VI-6.

The difference in scale between the CDCC commercial plant and the typical PC plant should be noted. A PC plant at the same scale as the commercial CDCC plant (45 MW) would have a lower generating efficiency than that of the typical PC plant. Other competing small scale coal-based power generation technologies include stokers (10 - 50 MW) and atmospheric fluidized beds (30 - 150 MW). Both of these would also have lower generating efficiencies than the typical PC plant. Thus the comparison with the typical PC plant is a conservative one.

Table VI-6. Comparison of Efficiency Performance of Commercial Coal Diesel Plant and Typical PC Steam Plant

Parameter	Units	Commercial Coal Diesel Plant	Typical Pulverized Coal Steam Plant
Coal burn rate	tph (dry)	11.7	117.6
Fuel type	-	CWS, 50% solids	PC
Fuel LHV	Btu/lb	6,825	11,742
Fuel heat input	MMBtu/hr	319	2,902
Diesel engine electrical output	MW	42.0	-
Steam turbine electrical output	MW	3.0	305.7
Plant output	MW	45.0	305.7
Plant heat rate (LHV)	Btu/kWh	7,078	9,018
Generating efficiency (LHV)	%	48.2	37.8

The major improvement in power generation efficiency between the CDCC plant and the reference plant stems from the use of a heat engine of fundamentally high thermodynamic efficiency as the major generating device. The addition of a steam bottoming cycle to give a combined cycle increases the CDCC efficiency advantage.

D. Cost of Electricity for Coal Diesel Technology

1. Summary of Cost Performance

The mature clean coal diesel technology is estimated to achieve a levelized cost of electricity as low as 6.0 ¢/kWh for installations in the 2000 - 2010 time period. This estimate is based on capital and operating cost estimates for a 45 MW commercial Coal Diesel Combined Cycle plant operating at an 80% percent capacity factor. The economics of the coal-fueled diesel plant has been continually re-evaluated as new information on coal fuels, emission control technologies, engine component performance and other factors has been developed. The analysis methodology includes the use of sensitivity analysis to explore trade-offs between coal fuel price, emission control costs, maintenance costs, etc. The framework for the economic analysis was originally formulated in 1986 (Arthur D. Little, 1986), and the analysis was repeated with revised assumptions in 1988 (Rao, et al., 1989), and again in 1989 (Benedek, et al., 1990). The results of these analyses can be found in published DOE reports and ASME papers. The present economic analysis is framed around the 45 MW Combined Cycle Coal Diesel plant described above (see Figure VI-7).

The cost estimates for the commercial Coal Diesel Combined Cycle (CDCC) plant are summarized and compared to those for conventional IC engines in Figure VI-9. our conclusions, as explained in detail below, are that:

- The projected levelized cost of coal diesel power is 6.0 ¢/kWh for installations in the 2000 - 2010 period.

- Coal diesel technology will be competitive with gas and oil technologies when gas and oil prices reach approximately $4.50/MMBtu (various projections by DOE and others indicate that this will occur in the 2000 - 2010 timeframe).

- Economic viability hinges on the cost of CWS production (the engine-grade CWS price must be kept below approximately $3/MM Btu).

2. Engine Component Cost Premiums Associated with Coal

The use of beneficiated coal-water slurry necessitates certain modifications to the standard large diesel engine, both in terms of special components and special maintenance practices. Modified coal-tolerant engine components include larger fuel pumps for CWS; larger hardened injectors; modified fuel cams; larger camshafts; a

Scenario: The cost of power for a coal-fueled diesel combined cycle plant will be competitive with that of an oil or gas-fueled plant.

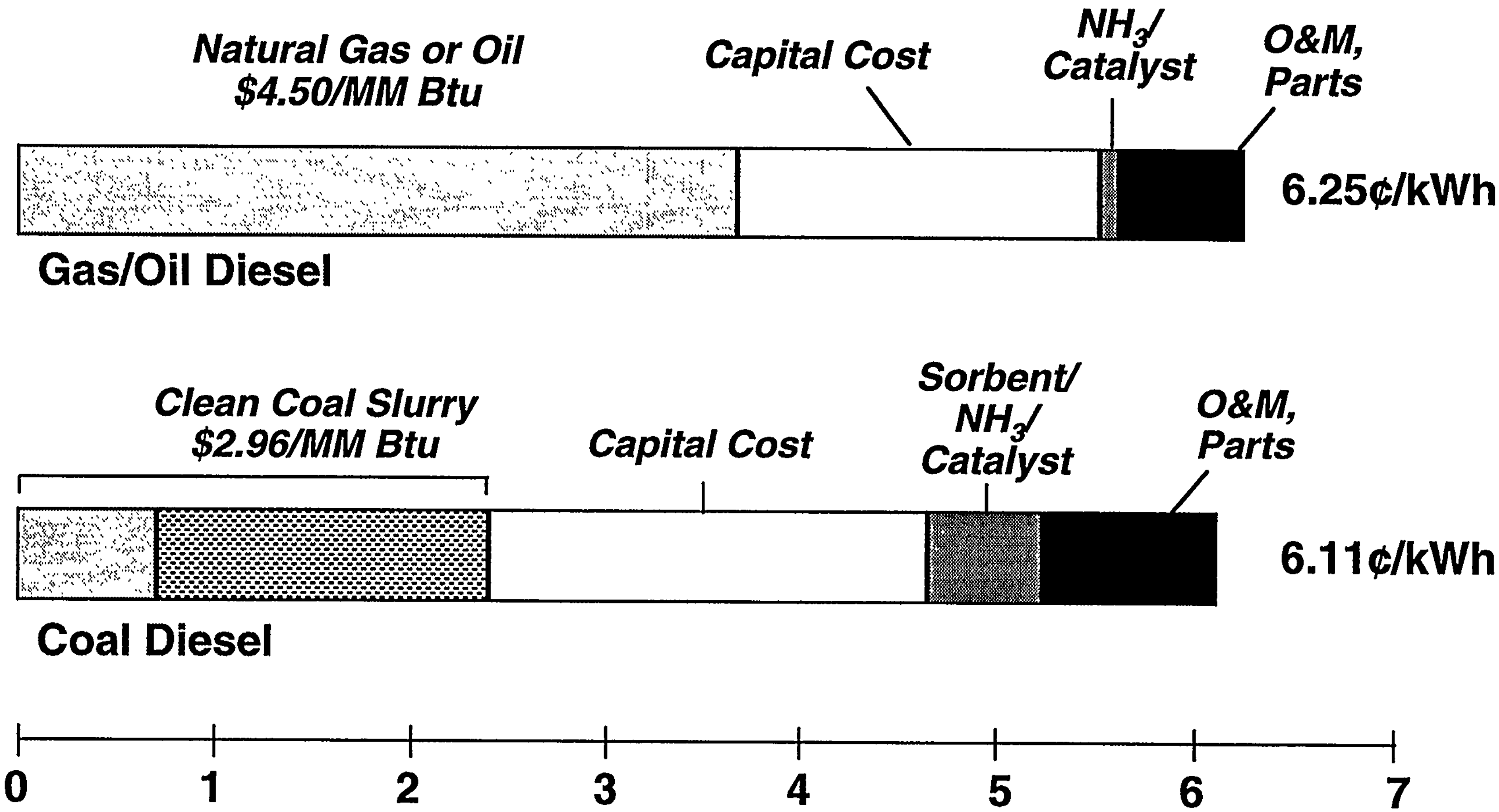


Figure VI-9. Cost Estimates for the Commercial Coal Diesel Combined Cycle (CDCC) Plant

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modified engine block; AMBAC-designed CWS injection system; ceramic coated rings and liners, hardened valve seats and valve stems; and a ceramic coated turbocharger.

For the initial field demonstrations using 6.3 MW Cooper-Bessemer engine/generator sets, the price premium for these coal-tolerant components will be approximately $1.1 million per engine, or approximately $175/kW. Later, for commercial CDCC plants, which will be built around high-efficiency coal diesel engines, this premium is expected to be reduced to $120/kW. The "learning curve" reduction from $175/kW to $120/kW is based on growth in manufacturing volume by Cooper-Bessemer and its licensees. Thus for the 45 MW commercial CDCC plant, the cost premium associated with coal-tolerant engine components will be an additional 0.24 ¢/kWh.

Present overhaul and parts replacement practices for Cooper-Bessemer oil or gas diesel engines are compared to those for the coal diesel in Table VI-7. The cost associated with the increased cost of parts replacement is projected to be $22.8 million over a 20 year period, versus $9.8 million for the standard diesels. This corresponds to 0.36 ¢/kWh for the coal engine versus 0.16 ¢/kWh for the standard diesel.

Table VI-7: Impact of Coal Fuel on Parts Replacement and Overhaul Practices

Item	Standard Diesel	Coal Diesel Projections	20-Year Overhaul Cost[1]
Injectors	2,000 hr	1,000 hr	$5.93M
Minor maintenance checks	8,000 hr	4,000 hr	$2.28M
Top-end overhaul	25,000 hr	12,000 hr	$4.56M
Major overhaul	100,000 hr	25,000 hr	$10.03M
Total	-		$22.8M

Thus, the engine purchase and parts replacement cost premium associated with the use of coal for the 45 MW commercial CDCC plant is approximately 0.44 ¢/kWh. Since the standard busbar cost of producing power with diesel fuel or natural gas can range from 5 to 7 ¢/kWh, depending on the prevailing fuel prices, this premium for coal-tolerant engine components represents less than a 10 percent increase in the cost of power.

3. *Cost to Produce Engine-Grade Coal Fuels*

An essential ingredient in the future of coal-fueled diesels is the eventual emergence of a price advantage of the engine-grade coal fuel. Recognizing that (a) fuel oil and natural gas prices will almost certainly rise during the 1995-2010 timeframe and (b) that the extent and timing of the oil price rise is virtually unpredictable, we have concentrated on the CWS fuel cost and how it might be reduced as much as possible. The CWS fuel cost delivered to the CDCC plan, based on our assumptions, is

$2.97/MMBtu (see Table VI-8). Chapter II provided details supporting the CWS production operation and maintenance cost breakdown and showed the capital costs associated with the cleaning and slurrying plants.

Our CWS costing is based on detailed inputs from CQ Inc., AMAX, and Otisca. Physical cleaning (resulting in a 0.5 - 2.0% ash premium coal product) is sufficient for the CWS to be compatible with the coal diesel. This eliminates expensive chemical cleaning steps. There are two possible approaches for premium coal production from run-of-mine coal:

1. Premium coal is the sole product. Coal sources are selected for high premium coal yields and cleaned using advanced coal cleaning technologies such as agglomeration or froth flotation.

2. Premium coal is the by-product. Coal sources that have some premium coal content are selected and the premium coal is extracted as a by-product of the existing coal cleaning operations using conventional coal cleaning technology such as heavy media cyclones.

Both of these approaches are feasible, however the by-product approach has less technical risk and allows for a much broader selection of coal sources. This is our preferred approach and is the basis of the CWS cost estimates used in this analysis. The overall process is thus as follows:

Table VI-8. Projected CWS Price for Regional Production

Cost Component	Cost ($/MMBtu)	% of Total CWS Cost
Feed coal[1]	0.888	29.9
Coal cleaning		
O&M	0.188	6.3
Capital recovery[2]	0.112	3.8
Subtotal	0.301	10.1
Coal transportation	0.264	8.9
Coal slurrying		
O&M	0.513	17.3
Capital recovery[2]	0.542	18.3
Subtotal	1.055	35.6
CWS transportation	0.461	15.5
Total	2.979	100.0

[1]Pre-cleaned feed coal is $24/ton. It is re-cleaned producing middlings that are sold for $22.98/ton and premium coal with a feed cost equivalent to $25.25/ton.
[2]Capital recovery includes 12.5% cost of money.

- Coal is physically cleaned at the mine in a 250-tph addition to an existing cleaning plant. Premium coal is extracted for transport to the slurrying plant and middlings are extracted for sale as utility steam coal.

- Premium coal is transported to a 100-tph slurrying plant located in the supply region where several coal diesel plants are located.

- Once produced at the slurry plant, the engine-grade CWS is transported by tanker truck (or rail if economical) to the coal diesel power plant a distance of up to about 25 miles.

4. Emissions Control System Cost

The major capital costs associated with the emission control equipment are:

- The SCR reactor; catalyst; ammonia storage, feeding and injection system; and control system.

- The sorbent storage, feeding and injection system; the baghouse vessel and filter bags; pulse-jet cleaning system; injection/baghouse controls; and solids waste handling.

Operating costs are principally:

- SCR system: Ammonia plus catalyst replacement every three to six years, depending on catalyst durability in coal flue gases.

- Sorbent/baghouse system: Sodium bicarbonate sorbent, plus solid waste handling and disposal costs.

Both of these emissions control options are commercial technologies with relatively well defined costs. For the 45 MW CDCC plant at a capacity factor of 80 percent, the cost premium for the SCR is estimated at 0.13 ¢/kWh (capital) and 0.16 ¢/kWh (operating), while that for the sorbent injection system is estimated at 0.16 ¢/kWh (capital) and 0.53 ¢/kWh (operating). Thus the SCR NO_x-control cost premium is 0.29 ¢/kWh and the SO_2/particulate control cost premium is 0.69 ¢/kWh, giving a total emissions control power cost premium of 0.98 ¢/kWh. This represents a power cost increment of 15-20 percent over an oil or gas-fired diesel plant, assuming that the oil or gas plant was not equipped with an SCR. If the oil or gas plant is equipped with an SCR (as will be increasingly common in order to meet current and future NOx emissions regulations), then the emissions control power cost premium for the coal diesel is only 0.69 ¢/kWh, a premium of approximately 10 - 15 percent.

5. Projected Cost of Power from 45 MW Commercial CDCC Plant

The information developed in the above discussions of costs associated with individual components of a commercial CDCC plant may be used to project the total cost of power and determine the relative importance of each economic parameter. Key findings from this analysis are presented in Table VI-9.

The plant heat rate is assumed to be the same for CWS firing as for oil/gas firing. It is also assumed that the oil/gas plant is equipped with an SCR. This is a reasonable assumption, given the trends in NO_x emissions regulations and permitting for large reciprocating engines. Non-fuel variable operating costs for the CDCC include ammonia for the SCR, replacement catalyst for the SCR, sorbent for sulfur control, and the cost of disposal of the reacted sorbent and engine particulate. For the oil/gas plant, only the ammonia and catalyst costs are considered. The capital and operating costs associated with the emissions control system are a significant component of the CDCC power cost. Based on the above analysis, the CDCC plant with a CWS cost of $2.95/MMBtu is competitive with and oil- or gas-fired diesel generating plant at an oil or gas price of approximately $4.50/MMBtu.

Table VI-9. Comparison of Projected Cost of Power for 45 MWS CDCC Plant and Oil/Gas Plant

Cost Element	Coal Plant ¢kWh	Oil/Gas Plant ¢/kWh	Notes
Capital (Installed)			
Engines + gen sets	1.45	1.22	
Emissions control	0.29	0.13	Oil/gas plant has SCR
Bottoming cycle	0.25	0.25	
Balance of plant	0.25	0.25	
Total	2.25	1.85	
Variable Operating			<u>Fuel</u>
Fuel	2.40	3.67	CWS $2.95/MBtu, Oil/Gas: $4.50/MMBtu
Non-fuel variable	0.58	0.10	<u>Non-Fuel</u> Coal: sorbent + waste NH_3 +
Total	2.98	3.77	catalyst Oil/Gas: NH_3 + catalyst
Fixed O & M	0.88	0.63	
Total Cost of Power	**6.11**	**6.25**	

Assumptions:
- 45 MW combined cycle plant
- Plant heat rate: 7078 Btu/kWh
- Plant capacity factor: 80%
- Future oil/gas price: $4.50/MMBtu

6. Comparison with Other Coal Technologies

The Arthur D. Little Power Generation Cost Model and Databases have been used to compare the capital cost, efficiency and cost of power among several competing coal-based power generation technologies. The key coal diesel plant characteristics listed in Table VI-10 were used in this analysis.

For the other coal-based technologies, the Arthur D. Little Power Generation Cost Database was used as a source of input data. Based on these sources of data, the trends in technology capital cost and efficiency for the period up to the year 2000 can be compared (see Figure VI-10). The coal diesel plant compares very favorable on efficiency with natural gas fuel cells and has lower capital costs than the coal-based integrated gasification fuel cell.

Table VI-10. Trends in Coal Diesel Combined Cycle Plant Characteristics

Status	Year	Installed Capital Cost ($/kW)	Efficiency (% LHV)	NO_x (Lb/MMBtu)	Remarks
Initial installations	2000	1600	45	0.15	Cooper-Bessemer simple cycle
Commercially viable plants	2010	1300	48	0.10	Combined cycle
Plants built after significant market share is reached	2030	1200	53	<0.10	Engine upgrade and in-cylinder NO_x control

The Arthur D. Little Power Generation Cost Model has also been used to calculate the cost of power for various competing coal-based power generation technologies, including the coal diesel. The model used standard lifecycle costing procedures commonly used by electric utilities, as outlined below.

$$E = \left(\frac{C_R \, x \; Capital\ Investment}{8760 \, x \; C_F} \right) + (Levelized\ Fuel\ Cost) + (Levelized\ O\&M\ Cost)$$

- E = Levelized cost of power
- C_R = Levelized capital carrying charge

  = 0.106 (typical of utility industry analyses, for constant dollar calculations)

- C_F = Capacity factor
- Levelized fuel cost, O&M costs, assume constant dollar discount rate of 6.2%, before taxes

DOE/EIA fuel price forecasts out to the year 2010 were used. Typical escalation factors were used for the period 2010 to 2030. The fuel price values used in the power cost projections are given in Table VI-11 and the results of the cost of power comparison was shown earlier in Figure VI-4 (see above).

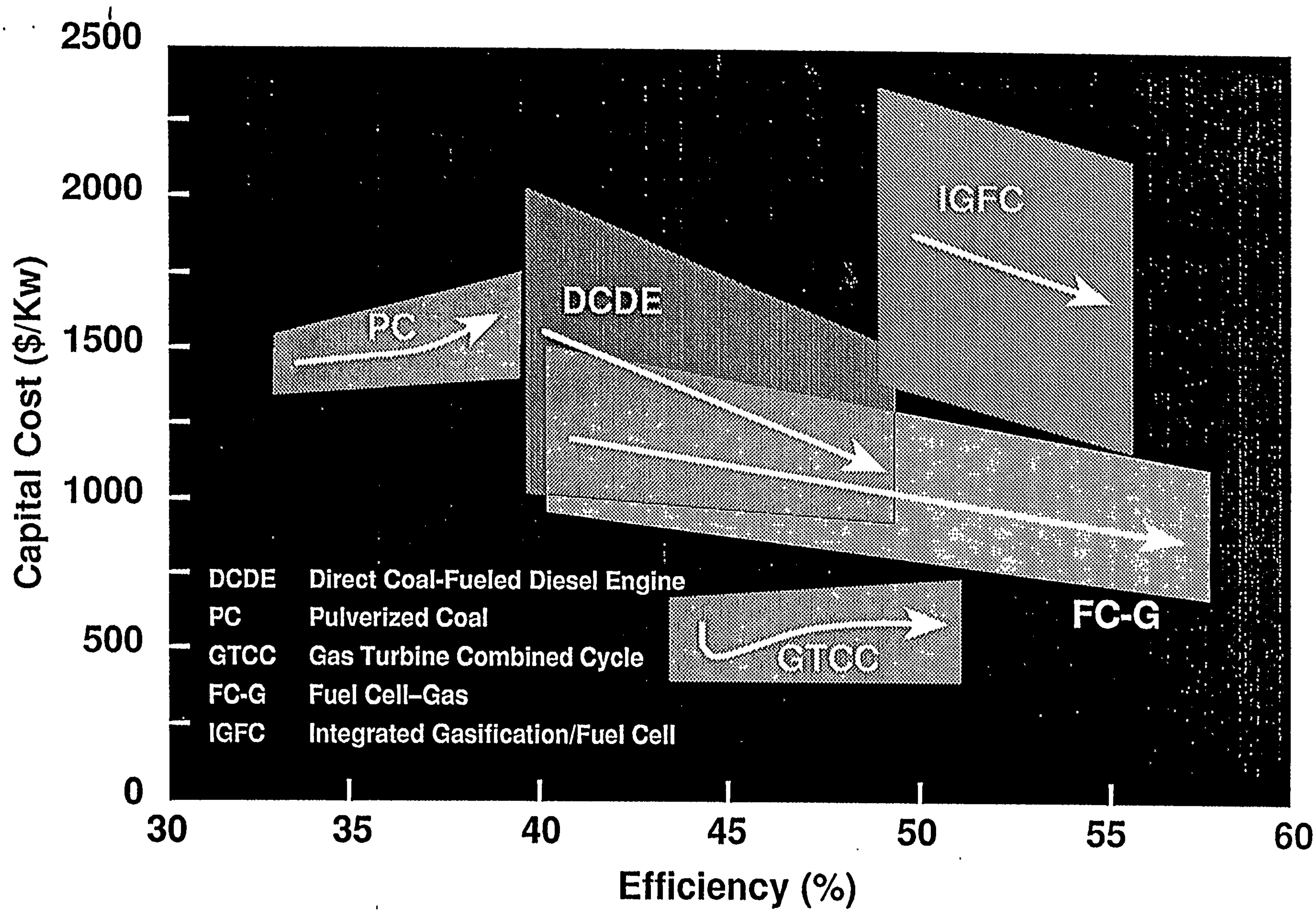


Figure VI-10. Trends in Power Generation Technology Performance, 1992-2010

At current natural gas price levels through about 2000, the calculations shown in Figure VI-4 indicate that clean coal diesel technology is not economically viable as compared to the other alternatives at any of the capacity levels considered. This is consistent with the current situation, whereby clean coal diesels are being assembled on a field test basis in pre-production quantities.

Table VI-11. Fuel Price Projections[1], $/MMBtu, Constant 1990 $

	Natural Gas				
	Baseline Scenario		High Growth Scenario		
Year	1990	1992	1990	1992	Coal
1990	2.51	2.51	2.60	2.51	1.54
1995	3.05	2.67	3.45	2.76	1.67
2000	4.10	3.47	4.71	4.14	1.74
2005	5.02	4.67	5.86	5.26	1.85
2010	5.61	5.78	6.97	6.01	1.97
2015	6.13	6.32	7.60	6.57	2.05
2020	6.70	6.91	8.30	7.27	2.09
2025	7.33	7.55	9.04	7.84	2.13
2030	8.01	8.25	9.87	8.56	2.17

[1]Based on DOE/EIA Annual Energy Outlook 1990 and 1992 through 2010, 1.8% per ;year escalation thereafter for gas prices, 0.4% per year for coal prices.

The analysis indicates that as natural gas prices rise to the $4 to $6/MMBtu level and the clean coal diesel technology matures, the economics of clean coal diesel plants are favored as compared to the alternatives considered, particularly at the lower capacity range (below 50 MW) of primary focus for distributed power applications.

E. Commercialization Potential

1. Target Markets

The basic motivation behind the commercialization effort will be to provide coal-burning heat engine technology primarily for 10-100 MW modular stationary power applications in the next decade and beyond, when oil and gas prices may return to the $5-7/MMBtu range. There are three major target markets for the clean coal diesel technology:

- Non-utility (NUG) new capacity (estimated at up to 1000 MW (20 plants) per year after gas prices rise to the level CWS is competitive).

- Small utility repowering (estimated at up to 800 MW (16 plants) per year).

- Exports to the developing countries of coal technology below 100 MW plant size (estimated at up to 600 MW (12 plants) per year in U.S. export).

These markets are discussed further below. These applications typically involve multiple engines at a single plant where requirements are favorable for coal firing: high annual utilization, existing infrastructure, space for coal handling and emission controls, and constant load operation. In this section we will show that the coal diesel, if successfully demonstrated and pursued in these markets, can capture up to 2400 MW per year (approximately 48 plants) of new capacity in these three markets in the 2010-2030 timeframe. The NUG market appears to be our primary target, with potential coal diesel sales of over 30,000 MW in capacity.

2. NUG Market

The Non-Utility Generation (NUG) phenomenon has grown rapidly over the last five years, and the clean coal technology "industry" until now has not responded with suitable 10-100 MW products for NUG applications. The average coal NUG plant is 38 MW in capacity--only small PC boilers and small CFBs are being installed in these coal applications, and these are not advanced clean coal technologies like the coal diesel. The NUG industry is expected to add 30,000 to 40,000 MW in each ten-year period in the U.S., or between 40%-50% of all new capacity installed.
Table VI-12 shows the breakdown of the estimated NUG market by time period, with estimated market share for coal diesel totalling over 30,000 MW for the period 2005-2030. The primary competition will be CFB units.

Table VI-12. NUG Market is the Primary Opportunity for Coal Diesels

			NUG Additions by Fuel Type, 1000 MW			Market Share Coal NUGs, 1000 MW
Period	Scenario	NUG Total Capacity Additions,* MW	Gas	Other	Coal	Coal Diesel Share
Prior to 2005	Clean Air Act compliance; coal diesel demonstration, but gas prices attractive	48,000	24	19	5	NA
Approx. 2010	Rising gas prices shift balance of NUG orders to clean coal. First CD orders.	18,000 MW every 5 years	2	4	12	800 MW/year
Approx. 2020 and thereafter	Coal diesel growth in share. Gas prices rise further.	40,000 MW every 10 years	4	6	30	1500 MW/year
		Total added: by 2030: 146 MW	34	35	72	Over 30,000 MW

*Estimates for 2010-2030

Recent EIA and NES projections confirm that 48,000 MW of new non-utility capacity is projected in the period 1991-2010, of which the dominant share will be for gas turbine and reciprocating engines. Figure VI-11 shows that coal-fueled heat engines will compete for 2500 MW/year to 4000 MW/year after the year 2005 in the non-utility power sector. This level of engine production activity would revitalize the U.S. large engine industry. The number of large stationary engines produced per year would increase from about 20-40 (current) to 200-400.

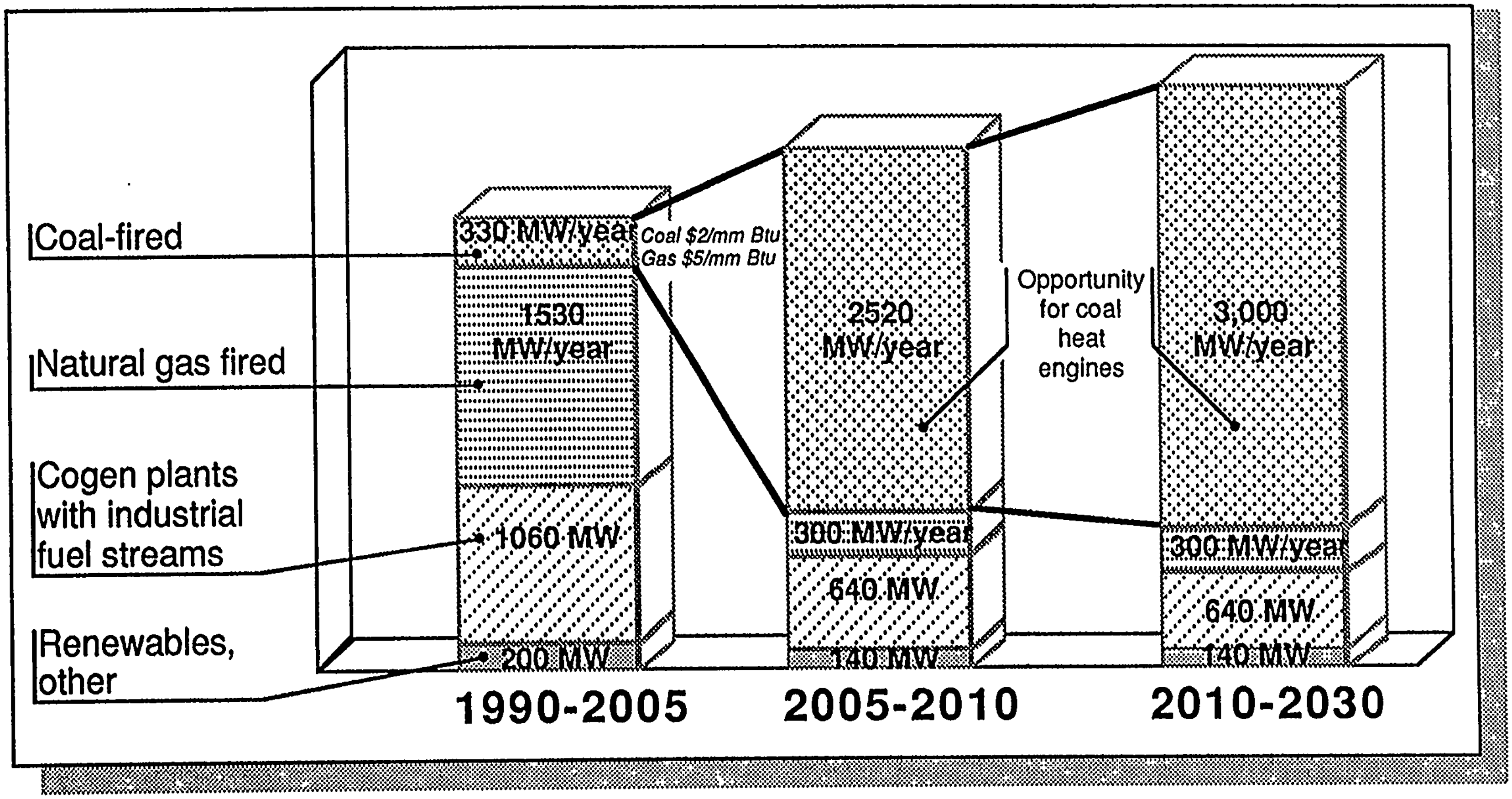
Trends in U.S. power requirements over the next 40 years appear to be highly favorable for the adoption of these coal-fired diesel applications. The coal diesel stationary power plant is being demonstrated at a time when there is an ongoing upheaval in the stationary power industry. This upheaval is in large part driven by the steady growth in U.S. power requirement, which continues to grow at 2.2 to 2.5% per year nationwide, and up to 6% per year in certain regions. For perspective, each extra 0.2% of growth nationwide represents 10,000 MW of new capacity on-line. Another driving force has been the major changes to the infrastructure such as PURPA and the rise of independent power plants (IPPs) and other NUGs. Traditional procedures whereby utilities added stationary power capacity in the form of large central station generators, in increments of 250 to 1300 MW, are gradually giving way to new procedures whereby modular plants are installed in the 10 to 200 MW size range, and many of these are non-utility owned generators (NUGs). At the present time, the fuels of choice for these modular plants are natural gas and oil, due to two factors:

- Gas and oil are currently priced at attractive levels ($2 to $4 million Btu).
- There is no competitive coal-fueled modular power technology available with low emissions yet, particularly in the 10 to 50 MW range.

If the technology can be successfully demonstrated in the field (and when gas prices rise), the coal diesel will become a new option for NUGs which will be competitive with gas and oil as early as 2005-2010.

Later when gas prices rise further and coal is extremely attractive relative to gas, coal diesels will compete with circulating fluid bed coal plants which are somewhat larger in size (30 MW is the minimum; most are over 80 MW), and not as efficient as coal diesel plants would be.

Another type of NUG application is cogeneration. Table VI-13 shows six industries which are expected to expand their cogeneration capacity. Diesel and gas reciprocating engines fit into these cogeneration expansion plans. Table VI-13 shows the potential market for 13000 MW of new capacity in cogeneration above the 10 MW plant size in the 1991-2010 timeframe. This 13,000 MW is part of the 48,000 MW NUG opportunity.



Source: EIA Annual Energy Outlook, 1992 and Arthur D. Little Estimates

Figure VI-11. Opportunities for Coal-Fueled Diesel Power Plants 199-2030

Table VI-13. Number of Existing Manufacturing Plants with Diesel Cogeneration Potential

	Cogeneration Plant Size		
Industry	2.0-9.9 MW	10.0-19.9 MW	20.0 MW and Over
Food	450	48	84
Textile	199	9	6
Chemical	157	65	131
Petroleum Refining	39	8	91
Rubber	48	6	3
Metals	107	19	34
Other	40	15	11
Totals	1,040	170	360

Source: Department of Energy

3. Small Utility Repowering Market

Repowering small electric utility plants (10-100 MW) is also a very promising application for the coal-fueled diesel. Figure VI-12 shows that there are older coal power plants below 100 MW in size in excess of 23000 MW nationwide capacity, most of which will require repowering. A significant shortfall in electric generating capacity is projected to occur beginning in the late 1990's, according to the "most probable" scenario of 2.2%/year demand growth. Utilities have not yet announced their plans for new plants to satisfy this shortfall, or for repowering. Most forecasts show a challenging time period between 1994 and 2000 when available U.S. generating capacity will start to be strained to meet projected demand. Most repowering estimates converge on 10,000 MW worth of repowering occurring during every five-year period. What this means is that there is an unprecedented opportunity for new modular power such as the coal diesel. Table VI-14 shows the small utility repowering market estimates by time period and the potential market share for coal diesels.

For several reasons, utilities are showing interest in modular capacity expansion. There is an increasing problem in finding suitable sites for large central power plants near urban areas, and the capital requirement for each of these large plants is also a problem. Earlier expectations that nuclear plants would grow to relieve this shortfall are now less certain. Modular plants are easier to find sites for, can be put on-line with relatively short lead time, can follow load swings more efficiently, have a lower first cost per kW installed, and, in the case of large diesels, offer efficiencies of 40-45% vs. 34-37% for modern coal steam electric. Proven availability is 92% or greater for large stationary reciprocating engine generator sets.

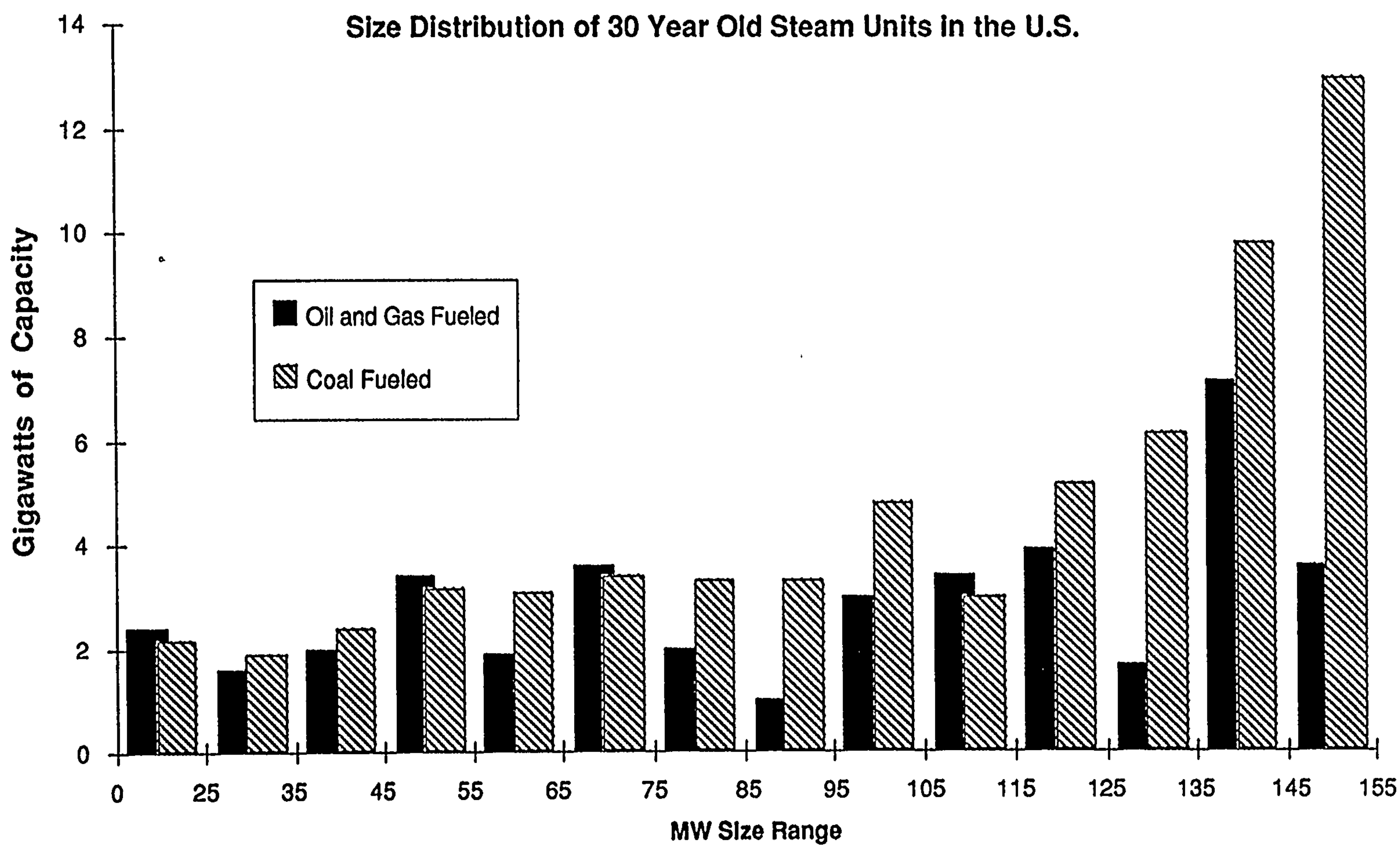


Figure VI-12. Older Coal-Fired Steam Plants Below 100 MW Size Total 23,000 MW

Table VI-14. Market for Small Utility Repowering with Coal Diesels

Period	Scenario	Utility Total Capacity Additions,* MW	Addition s Above 100 MW (30%)	Additions Below 100 MW (30%)	Coal Diesel Below 100 MW Share of Market
Prior to 2005	Clean Air Act compliance; coal diesel demonstration, but gas prices attractive	94,000	66,000	28,000	None
Approx. 2010(?)	Rising gas prices shift balance of NUG orders to clean coal. First CD orders.	25,000 MW every 5 years	17,000	8,000	25%; 400 MW/year
Approx. 2020(?) and thereafter	Coal diesel growth in share. Gas prices rise further.	42,000 MW every 10 years	30,000	12,000	60%; 600 MW/year
		Total added: by 2030: 189,000 MW	133,000	56,000	Up to 15,000 MW

*Based on NES and EIA Annual Energy outlook, 1992. Includes only oil, gas and coal capacity. Includes repowering of 10,000 MW of each five-year period.

Of course, there is uncertainty in both the degree and timing of the upcoming small utility repowering market and in the mix of solutions which will actually be implemented, such as:

- Power imports (from Canada and Mexico) and more extensive wheeling
- Load management and conservation
- Life extension of existing 10-100 MW units

Whatever mix eventually materializes, repowering will be an important element, and this is a key potential future market for the coal diesel powerplant. Coal-fueled heat engines, when developed and demonstrated, must compete against gas-fueled modular power and fluidized bed (coal) plants for share of this 4000-6000 MW per year market.

The projections for new clean coal capacity additions, including both utility and NUGs, are expected to be at a relatively high level in 2005-2030, as was shown in Tables VI-12 and VI-14. The coal diesel is designed to compete for the smaller plant sizes in the 10-100 MW range, and we estimate this market sector to be 28,000 MW between 2005-2030 (30% of the total utility capacity additions). As was indicated in Table VI-12, we estimate that the coal diesel will capture up to 25% market share of all 10-100 MW plants over this twenty-five-year period 2005-2030. This will amount to up to 15,000 MW or about one thousand 15 MW class engines. This is the manufacturing and licensing requirement which Cooper-Bessemer is preparing for.

4. Exports of Clean Coal Power Plants--The International Power Market

The international power market has always been well suited to large stationary diesels. This is because (a) the size of the required power plant fits what diesels offer, (b) maintenance and operation of diesels can be handled by local employees, and (c) the diesels are able to burn heavy fuel oil. For example, 44 diesel engines in the 10-40 MW engine size range were ordered in 1990 alone for the Far East, Central Asia, and Central America. These totalled 700 MW and were all installed for heavy fuel oil usage.

In the next eight years (1992-2000), the international power market is expected to experience the following growth:

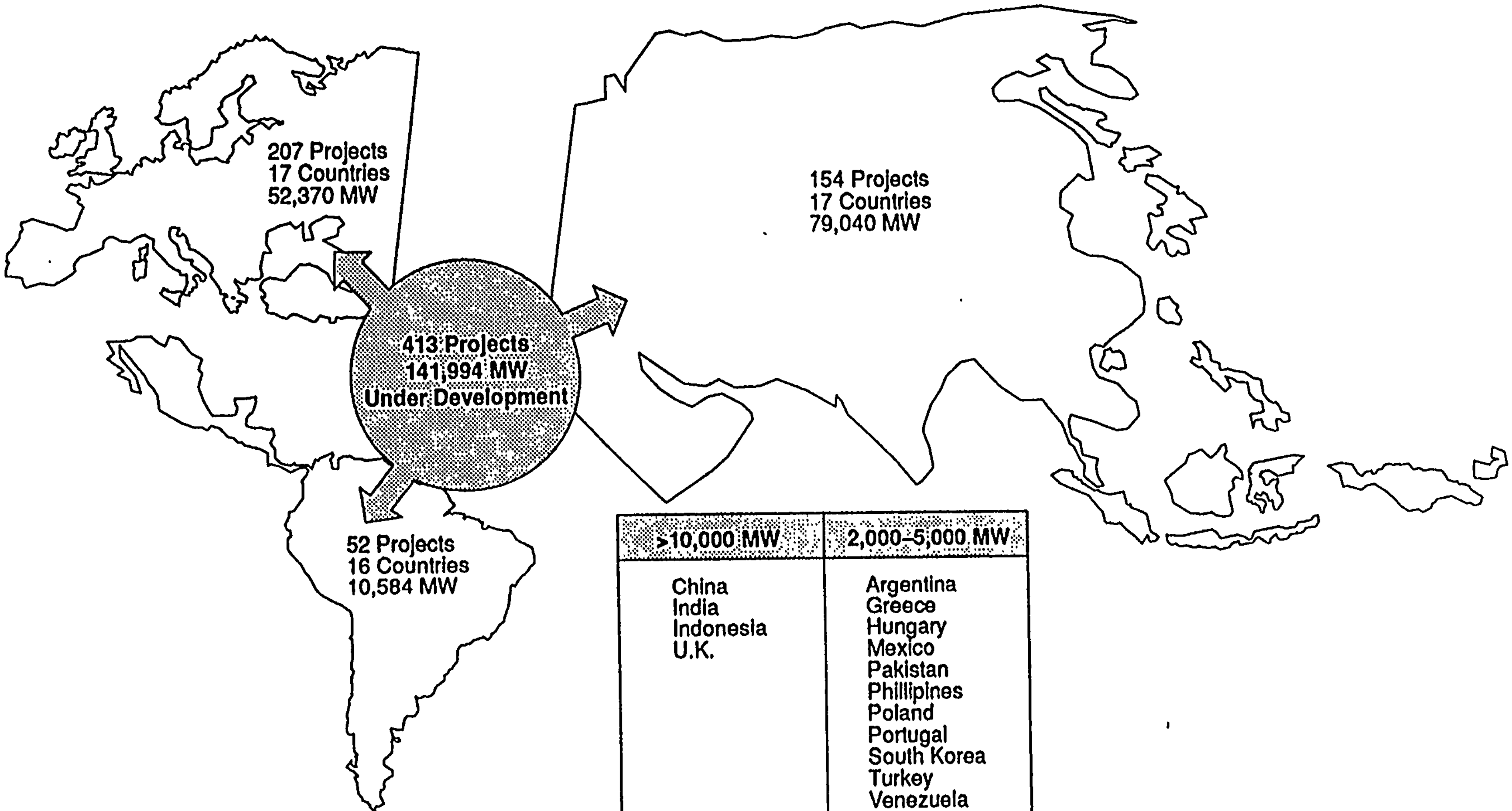
	Total GW	NUG Capacity GW
Latin America	45	10
Far East	220	40
Europe	90	30
Africa	70	5
	425 GW	85 GW (20%)

Source: Independent Energy, 1992.

Indeed, there are currently over 400 overseas projects underway totalling 142,000 MW, of which 46% are coal (70 projects, 60,000 MW). This is illustrated in Figure VI-13. The average NUG (or IPP) project overseas is much larger than in the U.S. (280 MW is the average).

The National Energy Strategy (Technical Annex 5) projected that 1,128,000 MW of capacity will be added overseas during the 20-year period 1991-2010. This is 560,000 MW each 10-year period, roughly consistent with the 425,000 MW listed above for the eight-year period. Table VI-15 shows estimates of the international coal powerplant market totalling 1110 GW through the year 2030. We estimate that of this 1110 GW, 30% will be below 100 MW coal plant size and eligible for the coal diesel.

As for U.S. exports into the international coal powerplant market, DOE has estimated that 10% of the international power market through the year 2000 could be captured by U.S. companies. As indicated in Table VI-15, it is estimated that the U.S. share can rise to 20% by the year 2010. The total potential coal diesel export market is projected to be up to 15,000 MW by 2030.



>10,000 MW	2,000-5,000 MW
China India Indonesia U.K.	Argentina Greece Hungary Mexico Pakistan Phillipines Poland Portugal South Korea Turkey Venezuela

Source: RGC/Hagler, Bailly, Inc.

Figure VI-13. The Independent Power Market Abroad

Table VI-15. International Power Market--An Opportunity for Coal Diesel Exports

				Potential U.S. Coal (Technology Export Activity, Below 100 MW)		Potential Coal Diesel Export Activity
Period	International Arena for U.S. Power Plant Technology	Total Capacity Additions, MW	Assumed Capacity Additions Below 100 MW Size (coal only)	%	MW	Potential Coal Diesel Export Activity
Prior to 2005	U.S. technologies enter international market	800,000 MW	108,000 MW	10*	11,000	Negligible
Approximately 2010	Worldwide gas price hikes shift balance of orders to plants (U.S. CCT technology orders)	300,000 MW every 5 years	45,000 MW every 5 years	15	7,000	70 MW/yr (one or two plants)
Approximately 2020 and thereafter	Gas prices rise further; U.S. CCT growth in share	300,000 MW every five years	45,000 MW every 5 years	20	9,000	600 MW/yr (15% of CCT exports)
		(Total added by 2030: 2,300,000 MW six times the U.S. capacity growth)	352,000 MW		57,000 MW	Up to 15,000 MW

5. Regions of Greatest Potential

Ten specific regions where NUG and small utility capacity additions expected to be greatest are listed in Table VI-16. The coal diesel plant is considered to be particularly appropriate to compete for these future NUG sites because of the size range, efficiency, low emissions, and multi-fuel capability. These utilities are expected to elect coal diesel (for smaller sub-100 MW plants) when the price of oil and gas increases. The ten regions shown in Table VI-16 are the prime target market opportunities for modular power in the 10-100 MW range. Four of these regions in Table VI-16 marked ("high") are judged particularly suitable for clean coal modular power (such as the coal diesel design).

Table VI-16. Regions with Above Average Capacity Additions and Their Potential for CWS Diesel

Region	States	Some Candidate CWS Metro Areas	Potential for 500 MW CWS Diesel Capacity
MACC	Pennsylvania, New Jersey, Maryland	Pittsburgh, Philadelphia, Baltimore	High
WSCC/CNV	California		Low
SERC/VAC	Virginia, Carolinas	DC, Richmond, Raleigh-Durham	High
NPCC/NY	New York	New York	Medium
ECAR	Ohio, Wyoming, Indiana, Kentucky, Michigan	Indianapolis, Detroit, Columbus	High
NPCC/NE POOL	New England	New Haven, Portland, Providence, Boston	Medium
SERC/FL	Florida	Miami, Tampa, Jacksonville	High
SPP/WC	Oklahoma	Tulsa	Low
WSCC/RMP	Colorado	Denver	Low
SERC/SOU	Southern (GA, AL, MS)	Atlanta	Medium

These four regions are as follows:

- PA, NJ, MD ("MAAC")
- VA, NC, SC ("SERC-VAC")
- OH, WV, MI, IN, KY ("ECAR")
- FL ("SERC-FL")

In these regions, there is not only a higher projected demand for new installations of modular power, but also a higher fuel price advantage for coal vis-a-vis natural gas. Also, each of these four regions offer several suitable barge terminals for coal delivery. These are also the regions where the actual electricity growth has exceeded utility forecasts. What this means is that the utilities and NUGs in these regions favor smaller plant additions and repowering rather than build new large central power stations to meet their demand growth. These regions are shown on a U.S. map in Figure VI-14, with the cross hatched areas representing significant potential for NUGs and small utility repowering.

Note that the regions of greatest potential for the coal diesel essentially form a bank about 800 miles wide down the Eastern U.S. This has a significant impact on our coal diesel commercialization plan and selection of source coals.

It should also be noted that environmental control considerations which evolve in each metropolitan area by 1995-2015 will play an important part in determining suitable regional markets for the coal diesel (which has extremely low emissions compared to CFB or Stoker/PC units). This is, choices for modular NUG plants may be driven as much or more by local environmental considerations as by fuel cost (once coal slurry is in the competitive range).

6. Market Share for Coal Diesel Systems--Other Competing Technologies

The market share for coal diesels in the initial market period can be projected based on current trends in new orders for stationary reciprocating engines vis-a-vis turbines. That is, in the initial market entry phase, coal diesels would be expected to capture the same share of new stationary engine orders as diesels do now (once coal slurry is competitive with diesel oil and gas). Below 15,000 kW (15 to 60 MW plant size), diesels and gas recips currently take 58% of the market in North America (42% goes to gas turbines). Table VI-17 shows the scenario whereby coal diesels could capture 500 MW in the U.S. (15 to 20 plants) during the initial five years after gas prices rise to the point that CWS is competitive. The export market is also projected to be robust for coal diesels in this timeframe--up to 220 MW (7 to 9 plants) in this same timeframe.

The earliest applications of coal diesels following the initial demonstration (scheduled to be completed in 2000) will be as dual fuel (gas-CWS) stationary reciprocating engines in the 2000-2010 period. At these early "market entry" sites, the engines will be run on CWS part of the time (for demonstration) and on natural gas the rest of the year. During this period, relative fuel prices will dictate whether the coal diesel is competitive with the large gas-diesel engines running on natural gas and diesel fuel. In this period, commercial interest will depend upon how rapidly natural gas and oil prices rise and how soon the cost of producing clean CWS as engine fuel can be

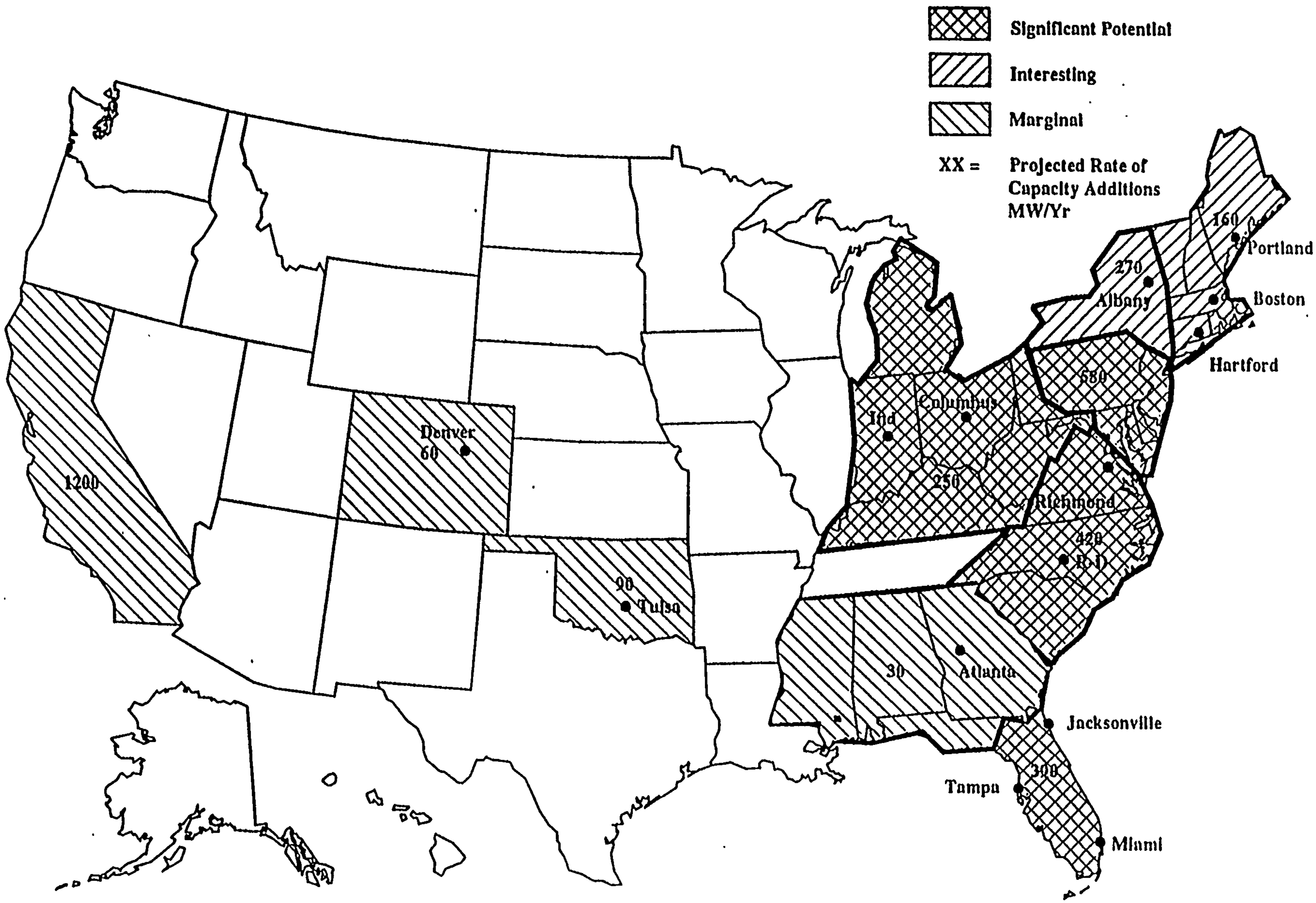


Figure VI-14. U.S. Regions for Potential Coal-Fueled Diesel Applications reduced to the competitive range of $3.00/MMBtu by building up the CWS infrastructure and order books.

In the longer term, once high efficiency and reliability have been demonstrated, the coal diesel will become the preferred option for small, low-emission coal-fueled power (below 100 MW plant size). In addition to merely displacing oil and gas stationary recips, the coal diesel will begin to capture market share from gas turbines, depending on the commercial status of coal-fueled turbines in this size range.

Other advanced coal-power technologies will compete with the direct coal-fired diesel for these future opportunities. These include:

- Conventional steam turbine with fluid-bed coal combustor (CFB units) or PC boiler.

- Closed-cycle air turbine, indirect-fired, with fluid-bed combustor.

- Coal-fueled combined cycle gas turbine systems with pressurized combustor and hot-gas clean-up (Solar, Westinghouse, and Allison have done development work; the prototype designs resemble more and more the full IGCC designs).

The target coal-diesel markets were selected so as to exploit the anticipated competitive advantages of the diesel vis-a-vis these emerging technologies. For example, premium coal-derived fuels ("mild gasification") may be most competitive for smaller engines (below 2000 hp) for high engine RPM (for transportation applications). Coal-fueled gas turbines and steam turbines with fluid beds will be more competitive at 30 MW turbine size and up (on a large enough scale to justify the hot gas clean-up and/or pressurized gasifier technology); this corresponds to 100 MW plant size and up.

All four advanced-coal technologies listed above deserve continued demonstration efforts in order to prepare new options for future U.S. power requirements, including improved air pollution controls. However, our assessment is that neither the fluid-bed steam turbine nor the two gas turbine options will be able to compete with the coal diesel below 15 MW unit engine size (say 50 MW plant size). The coal diesel (CDCC) is an extremely attractive technology for this market segment.

The development of coal diesel market share can be expected to proceed in stages, as shown in Table VI-17.

Table VI-17. Scenario for Coal Diesel Market Penetration

		NUG and Small Utility Market Shares (U.S.) (10-100 MW Plants)			Export Market Share (10-100MW)
Approximate Timeframe	Scenario	Coal Diesel	Gas/Oil Recip	Gas/Oil Gas Turbine/ GTCC	Coal Diesel
1995-2005	Field demonstrations as natural gas/CWS dual fuel engine (CCT-V and other)	NA	3000 MW every 5 years	18000 MW every 5 years	NA
Approximately 2010	As price of gas and oil begins to rise, coal diesel becomes competitive, and captures, say, 1/5 of new orders for stationary recip engines	500 MW in first 5 years	2500 MW in same period	18000 MW in same period	40 MW/yr
Approximately 2020 and thereafter	Gas and oil prices rise further so that clean CWS is now a preferred choice in NUG and small utility markets	2000 MW/yr	200 MW/yr	600 MW/yr	180 MW/yr

7. Commercialization Plan

Cooper-Bessemer is the premier manufacturer of 300-450 rpm stationary engines in the United States, and is the only U.S. manufacturer of 300-450 rpm engines which has successfully operated a test engine on coal water slurry. The company's commercialization track record is borne out by their Model LS engine (1000-6300 kW), which is widely used in small utilities, IPPs, and cogeneration. This model accounts for over 2.5 million installed horsepower capacity. Cooper-Bessemer will lead the commercialization effort by offering the new coal engine to the marketplace through selected leading A&E firms. Those A&Es which specialize in 10-100 MW power plants handle most small utility repowering and new NUG power plant bidding and construction.

The commercialization strategy for the Cooper-Bessemer coal diesel, a new option for 10-100 MW modular power, includes these elements:

- Duel-fuel natural gas/coal and fuel oil coal engines as the "entry" technology (this will allow plant owners to take advantage of oil and natural gas price swings).
- Regional concentration so as to build a network of coal diesel plants with enough "critical mass" to operate a full-scale clean coal slurry plant (100 ton/hr).
- Aggressively promote the new engine's lower cost of electricity and high efficiency (44 to 48%).

- Aggressively promote the new engine's ultra-low emissions, competitive with gas turbines burning natural gas.

- Cooper-Bessemer will support the initial coal diesel installations in the 2000-2010 period with intensive field engineering, just as they have done in the past with new low-emission gas-diesel models.

- Cooper-Bessemer must gear up their production capacity proactively to meet what promises to be a very robust market. Cooper-Bessemer plans to make license arrangements with other engine manufacturers to make the novel coal diesel technology available to a wider set of customers.

- Developing the infrastructure for low cost clean coal slurry processing capacity ($3/MMBtu delivered).

- Exploiting opportunities to export coal diesel to Europe, Far East, and other areas where natural gas prices are expected to rise sooner than the U.S.

Based on Table VI-17, the commercialization scenario for the Cooper-Bessemer coal diesel assumes that new coal engine installations will increase from 10-15 engines (150 MW) per year in the 2005-2010 timeframe to 150-250 engines (2500 MW) per year in the 2010-2020 timeframe. Table VI-18 shows how Cooper-Bessemer plans to meet the increasing demand for coal diesels. Cooper-Bessemer recognizes that this cannot be accomplished using its own manufacturing resources alone. In the peak years, Cooper-Bessemer production capacity was 50 diesel engines (LS Model) per year (in the aggregate, about 200 MW per year), plus 100-125 spark-gas engines for pipeline applications per year. Therefore, Cooper-Bessemer's strategy would be (a) share the technology with Superior and other Cooper-Bessemer Divisions, and (b) to license the coal diesel technology to other engine manufacturers in anticipation that the coal diesel market demand will exceed 200 engines (1000 MW) per year. Cooper-Bessemer will contact a limited number of other diesel engine builders with some current share of the U.S. cogeneration and IPP markets. Finally, Cooper-Bessemer plans to enter the export market using their international resources, as shown in Table VI-19.

Table VI-18. Cooper-Bessemer Plan to Respond to the Future Growth in Coal Diesel Demand

Level of Demand	Cooper-Bessemer Plan
Up to 35 engines per year @ 6 MW	Grove City Manufacturing and assembly plant (capacity 50 engines, plan for 15 gas)
Up to 100 engines per year	Convert one of the other Cooper-Bessemer Division major engine facilities, such as Gardner Denver IMD in Illinois
Up to 200 engines per year	Build dedicated Greenfield engine plant. Cooper-Bessemer Division has a record for taking major steps (e.g., valve plant, turbocharger plant, Superior)
Up to 400 engines per year	License the coal technology to other U.S. engine manufacturer.

Table VI-19. Cooper-Bessemer International Export and License Options for Coal Diesel Engines

Germany	Cooper-Vulcan, marine engine manufacturer 600-13,500 hp (up to 500/year)
France and French speaking countries	Thermodyn/Creusot-Loire
England	Cooper-Bessemer wholly-owned subsidiary in Liverpool capable of recip engine manufacturing
Japan	Kobe Steel, builder of Cooper-Bessemer engines
China	Cooper-Bessemer facility
Worldwide Aftermarket Support	Dusseldorf, Moscow, Liverpool, Dubai, Singapore

Current technology allows diesel (non-coal) engines to operate at greater than 92% availability at full load operation. The key to achieving this level of availability is the development and implementation of a preventative maintenance program to maintain those parts with a lesser life at intervals that preclude unscheduled breakdown. This same philosophy of development would be applied to the coal engine.

During the ten-year period, 2000-2010, as soon as CWS fuel prices are favorable, Cooper-Bessemer plans to introduce coal-fueled diesel systems into selected customers' sites, both new and retrofit installations. Basic development and integrated testing will continue in the Cooper-Bessemer R&D laboratory. However, the physical size of the medium speed engine produces exhaustive testing in the laboratory environment. Therefore, the Cooper-Bessemer approach in the sale of initial models of new units and of conversion kits is to enlist the cooperation of customers and to establish an aggressive field follow-up support program to further enhance engine durability in order to achieve the desired service life between maintenance intervals. We assume total sales of coal diesels will increase to the level of about 150 MW/year by the year 2010, based on the assumptions that gas prices will rise and cogeneration and IPP/small utility installations of reciprocating engines will occur as was projected in Table VI-17 for the period 1995-2010. The commercialization plan is designed so that the coal diesel will be proven and in position to capture a share of the market by the 2005-2010 timeframe. Other engine manufacturers will follow suit and, if necessary, license the technology from Cooper-Bessemer. Years 2005-2015 define the time period when the coal diesel system may become widely used both by utilities for modular power and by industries for cogeneration. Based on Cooper-Bessemer's dominant position in the U.S. large stationary engine market for the last 40 years, it is reasonable to assume that 60% of all coal diesel systems produced from year 2005 to 2030 will be produced by Cooper-Bessemer and its licensees.

Other team members will assist with commercialization. CQ Inc., will support the transition of their coal slurry plant from a 7 ton/hr operation to a 100 ton/hr regional production plant. This is shown in Figure VI-15. Other elements of CQ Inc.'s strategy are as follows:

(a) Slurry for the first power plants: Early production in 2000-2005 will be tied directly to single consumer (the first installed plant). CWS cost could be in the range of \$6.00 to \$8.00/MMBtu because of the small scale of the production plant (15 to 50 tons/hr). To stimulate CWS production and usage at these prices, CQ will seek to develop ways to discount the charges to the first installed power plant until two or three other power plants are built. CQ estimates that a 25 ton/hr slurry plant will produce CWS at \$4.50/MMBtu, which is expected to be competitive with natural gas in 2000-2005. This corresponds to 60 MW capacity (3 x 20 MW plants or 4 x 15 MW plants).

(b) Regional development: CQ's strategy will be to encourage regional development. As more engines are added in a region, each supply source will consider expansion to supply CWS for the new sites as a means of reducing its own fuel cost. Once a supplier is supplying fuel for 10 engines (60 MW) on multiple sites, a regional supply is emerging. To build additional capacity, suppliers will want to have contracts for at least 60% of the capacity in order to secure financing.

(c) Other uses for slurry: Commercialization could be accelerated by demand for CWS of the same specification within the region for other uses:

- Industrial firetube boilers
- Combustion turbines
- Converted industrial oil-fired wall boilers

(d) Equipment cost: CQ, Inc., will seek to lower the capital cost with introduction of lower cost coal micronizers.

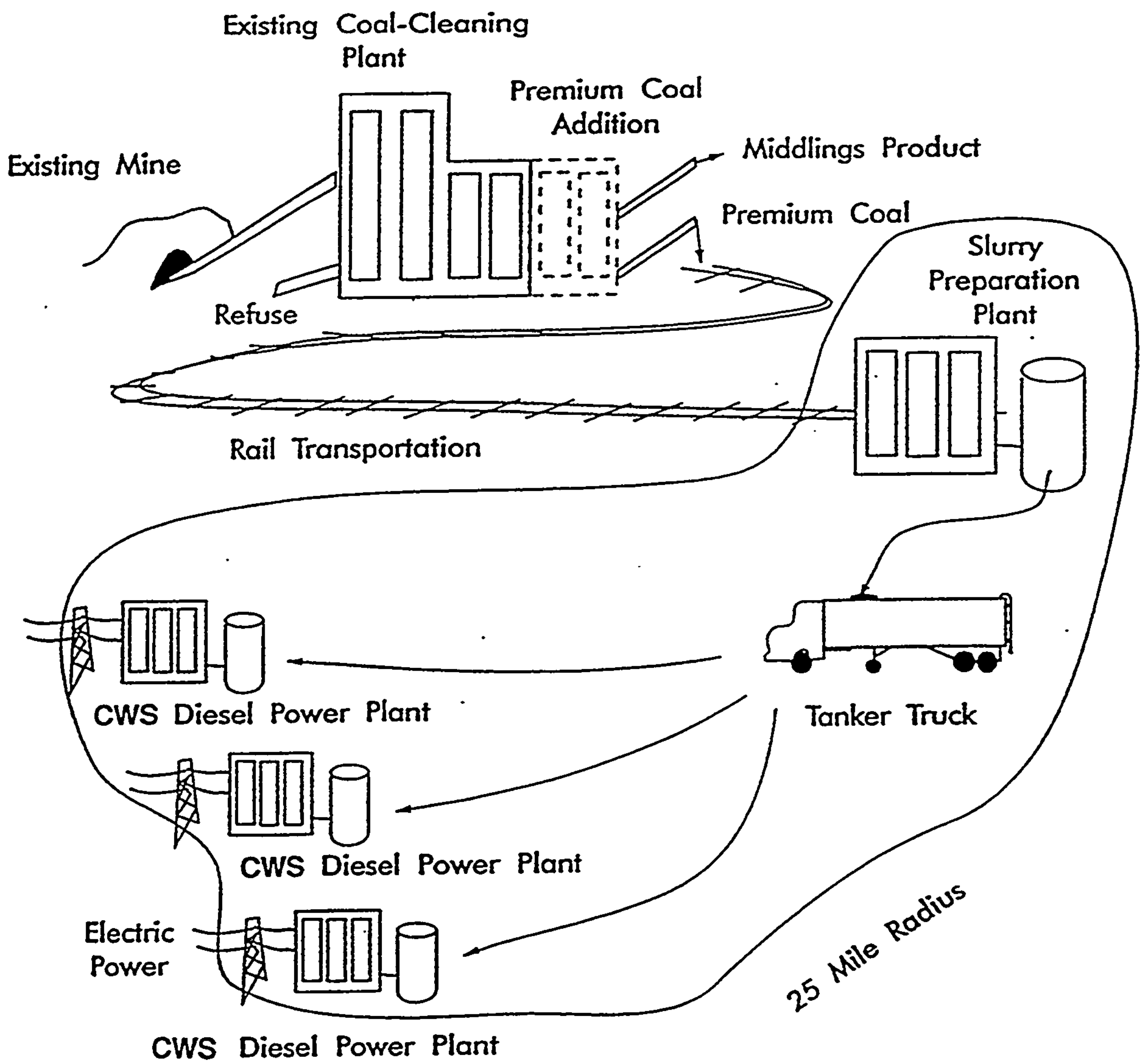


Figure VI-15. Engine-Grade, Coal-Fuel Preparation Strategy

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Appendix A

Coal and Coal-Water-Slurry Analyses for

Coal-Diesel Fuels Prepared by CQ, Inc.

COAL-WATER FUEL FEED STOCK ANALYSIS

CQ Run Num 92091101

I.D. : Coal Feed Stock for First 10,000 gallon Batch of CWF

CLEAN COAL ANALYSIS

CLEAN COAL I.D.:

Source: Clean coal from Wentz, recleaned at CQDC using heavy media cyclone

PARENT COAL I.D.:

Bed Name:	Upper Elkhorn #3
Bed No.:	151
Local Name:	Taggart Seam
Mining Co.:	Westmoreland Coal Co.
Mine Name:	Wentz #1
Cleaning Plant:	Wentz
State:	Virginia
County:	Wise

ASH CHEMISTRY (Wt %)

	In Ash	In Coal	lb/MBtu
SiO2	44.84	0.81	0.53
Al2O3	32.82	0.59	0.39
Fe2O3	12.12	0.22	0.14
CaO	2.99	0.05	0.04
MgO	0.78	0.01	0.01
Na2O	2.48	0.04	0.03
K2O	0.90	0.02	0.01
TiO2	1.74	0.03	0.02
MnO2	0.04	0.00	0.00
P2O5	0.13	0.00	0.00
SO3	1.15	0.02	0.01
Total	100.00	1.80	1.19

ASH FUSION TEMPERATURES (deg. F)

Reducing/Oxidizing Atmosphere		
Initial	2450	2670
Softening	2560	2720
Hemispherical	2650	2755
Fluid	2800 +	2800 +
Variation (deg.)	350	130

PROXIMATE ANALYSIS (dry basis)

Volatile Matter (WT %)	37.91
Fixed Carbon (Wt %)	60.29
Ash (Wt %)	1.80
Sulfur (Wt %)	0.60
Heating Value (Btu/lb)	15,126

Rank mvb

HARDGROVE GRINDABILITY INDEX (Typical)
45

SOLID SPECIFIC GRAVITY
1.29

ULTIMATE ANALYSIS (dry basis)

Carbon (Wt %)	85.26
Hydrogen (Wt %)	5.51
Nitrogen (Wt %)	1.52
Oxygen (Wt % by difference)	5.31

Free Swelling Index 7.5

FORMS OF SULFUR (dry basis)

Pyritic (Wt %)	0.03
Sulfate (Wt %)	0.01
Organic (Wt % by difference)	0.56

COAL-WATER FUEL ANALYSIS

Run Number: 92091101

Quantity: 10,000 gal
I.D.: First 10,000 Gallon Batch of CWF

SLURRY CONTENT

Moisture (Wt %)	47.23
Coal (Wt %)	52.29
Additive Content (Wt %)	0.48

Additive	Type	Wt % of Coal	Wt % of Slurry
Dispersant	MCG 32A-LS	1.00	0.47
Stabilizer	Flocon 4800C	0.015	0.01
Biocide		None	None

SLURRY PROPERTIES

Solids (Wt %)	52.77	By Geochemical Testing
Density (g/cc)	1.13	
Ash (Wt % of Solids)	1.89	By Geochemical Testing
Ash (Wt %)*	0.99	
Sulfur (Wt %)*	0.38	

PARTICLE SIZE ANALYSIS

CQ Inc Microtrac Analysis Summary by CQ Inc.

Mean Diamater--Volume based (MV) =	12.39	microns
Mean Diameter--Area based (MA) =	5.07	microns
100 % less than	88	microns

VISCOSITY MEASUREMENT by Energy International Corp.

LOW SHEAR VISCOSITY (Haake Rotoviscometer)
55 cps @ 100/sec and 70 degree F
68 cps @ 200/sec and 70 degree F

Power Law Factor = 1.236

HIGH SHEAR VISCOSITY (Burrell Viscometer)
Not Determined

*Based on weight of coal in slurry

COAL-WATER FUEL ANALYSIS

Page 1 of 2

Run Number: 93073001

Quantity: 15,300 gal

I.D.: First 5,300 gallons of the 23,000 Gallon Batch of CWF For Cooper-Bessemer's 100 Hour Engine Test

SLURRY CONTENT

Moisture (Wt %)	47.30
Coal (Wt %)	52.21
Additive Content (Wt %)	0.49

Additive	Type	Wt % of Coal	Wt % of Slurry
Dispersant	MCG 32A-LS	1.00	0.47
Stabilizer	Flocon 4800C	0.03	0.01
Biocide		None	None

SLURRY PROPERTIES

Solids (Wt %)	52.70	By Geochemical Testing
Density (g/cc)	1.127	
Ash (Wt % of Solids)	1.52	By Geochemical Testing
pH	n/a	
Ash (Wt %)*	0.79	
Sulfur (Wt %)*	0.36	
Heating Value (Btu/lb)*	n/a	

*Based on weight of slurry

PARTICLE SIZE ANALYSIS

CQ Inc Microtrac Analysis Summary

Mean Diamater--Volume based (MV) =	13.9	microns
Mean Diameter--Area based (MA) =	5.5	microns
100 % less than	88	microns

VISCOSITY MEASUREMENT by Energy International Corp.

LOW SHEAR VISCOSITY (Haake Rotoviscometer)

76 cps @ 100/sec and 70 degree F

92 cps @ 200/sec and 70 degree F

Power Law Factor 1.193

HIGH SHEAR VISCOSITY (Burrell Viscometer)

Not Determined

COAL-WATER FUEL ANALYSIS

Run Number: 93080201

Quantity: 8,000 gal
I.D.: Last 8,000 gallons of the 23,000 Gallon Batch of CWF For Cooper-Bessemer's 100 Hour Engine Test

SLURRY CONTENT

Moisture (Wt %)	48.69
Coal (Wt %)	50.81
Additive Content (Wt %)	0.50

Additive	Type	Wt % of Coal	Wt % of Slurry
Dispersant	MCG 32A-LS	1.00	0.49
Stabilizer	Flocon 4800C	0.03	0.01
Biocide		None	None

SLURRY PROPERTIES

Solids (Wt %)	51.31	By Geochemical Testing
Density (g/cc)	1.127	
Ash (Wt % of Solids)	1.71	By Geochemical Testing
pH	n/a	
Ash (Wt %)*	0.87	
Sulfur (Wt %)*	0.36	
Heating Value (Btu/lb)*	n/a	

*Based on weight of slurry

PARTICLE SIZE ANALYSIS

CQ Inc Microtrac Analysis Summary

Mean Diamater--Volume based (MV) =	15.0	microns
Mean Diameter--Area based (MA) =	5.8	microns
100 % less than	88	microns

VISCOSITY MEASUREMENT by Energy International Corp.

LOW SHEAR VISCOSITY (Haake Rotoviscometer)
70 cps @ 100/sec and 70 degree F
81 cps @ 200/sec and 70 degree F

Power Law Factor 1.138

HIGH SHEAR VISCOSITY (Burrell Viscometer)
Not Determined

CQ Run Number: 92011501

I.D.: 100 Hour Engine Test

CLEAN COAL ANALYSIS

CLEAN COAL I.D.:

Source: Clean coal from Wentz, recleaned at CQDC using heavy media cyclone

PARENT COAL I.D.:

Bed Name:	Upper Elkhorn #3
Bed No.:	151
Local Name:	Taggart Seam
Mining Co.:	Westmoreland Coal Co.
Mine Name:	Wentz #1
Cleaning Plant:	Wentz
State:	Virginia
County:	Wise

ASH CHEMISTRY (Wt %)

	In Ash	In Coal	Btu
SiO2	39.66	0.67	0.44
Al2O3	33.28	0.56	0.37
Fe2O3	15.52	0.26	0.17
CaO	3.31	0.06	0.04
MgO	0.81	0.01	0.01
Na2O	1.30	0.02	0.01
K2O	0.96	0.02	0.01
TiO2	1.55	0.03	0.02
MnO2	0.06	0.00	0.00
P2O5	0.15	0.00	0.00
SO3	2.44	0.04	0.03
Total	99.04	1.67	1.10

ASH FUSION TEMPERATURES (deg. F)

Reducing/Oxidizing Atmosphere		
Initial	2440	2700
Softening	2520	2750
Hemispherical	2560	2800 +
Fluid	2690	2800 +
Variation (deg.)	250	100

PROXIMATE ANALYSIS (dry basis)

Volatile Matter (WT %)	36.85
Fixed Carbon (Wt %)	61.46
Ash (Wt %)	1.69
Sulfur (Wt %)	0.60
Heating Value (Btu/lb)	15,187

Rank mvb

HARDGROVE GRINDABILITY INDEX (Typical)

45

SOLID SPECIFIC GRAVITY (Typical)

1.27

ULTIMATE ANALYSIS (dry basis) (Typical)

Carbon (Wt %)	84.31
Hydrogen (Wt %)	5.69
Nitrogen (Wt %)	1.27

FORMS OF SULFUR (dry basis)

Pyritic (Wt %)	n/a
Sulfate (Wt %)	n/a
Organic (Wt % by difference)	n/a

COAL-WATER FUEL FEED STOCK ANALYSIS

CQ Run Number: 93051701

I.D.: 100 Hour Engine Test

CLEAN COAL ANALYSIS

CLEAN COAL I.D.:

Source: Clean coal from Wentz, recleaned at CQDC using heavy media cyclone

PARENT COAL I.D.:

Bed Name:	Upper Elkhorn #3
Bed No.:	151
Local Name:	Taggart Seam
Mining Co.:	Westmoreland Coal Co.
Mine Name:	Wentz #1
Cleaning Plant:	Wentz
State:	Virginia
County:	Wise

ASH CHEMISTRY (Wt %)

	In Ash	In Coal	Btu
SiO2	42.16	0.63	0.42
Al2O3	32.66	0.49	0.32
Fe2O3	12.06	0.18	0.12
CaO	3.09	0.05	0.03
MgO	0.74	0.01	0.01
Na2O	3.03	0.05	0.03
K2O	2.98	0.04	0.03
TiO2	1.92	0.03	0.02
MnO2	0.02	0.00	0.00
P2O5	0.13	0.00	0.00
SO3	0.68	0.01	0.01
Total	99.47	1.49	0.99

ASH FUSION TEMPERATURES (deg. F)

Reducing/Oxidizing Atmosphere		
Initial	2470	2700
Softening	2510	2770
Hemispherical	2550	2790
Fluid	2620	2800
Variation (deg.)	150	100

PROXIMATE ANALYSIS (dry basis)

Volatile Matter (WT %)	38.70
Fixed Carbon (Wt %)	59.80
Ash (Wt %)	1.50
Sulfur (Wt %)	0.59
Heating Value (Btu/lb)	15,074

Rank mvb

HARDGROVE GRINDABILITY INDEX (Typical)

45

SOLID SPECIFIC GRAVITY (Typical)

1.27

ULTIMATE ANALYSIS (dry basis)

Carbon (Wt %)	86.14
Hydrogen (Wt %)	5.62
Nitrogen (Wt %)	1.64

FORMS OF SULFUR (dry basis)

Pyritic (Wt %)	0.02
Sulfate (Wt %)	0.00
Organic (Wt % by difference)	0.57

COAL-WATER FUEL FEED STOCK ANALYSIS

Run Number: 93040201

I.D.: CWF for Second 24 Hour Test
Dry Solids from a Composite CWF Sample

CLEAN COAL ANALYSIS

CLEAN COAL I.D.:

Source: Clean coal from
Wentz, recleaned at CQDC
using heavy media cyclone

PARENT COAL I.D.:

Bed Name:	Upper Elkhorn #3
Bed No.:	151
Local Name:	Taggart Seam
Mining Co.:	Westmoreland Coal Co.
Mine Name:	Wentz #1
Cleaning Plant:	Wentz
State:	Virginia
County:	Wise

ASH CHEMISTRY (Wt %)

	In Ash	In Coal	lb/MBtu
SiO2	42.31	0.76	0.50
Al2O3	29.73	0.53	0.35
Fe2O3	15.38	0.28	0.18
CaO	3.48	0.06	0.04
MgO	0.93	0.02	0.01
Na2O	2.90	0.05	0.03
K2O	0.80	0.01	0.01
TiO2	1.69	0.03	0.02
MnO2	0.07	0.00	0.00
P2O5	0.18	0.00	0.00
SO3	3.22	0.06	0.04
Total	100.69	1.80	1.19

ASH FUSION TEMPERATURES (deg. F)

Reducing/Oxidizing Atmosphere		
Initial	2230	2490
Softening	2340	2560
Hemispherical	2570	2620
Fluid	2710	2740
Variation (deg.)	480	250

PROXIMATE ANALYSIS (dry basis)

Volatile Matter (WT %)	36.55
Fixed Carbon (Wt %)	61.66
Ash (Wt %)	1.79
Sulfur (Wt %)	0.65
Heating Value (Btu/lb)	15,101

Rank mvb

HARDGROVE GRINDABILITY INDEX (Typical)

45

SOLID SPECIFIC GRAVITY (Typical)

1.27

ULTIMATE ANALYSIS (dry basis)

Carbon (Wt %)	85.89
Hydrogen (Wt %)	5.70
Nitrogen (Wt %)	1.57

FORMS OF SULFUR (dry basis)

Pyritic (Wt %)	0.05
Sulfate (Wt %)	0.01
Organic (Wt % by difference)	0.59

COAL-WATER FUEL ANALYSIS

Run Number: 93040201

Quantity: 5200 gal

I.D.: CWF Stored in Tank 'A'
Sampled April 20, 1993
CWF for Second 24 Hour Test

SLURRY CONTENT

Moisture (Wt %)	49.04
Coal (Wt %)	50.45
Additive Content (Wt %)	0.51

Additive	Type	Wt % of Coal	Wt % of Slurry
Dispersant	MCG 32A-LS	1.00	0.49
Stabilizer	Flocon 4800C	0.03	0.02
Biocide		None	None

SLURRY PROPERTIES

Solids (Wt % of Slurry)	50.96	By CQ Inc.
Density (g/cc)	1.128	
Ash (Wt % of Solids)	1.79	By Geochemical Testing
Ash (Wt % of Slurry)	0.90	
Sulfur (Wt % of Slurry)	0.33	

PARTICLE SIZE ANALYSIS

CQ Inc Microtrac Analysis Summary by CQ Inc.

Mean Diamater--Volume based (MV) =	13.25	microns
Mean Diameter--Area based (MA) =	5.26	microns
100 % less than	88	microns

VISCOSITY MEASUREMENT by Energy International Corp.

LOW SHEAR VISCOSITY (Haake Rotoviscometer)

68 cps @ 100/sec and 70 degree F

80 cps @ 200/sec and 70 degree F

Power Law Factor 1.146

HIGH SHEAR VISCOSITY (Burrell Viscometer)

Not Determined

COAL-WATER FUEL ANALYSIS

Run Number: 92091101

Quantity: 2300 Gallons
I.D.: CWF Stored in Tank 'B'
Sampled April 20, 1993

SLURRY CONTENT

Moisture (Wt %)	50.43
Coal (Wt %)	49.05
Additive Content (Wt %)	0.52

Additive	Type	Wt % of Coal	Wt % of Slurry
Dispersant	MCG 32A-LS	1.00	0.50
Stabilizer	Flocon 4800C	0.03	0.01
Biocide		None	None

SLURRY PROPERTIES

Solids (Wt % of Slurry)	49.57	By CQ Inc.
Density (g/cc)	1.128	
Ash (Wt % of Solids)	1.89	By Geochemical Testing
Ash (Wt % of Slurry)	0.93	
Sulfur (Wt % of Slurry)	0.36	

PARTICLE SIZE ANALYSIS

CQ Inc Microtrac Analysis Summary by CQ Inc.

Mean Diamater--Volume based (MV) =	12.59	microns
Mean Diameter--Area based (MA) =	5.19	microns
100 % less than	88	microns

VISCOSITY MEASUREMENT by Energy International Corp.

LOW SHEAR VISCOSITY (Haake Rotoviscometer)

n/a cps @ 100/sec and 70 degree F

n/a cps @ 200/sec and 70 degree F

Power Law Factor n/a

HIGH SHEAR VISCOSITY (Burrell Viscometer)

Not Determined

Appendix B

Chronology of Cooper-Bessemer LSC-1 and LSC-6 Engine Test Results

July 1991 - July 1993

LS Engine Test Results (July 1991)
(All data reported for 375 rev/min, 146 psi bmep, 275°F MAT)

Purpose of Test	First Experiment on LSC-1 Engine Using CWS Fuel		Test the 32mm L'Orange Injection Pump and 2nd Sapphire Nozzle Tip	
Date	11 July	12 July	17 July	18 July
CWS duration (hr)	0.3 + 1	1.5	1.6 + 1.1 = 2.7	1.5
CWS fuel	Otisca #6 1.4% ash; 48% solids	Same	Same	Same
Injection configuration	Sapphire insert nozzle tip #1 37x0.35mm holes 36mm jerk pump	Sapphire insert nozzle tip #1 37x0.35mm holes 36mm jerk pump	Sapphire nozzle tip #2 37x0.35mm holes 32mm jerk pump	Sapphire nozzle tip #2 37x0.35mm holes 32mm jerk pump
Port closure	28° BTC	Approx 32° BTC	27°BTC	27°BTC
Main injection Start/end/duration	-17/5/22	-21/7/28	-17/12/29	-19/9/28
Main injection pressure (psi)	12500	13500	14000	14000
Peak cylinder pressure (psi)	1200	1420	1280	1280
Peak heat release rate (Btu/deg)	11	11	8	8
Exhaust Temperature (°F)	700	630	630	580
IHP (CWS Cylinder)	280	290	260	210
ISFC (Btu/ihp•hr)	-	5700	6000	5800
Approx BSFC (CWS + pilot)	-	7000	7600	7400
Comments	• The maiden CWS test was successful • Sapphire nozzle tip survived LS engine test • Max rack setting achieve during test was 38.5mm • 20° BTC port closure yielded ignition delay > 20 degrees; pump timing was advanced to 28° BTC on July 11 and then to 32° BTC on July 12. • Load limited by peak cylinder pressure (limit=1400 psi)		• 32mm pump did not provide the IHP achieved by the 36mm pump • 32mm pump test had a higher injection line pressure. 11 nozzle holes were found plugged after the run.	

LS Engine Test Results (Sept 1991)
(All data reported for 375 rev/min, 275°F MAT)

Purpose of Test	Full Load Tests CQI Fuel Tests				
Date	24 Sept	25 Sept	26 Sept	27 Sept	30 Sept
CWS Duration (hr)	3.0	0.2	1.7	1.3	1.8
CWS Fuel	CQI 52% solids	CQI 52% solids	CQI plus Triton x114	Same	Same
Injection Configuration	Sapphire insert nozzle tip 37x0.457mm holes 36mm jerk pump (20° BTC port closure)	Sapphire insert nozzle tip 37x0.457mm holes 36mm jerk pump (27° BTC port closure)	Sapphire insert nozzle tip 37x0.457mm holes 36mm jerk pump	Sapphire nozzle tip 37x0.457mm holes 36mm jerk pump	Sapphire nozzle tip 37x0.457mm holes 36mm jerk pump
Pilot Configuration	DF2 Jet Cell	DF2 Jet Cell	DF2 Jet Cell	DF2 Direct Pilot	DF2 Direct Pilot
Main Injection Start/end/duration	-13	N.A.	-22/4/26	-21/5/26	-20/7/27
Main Injection pressure (psi)	9000		9000	9000	9000
Peak cylinder pressure (psi)	940		1490	1580	1580
Peak heat release rate (Btu/deg)	5		17	19	19
Exhaust Temperature (°F)	890		1060	1080	1090
IHP (CWS Cylinder)	222		475	516	532
Approx BSFC (CWS + pilot)	N.A.		7900	7400	7100
Comments	• 20° BTC pump timing (13° BTC start of main injection) prevented full load operation • Low peak pressure and mostly late burning	• CWS packed solid at check valve • CWS appeared to interact w/DF2 (during overnight shutdown) • Test run was too short to obtain performance data	• Full load was achieved from CWS cylinder (increased nozzle needle lift and even clamping solved fuel delivery problem) • Addition of Triton x114 solved fuel interaction problem • DF2 jet cell or direct pilot injection can be used as an ignition aid		

LS Engine Test Results (Oct 1991)
(All data reported for 375 rev/min, 275°F MAT)

Purpose of Test	Effect of Injection Rate (Smaller Jerk Pump Size and Nozzle Hole Size)		
Date	2 Oct 91		27 Sept 91 (for comparison)
CWS Duration (hr)	1.3		1.3
CWS Fuel	CQI plus Triton x 114 approx. 52% solids		Same
Injection Configuration	Sapphire insert nozzle tip 37x0.385mm holes 32mm jerk pump (27 BTC port closure) KEY CHANGE		Sapphire insert nozzle tip 37x0.457mm holes 36mm jerk pump
Pilot Configuration	DF2 Direct Pilot		Same
Main injection Start/end/duration	-22/8/30	-20/15/35	-21/5/26
Main injection pressure (psi)	10,600	11,000	9000
Peak cylinder pressure (psi)	1400	1,460	1580
Exhaust Temperature (°F)	850	940	1080
IHP (CWS Cylinder)	370	470	520
Approx BSFC (CWS + pilot)	7700	7200	7400
Comments	• Smaller jerk pump increased injection duration and decreased injection rate. • Smaller nozzle holes increased injection pressure. • Peak cylinder pressure was reduced (1460 vs. 1580 psi). • Exhaust temperature was reduced (940°F vs. 1080°F).		

LS Engine Test Results (January 1992)
(All data reported for 400 rev/min, 275°F MAT - Engine at 175 psi bmep)

Purpose of Test	Injection Configuration Optimization: Number and Size of Nozzle Holes, Spray Angle, Injection Rate, and Injection Timing						
Date	14 Jan 92	15 Jan 92	15 Jan 92	16 Jan 92	16 Jan 92	17 Jan 92	21 Jan 92
CWS Duration (hr)	3.9	2.5	2.0	1.0	1.6	2.0	1.1
CWS Fuel	Baseline CQI plus Triton x 114 (approx 1.8% ash; 51% solids)	Same	Same	Same	Same	Same	Same
Injection Configuration	Sapphire insert nozzle tip (19 x 0.533mm dia holes) 120 degree spray angle 36mm injection pump	Same except 155 degree spray angle	Same except 140 degree spray angle	Same except 32mm injection pump	Same except 120 degree spray angle	Same except 155 degree spray angle	Sapphire insert nozzle tip 25x0.533mm dia holes) 155 deg spray angle 32mm injection pump
Injection Timing	23 BTC port closure	Same	Same	Same	Same	Same	Same
Pilot Configuration	Twin DF2 pilot injectors (120mm³/stroke/injector)	Same	Same	Same	Same	Same	Same
Main Injection Start/end/duration (crank degrees)	-21/6/27	-21/6/27	-21/4/25	-21/5/26	-21/11/32	NA	-20/5/25
Main Injection Pressure (psi)	10000	10000	10000	10000	10500	NA	8500
Peak Cylinder Pressure (psi)/std. dev. (psi) @ peak location (degrees ATC)	1470/18 @ 15	1500/28 @ 15	1540/16 @ 12	1470/11 @ 11	1420/26 @ 14	1470/10	1510/14 @ 13
Exhaust Temperature (°F)	900°F	960°F	870°F	740°F	840°F	740°F	820°F
CWS Fuel Rate (lb/hr)/DF2 Fuel Rate (lb/hr)	394/645	388/603	344/646	264/680	318/665	283/675	
IHP (CWS Cylinder) IHP (CWS) as % of Normal IHP/Cylinder (418 hp)	520±20 124	525±20 125	460±20 110	360±20 86	420±20 100	370±20 88	430±20 103
Approx BSFC (CWS + Pilot) Based on Assumed 60 FHP	6700±300	6400±300	6700±300	6600±300	6900±300	7000±300	6500±300
Comments	• Repeat run (Dec. 19) to determine effect of failing rings in December • BSFC significantly better (6700 vs. 7700). This improvement may have come in part from correcting the ring problem. • Higher IHP for the same rack setting (520 vs. 460)	• Peak cylinder pressure somewhat sensitive to spray angle		• Could not achieve full load (200 psi bmep) without exceeding peak pressure limit. At 21 BTC start of injection, and with this 32mm pump (35% reduction in instantaneous rate of injection), the burn rate was faster and led to higher peak pressure. • Rack limit for this pump was marginal.			

LS Engine Test Results (January 1992)
(All data reported for 400 rev/min, 275°F MAT - Engine at 175 psi bmep)

Purpose of Test	Injection Configuration Optimization: Number and Size of Nozzle Holes, Spray Angle, Injection Rate, and Injection Timing						
Date	22 Jan 92		22 Jan 92	23 Jan 92	23 Jan 92	24 Jan 92	24 Jan 92
CWS Duration (hr)	1.4		1.0	1.7	2.0	1.4	1.7
CWS Fuel	Same		Same	Same	Same	Same	Same
Injection Configuration	Same		Same except 120 degree spray angle	Sapphire insert nozzle tip 19x0.533mm dia holes 155 degree spray angle 32mm injection pump	Same except 140 degree spray angle	Sapphire insert nozzle tip 31x0.533mm dia holes 155 degree spray angle 32mm injection pump	Same except 120 degree spray angle
Injection Timing	21 BTC port closure		Same	Same	Same	Same	Same
Pilot Configuration	Same		Same	Same	Same	Same	Same
Main Injection Start/end/duration (crank degrees)	-18/6/24	-18/13/31	-17/14/31	-18/18/36	-17/18/35	-18/12/30	-18/14/32
Main Injection Pressure (psi)	8500	8500	9000	11000	11000	8000	8000
Peak Cylinder Pressure (psi)/std. dev. (psi) @ peak location (degrees ATC)	1430/17 @ 14	1500/18 @ 13	1350/23 @ 17	1470/15 @ 12	1480/14 @ 13	1360/22 @ 17	1293/27 @ 19
Exhaust Temperature (°F)	810°F	930°F	950°F	940°F	940°F	970°F	1030°F
CWS Fuel Rate (lb/hr)/DF2 Fuel Rate (lb/hr)	302/675	378/643	377/646	381/648	378/644	402/632	476/620
IHP (CWS Cylinder) IHP (CWS) as % of Normal IHP/Cylinder	420±20 100	510±20 122	480±20 115	500±20 120	490±20 117	510±20 122	510±20 122
Approx BSFC (CWS + Pilot) Based on Assumed 50 FHP	6500±300	6600±300	7000±300	6800±300	6800±300	7000±300	8100±300
Comments	• 2 degree injection retard dropped peak pressure approx. 80 psi without bsfc penalty.		• Acceptable performance and good bsfc.				• High fuel consumption observed; could be due to plugged nozzle hole or ring wear. Further diagnosis in progress.

LS Engine Test Results (February 1992)
(All data reported for 400 rev/min, 275°F MAT - Engine at 175 psi bmep)

Purpose of Test	Injection Configuration Optimization: Number and Size of Nozzle Holes, Spray Angle, Injection Rate, and Injection Timing					
Date	14 Feb 92	14 Feb 92	18 Feb 92	19 Feb 92	20 Feb 92	21 Feb 92
CWS Duration (hr)	1.7	2.0	1.7	2.0	2.7	2.6
CWS Fuel	Baseline CQI plus Triton x 114 (approx 1.6% ash; 51% solids)	Same	Same	Same	Same	Same
Injection Configuration	Sapphire insert nozzle tip 19 x 0.633mm dia holes 140 degree spray angle 36mm injection pump	Same except 155 degree spray angle	Same except 26x 0.533 mm holes 155° spray angle	Same except 17 holes, 0.635mm 155° spray angle	Same except 19 holes 0.633mm 155° spray angle 32mm injection pump	Same
Injection Timing	21 BTC port closure	Same	Same	Same	Same	Same
Pilot Configuration	Twin DF2 pilot (120mm³/stroke/injector)	Same	Same	Same	Same	Same
Main Injection Start/end/duration (crank degrees)	-16/10/26	-17/9/26	-16/5/21	-18/7/25	-15/13/28	-14/19/33
Main Injection Pressure (psi)	10500	9500	10800	10950	11600	10800
Peak Cylinder Pressure (psi)/std. dev. (psi) @ peak location (degrees ATC)	1590/16	1460/23	1568/- @ 14	1557/16 @ 13	1421/32 @ 15	1428/25 @ 18
Exhaust Temperature (°F)	970°F	960°F	880°F	883°F	938°F	942°F
CWS Fuel Rate (lb/hr)/DF2 Fuel Rate (lb/hr)	369/623	382/630	347/648	345/646	380/632	382/635
IHP (CWS Cylinder) IHP (CWS) as % of Normal IHP/Cylinder (418 hp)	460±20 110	450±20 107	370±20 88	370±20 88	430±20 102	430±20 102
Approx BSFC (CWS + Pilot) Based on Assumed 50 FHP	7100±300	7600±300	8700±300	8600±300	7900±300	7900±300
Comments	• Acceptable performance and good bsfc • Peak cylinder pressure sensitive to spray angle (similar to Jan. 15 results)		• Could not achieve full load without exceeding peak pressure limit		• Baseline performance deteriorating - bsfc increasing - Indicated horsepower decreasing • Cylinder tear down showed rings were worn and liner was damaged by the piston crown	

LS Engine Test Results (March 1992)
(All data reported for 400 rev/min, 275°F MAT - Engine at 175 psi bmep)

Purpose of Test	Injection Configuration Optimization: Number and Size of Nozzle Holes, Spray Angle, Injection Rate, and Injection Timing Using "Slow Rate" Injection Cam									
Date	10 Mar 92	11 Mar 92	17 Mar 92	18 Mar 92	18 Mar 92	19 Mar 92	20 Mar 92	23 Mar 92	25 Mar 92	27 Mar 92
CWS Duration (hr)	2.0	2.7	2.1	2.2	1.9	2.5	3.6	5.0	3.6	5.5
CWS Fuel: Baseline CQI plus Triton x 114 (approx. 1.8% ash; 51% solids)	Same	Same	Same	Same	Same	Same	Same	Same	Same	Same
Injection Configuration: Sapphire insert nozzle tip; 38mm injection pump; slow rate cam (LSC-16-1H)	19 x 0.533mm dia holes 140 degree spray angle	Same except 0.633mm dia holes	Same except 17x0.635mm holes 155° spray angle	Same except 140° spray angle	Same except 19 x 0.533mm holes	Same except 25 x 0.533mm holes 140° spray angle	Same except 155° spray angle	Same except 19x0.635mm holes 140° spray angle	Same except 19x0.533mm holes	Same
Injection Timing	21 BTC port closure	Same	Same	Same		Same	Same	Same	23 deg BTC port closure	Same
Pilot Configuration	Twin DF2 pilot (120mm³/stroke/injector)	Same	Same	Same		Same	Same	Same	Same	Same
Main Injection Start/end/duration (crank degrees)	•	-16/26/42	-15/26/41	-12/23/35	-15/26/41	-14/26/40	-13/27/40	-15/24/39	-16/24/40	-15/20/35
Main Injection Pressure (psi)	10000	7500	9000	9000	11000	9000	9000	8000	10500	10500
Peak Cylinder Pressure (psi)/std. dev. (psi) @ peak location (degrees ATC)	1410/31	1340/24 @ 18	1330/21 @ 16	1360/22 @ 21	1370/18	1320/19 @ 18	1350/14 @ 16	1310/28 @ 19	1440/21 @ 13	1440/21 @ 17
Exhaust Temperature (°F)	1060°F	1170°F	1110°F	1100°F	1040°F	1080°F	1080°F	1140°F	1020°F	1020°F
CWS Fuel Rate (lb/hr)/DF2 Fuel Rate (lb/hr)	445/614	514/606	469/612	460/604	440/616	467/606	462/610	508/604	447/616	449/606
IHP (CWS Cylinder) IHP (CWS) as % of Normal IHP/Cylinder (416 hp)	500±20	490±20	480±20	520±20	470±20	470±20	470±20	480±20	450±20	490±20
Approx BSFC (CWS + Pilot) Based on Assumed 60 FHP	7900±300	9200±300	8900±300	8000±300	8300±300	8700±300	9100±300	9100±300	8800±300	7900±300
Comments	• Slow rate cam resulted in longer injection duration than original CWS injection cam (approximately 60% longer duration) • Larger holes resulted in lower injection pressure (7500 vs. 10000 psi) and lower peak cylinder pressure, but significantly higher bsfc (9200 vs. 7900) and higher exhaust temperature		• 17 hole performance similar to 19 hole • 140 degree spray angle has lower bsfc than 155 degree spray angle (8000 vs. 8900)		• Repeat run of 10 March with similar combustion performance	• 140 degree spray angle has better performance than 155 degree spray angle		• Repeat run of 11 March with similar results	• Check valve plugged during the run	• 2 degree timing advance increased peak pressure slightly (1440 vs. 1410) with no impact on bsfc (compared to 10 March)

LS Engine Test Results (April 1992)
(All data reported for 400 rev/min, 275°F MAT - Engine at 175 psi bmep)

Purpose of Test	Injection Configuration Optimization: "Fast Rate" Injection Cam
Date	30 April 92
CWS Duration (hr)	1.7
CWS Fuel	Baseline CQI plus Triton x 114 (approx. 1.8% ash; 51% solids)
Injection Configuration	Sapphire insert nozzle tip; 19x0.633mm dia holes 140 degree spray angle 36mm injection pump; fast rate cam (LSC-16-1C)
Injection Timing	23 BTC port closure
Pilot Configuration	Twin DF2 pilot (120mm^3/stroke/injector)
Main Injection Start/end/duration (crank degrees)	-21/3/24
Main Injection Pressure (psi)	10000
Peak Cylinder Pressure (psi)/std. dev. (psi) @ peak location (degrees ATC)	1440/15 @ 17
Exhaust Temperature (°F)	860°F
CWS Fuel Rate (lb/hr)/DF2 Fuel Rate (lb/hr)	340/650
IHP (CWS Cylinder) IHP (CWS) as % of Normal IHP/Cylinder (418 hp)	423±20 101%
Approx BSFC (CWS + Pilot) Based on Assumed 50 FHP	7300 ± 300
Comments	• Repeat run of 15 Jan 92 • Slightly lower and later peak pressure (1440 vs. 1540 psi) • Slightly higher bsfc (7300 vs. 6700)

LS Engine Test Results (May 1992)
(All data reported for 400 rev/min, 275°F MAT - Engine at 175 psi bmep)

Purpose of Test	1. Reduced stabilizer CWS formulation 2. Internal oiler modification				Combustion performance confirmation of proposed LSC-6 injection configuration	
Date	1 May 92	4 May 1992	5 May 1992	6 May 1992	7 May 1992	8 May 1992
CWS Duration (hr)	1.9	0.9	2.7	2.6	1.8	0.6
CWS Fuel	Baseline CQI plus Triton x 114 (approx. 1.8% ash; 51% solids)	CQI fuel from 500 gal tank (530 hrs mixing time)	Same	Same	Baseline CQI plus Triton x 114 (approx 1.8% ash; 51% solids)	Same
Injection Configuration	Sapphire insert nozzle tip; 17x0.633mm dia holes 140 degree spray angle 36mm injection pump; fast rate cam (LSC-16-1C)	Same	Same	Same except reduced coal side shuttle land	Same except 19 hole x 0.633mm dia holes	Same
Injection Timing	23 BTC port closure	Same	Same	Same	26 BTC port closure	Same
Pilot Configuration	Twin DF2 pilot (120mm³/stroke/injector)	Same	Same	Same	Same	Same
Main Injection Start/end/duration (crank degrees)	-14/12/26	-14/6/20	-14/11/25	-16/15/31	-18/4/22	-18/0/18
Main Injection Pressure (psi)	10,500	10,000	10,500	11400	10,000	9600
Peak Cylinder Pressure (psi)/std. dev. (psi) @ peak location (degrees ATC)	1430/26 @ 17	1330/32 @ 19	1430/37 @ 19	1500/32 @ 19	1510/24 @ 16	1430/22 @ 15
Exhaust Temperature (°F) CWS Rack Setting	860°F 41.5	710 32	830 41.5	910 48	790 36	NA 32
CWS Fuel Rate (lb/hr)/DF2 Fuel Rate (lb/hr)	334/661	258/687	327/652	392/622	305/655	278/675
IHP (CWS Cylinder) IHP (CWS) as % of Normal IHP/Cylinder (418 hp)	390±20 93%	320±20 77%	420±20 100%	500±20 120%	410±20 98%	350±20 85%
Approx BSFC (CWS + Pilot) Based on Assumed 50 FHP	7700 ± 300	7500 ± 300	7100 ± 300	6900 ± 300	6700 ± 300	7200 ± 300
Comments	• 17 hole tip had similar performance to 19 hole tip (April 30 comparison)	• Part load operation only on May 4	• Reduced stabilizer concentration had slightly better bsfc (7100 vs. 7700) than standard CWS formulation	• Shuttle modification improved internal oiler operation	• Repeat of January 15 configuration and results confirm 30 April run • Selected as best configuration for LSC-6	• Part load operation only on May 8

LSC-6 Engine Test Results (Sept 1992)
(All data reported for 400 rev/min, 275°F MAT - Engine at 150 psi bmep)

Purpose of Test	Initial Six-Cylinder CWS Test	Diesel Baseline
Date	18 Sept 92	18 Sept 92
CWS Duration (hr)	1 + 0.4	-
CWS Fuel	Baseline CQI plus Triton x 114 (approx. 1.8% ash; 49% solids)	DF2 only
Injection Configuration	Sapphire insert nozzle tip; 19x0.633mm dia holes 140 degree spray angle 36mm injection pump; fast rate cam (LSC-16-1C)	Same
Injection Timing	23 BTC port closure	Same
Pilot Configuration	Twin DF2 pilot (120mm³/stroke/injector)	Same (except 60 mm³/stroke/injector)
Peak Cylinder Pressure (psi)/std. dev. (psi)	900	1320
Cylinder Exhaust Temperature (°F)	870 (range 850 to 900)	930
CWS Fuel Rate (lb/hr)/DF2 Fuel Rate (lb/hr)	1864/54	754 (DF2 only)
BHP BHP/cyl	1890 315	1890 315
BSFC (Btu/hp•hr)	7800 ± 300 (HHV) 7500± 300 (LHV)	7300 ± 200 (LHV)
NO_x (ppm) CO (ppm) CO_2 O_2	460 300 7.5% 11.5%	1180 2100 5.8% 12.3%
Comments	• Load limited to 150 psi bmep by governor	• Diesel comparison using same injection system and timing as CWS run

LSC-6 Engine Test Results (November 1992)
(All data reported for 400 rev/min, 275°F MAT)

Purpose of Test	Initial Six-Cylinder CWS Test	Full Load Six-Cylinder CWS Tests Plus Integrated Emission Control Test	
Date	18 Sept 92	24 Nov 92	
CWS Duration (hr)	1 + 0.4	2.5	
CWS Fuel	Baseline CQI plus Triton x 114 (approx. 1.8% ash; 49% solids)	Same	
Injection Configuration	Sapphire insert nozzle tip; 19x0.633mm dia holes 140 degree spray angle 36mm injection pump; fast rate cam (LSC-16-1C)	Same	
Injection Timing	23 BTC port closure	22 BTC port closure	
Pilot Configuration	Twin DF2 pilot (120mm³/stroke/injector)	Same	
Load (psi bmep)	150	175	200
Peak Cylinder Pressure (psi)/std. dev. (psi)	900	1400	1600
Cylinder Exhaust Temperature (°F)	870	890	920
CWS Fuel Rate (lb/hr)/DF2 Fuel Rate (lb/hr)	1864/54	2019/59	2283/59
BHP BHP/cyl	1890 315	2200 367	2520 420
BSFC (Btu/hp•hr)	7800 ± 300 (HHV) 7500 ± 300 (LHV)	7050 (HHV) 6750 (LHV) - 40.5%	7000 (HHV) 6700 (LHV) - 40.7%
NO_x (ppm) engine out CO (ppm) CO_2 O_2	460 ppm 300 ppm 7.5% 11.5%	850 ppm 150 ppm - 10.8%	1020 ppm - - 10.6%
NO_x (after SCR)	SCR not operated	84 ppm (0.13 lb/MMBtu)	100 ppm (0.16 lb/MMBtu)
Comments	• Listed for reference • Load limited to 150 psi bmep by governor	• Successful full load operation with integrated emission control system. NO_x levels achieved 90% reduction from engine-out levels by using SCR • Excellent BSFC • "Invisible" plume	

LSC-6 Engine Test Results (March 1993)
(All data reported for 400 rev/min)

Purpose of Test	Durability/reliability of key components prior to 24-hr test; stack emission characterization; injection timing optimization and part load performance; consumption of existing CWS fuel supply								
Date	9 Mar 93			10 Mar 93			11 Mar 93		
CWS Duration (hr)	5.7			3.3			3.6		
CWS Fuel	CQI plus Triton x-114 (approx 1.8% ash; 44% solids)			Same			Same		
Injection Configuration	Sapphire insert nozzle tip; 19x0.633mm dia holes 140 degree spray angle 36mm injection pump; fast rate cam (LSC-16-1C)			Same			Same except adjust rack linkage on cyl 6 main injection pump		
Injection Timing	22 BTC port closure			Same except cyl 1 & 3 main injection pump timing retarded 1 crank degree			Same		
Pilot Configuration	Twin DF2 pilot (120mm³/stroke/injector)			Same			Same		
Load (psi bmep)	150	175	200	150	175	200	150	175	200
Peak Cylinder Pressure (psi)/std. dev. (psi)	940	1160	1440	930	1210	1440	950	1180	1440
Cylinder Exhaust Temperature (°F)	875	900	930	900	920	945	910	935	945
CWS Fuel Rate (lb/hr)/DF2 Fuel Rate (lb/hr)	2051/49	2278/49	2549/48	2052/52	2334/51	2540/52	2040/53	2303/53	2583/57
BHP BHP/cyl	1887 314	2201 367	2516 419	1887 314	2201 367	2516 419	1887 314	2201 367	2516 419
BSFC (Btu/hp·hr)	7800(HHV) 7450(LHV)	7400(HHV) 7050(LHV)	7150(HHV) 6800(LHV)	7900 7550	7600 7250	7250 6900	7800 7450	7500 7150	7300 6950
NO_x (ppm) CO (ppm) CO_2 O_2 engine out; NO_x after SCR & bag-house was 90-95% lower	460 290 7.1 11.8	600 210 7.8 11.1	680 200 8.2 10.7	480 290 7.9 •	600 250 8.1 10.7	680 190 8.4 10.5	470 340 7.6 11.4	580 250 7.9 11.0	670 230 8.3 10.5
Comments	• Steady CWS operation • Peak firing pressure on cylinders 1 and 3 slightly higher than average; cylinder 6 significantly lower than average • Cylinder 6 pilot injector (1 of 2) stuck open			• Cylinder 1 and 3 peak firing pressure dropped • Cylinder 6 peak pressure still low			• Cylinder 6 main pump adjustment restored cylinder 6 performance • Excellent cylinder to cylinder balance • Integrated emission control system operating • Ready for 24-hr test!		

LSC-6 Engine Test Results (April)
(All data reported for 400 rev/min)

Purpose of Test	"Around the clock" CWS Endurance Run							
Date	30 Mar 93		31 Mar 93			1 April 93		
CWS Duration (hr)	4.7		12.5			7.6		
CWS Fuel	CQI plus Triton x114 (approx 1.8% ash) 50% solids)		Same			Same		
Injection Configuration	Sapphire insert nozzle tip; 19x0.633mm dia holes 140 degree spray angle 36mm injection pump; fast rate cam (LSC-16-1C)		Same			Same		
Injection Timing	22 BTC port closure		Same			Same		
Pilot Configuration	Twin DF2 pilot (120mm³/stroke/injector)		Same			Same		
Load (psi bmep)	150	175	175	200	175	175	200	150
Brake Power (bhp)	1887	2201	2201	2516	2201	2201	2516	1887
Peak Cylinder Pressure (psi)	940	1180	1200	1410	1160	1200	1420	1000
Cylinder Exhaust Temperature (°F)	890	930	930	960	930	930	960	960
CWS Fuel Rate (lb/hr)/DF2 Fuel Rate (lb/hr)	1750/53	1980/53	2023/53	2276/52	2057/52	2075/52	2323/53	1982/52
BSFC (Btu/hp•hr)	7200(LHV) 7550(HHV)	6900 7250	7050 7400	6850 7200	7200 7550	7250 7600	7050 7400	7800 8150
NO_x / CO / O_2 — Engine out	580 ppm (1.6 lb/MBtu) 330 ppm 11.0%	790 (2.2) 340 10.9	810 (2.2) 370 10.6	860 (2.4) 460 10.3	745 (2.0) 460 10.9	740 (2.0) 530 11.0	840 (2.3) 530 10.5	610 (1.5) 740 11.2
NO_x (lb/MBtu) After Emission Control System	NA	NA	0.13	NA	0.16	0.16	0.23	0.23
Comments	• DF2 fuel line fitting broke, temporarily halting tests.		• No problems.			• Completed 20 hr continuous operation (started on 31 March) w/planned shutdown. • Slight degradation in bsfc observed compared to start of test (approx. 3 to 5%). • Three nozzle tip inserts out of total 114 were found fractured at end of test (2 in cyl 2 tip and 1 in cyl 5 tip), which may have contributed to increased bsfc and higher exhaust temp. for those cylinders.		

LSC-6 Engine Test Results (May)
(All data reported for 400 rev/min)

Purpose of Test	CWS Endurance Run								
Date	4 May 93			5/6 May 93			25/26 May 93		
CWS Duration (hr)	4.3			4.2			15.4		
CWS Fuel	CQI plus Triton x114 (approx 1.8% ash) 50% solids)			Same			Same		
Injection Configuration	Sapphire insert nozzle tip (new); 18x0.633mm dia holes 140 degree spray angle 36mm injection pump; fast rate cam (LSC-16-1C)			Same			Same		
Injection Timing	Nom. 22 BTC port closure			Same			18 BTC port closure (4 degrees retard)		
Pilot Configuration	Twin DF2 pilot (120mm³/stroke/injector)			Same			Same		
Load (psi bmep)	150	175	200	150	175	200	150	175	200
Brake Power (bhp)	1887	2201	2516	1887	2201	2516	1887	2201	2516
Peak Cylinder Pressure (psi)	1010	1270	1460	1120	1400	1660	970	1270	1540
Cylinder Exhaust Temperature (°F)	870	890	950	840	860	890	830	860	890
CWS Fuel Rate (lb/hr)/DF2 Fuel Rate (lb/hr)	1707/52	1944/52	2218/52	1662/52	1891/52	2130/52	1760/54	1957/54	2188/50
BSFC (Btu/hp•hr)	7050(LHV) 7400(HHV)	6850 7150	6750 7100	6900 7200	6650 7000	6500 6800	7250 7600	6900 7200	6650 6950
NO_x CO Engine out O_2	530 ppm (1.5 lb/MMBtu) 220 ppm 11.6 %	675 (1.9) 190 11.1%	- - -	- - -	- - -	- - -	500 (1.4) 240 12.6	680 (1.9) 130 12.3	730 (2.0) 110 12.0
NO_x (lb/MBtu) After Emission Control System	NA	NA					NA	0.32	0.39
Comments	• New 18 hole nozzle tips (no center hole). • Shutdown to fix exhaust leaks and check cylinder #2 injector. • Additional exhaust leaks developed during testing--decision made to run remainder of test without cyclone.			• Operation without cyclone (at the same injection timing) increased turbo boost pressure and flow rate, increased peak firing pressure and improved BSFC 3 to 4%. • Turbocharger bearing failed (unrelated to CWS operation).			• Completed 15-hour continuous operation with planned shutdown. • No significant turbocharger deterioration observed running without cyclone. • 4 degree injection timing retard reduced peak firing pressure to within acceptable range for operation without cyclone (2 to 5% fuel consumption penalty due to injection timing retard).		

LSC-6 Engine Test Results (June)
(All data reported for 400 rev/min)

Purpose of Test	CWS Endurance Run						
Date	15 June 93			17 June 93			
CWS Duration (hr)	2.2			12.3			
CWS Fuel	CQI plus Triton x114 (approx 1.8% ash; 50% solids)			Same			
Injection Configuration	Sapphire insert nozzle tip (new); 18x0.633mm dia holes 140 degree spray angle 36mm injection pump; fast rate cam (LSC-16-1C)			Same			
Injection Timing	Nom. 18 BTC port closure			Same			
Pilot Configuration	Twin DF2 pilot (120mm³/stroke/injector)			Same			
Load (psi bmep)	150	175	200	150	175	200	208
Brake Power (bhp)	1887	2201	2516	1887	2201	2516	2616
Peak Cylinder Pressure (psi)	950	1110	1300	960	1190	1500	1540
Cylinder Exhaust Temperature (°F)	890	940	990	850	900	930	910
CWS Fuel Rate (lb/hr)/DF2 Fuel Rate (lb/hr)	1770/55	2044/54	2310/56	1746/59	1987/56	2243/59	•
BSFC (Btu/hp•hr)	7350 (LHV) 7700 (HHV)	7150 7500	7050 7400	7250 7600	7000 7350	6900 7200	• •
NO_x CO Engine out O_2	520 ppm (1.5 lb/MMBtu) 280 ppm 11.5%	640 (1.8) 250 10.7%	• • •	470 (1.3) 270 12.0%	620 (1.7) 180 11.5%	750 (2.1) 170 11.1	• • •
NO_x (lb/MBtu) After Emission Control System	•	•	•	•	0.28 to 0.39	0.41 to 0.46	•
Comments	• Operation with cyclone. • Exhaust cyclone elbow cracked during testing--decision made to run remainder of test without cyclone.			• Operation without cyclone (at the same injection timing) increased turbo boost pressure and flow rate, increased peak firing pressure and improved BSFC approximately 2%. • Results similar to May 25/26 operation without cyclone. • Achieved 208 psi bmep on CWS (rated bmep for the LSB on diesel fuel).			

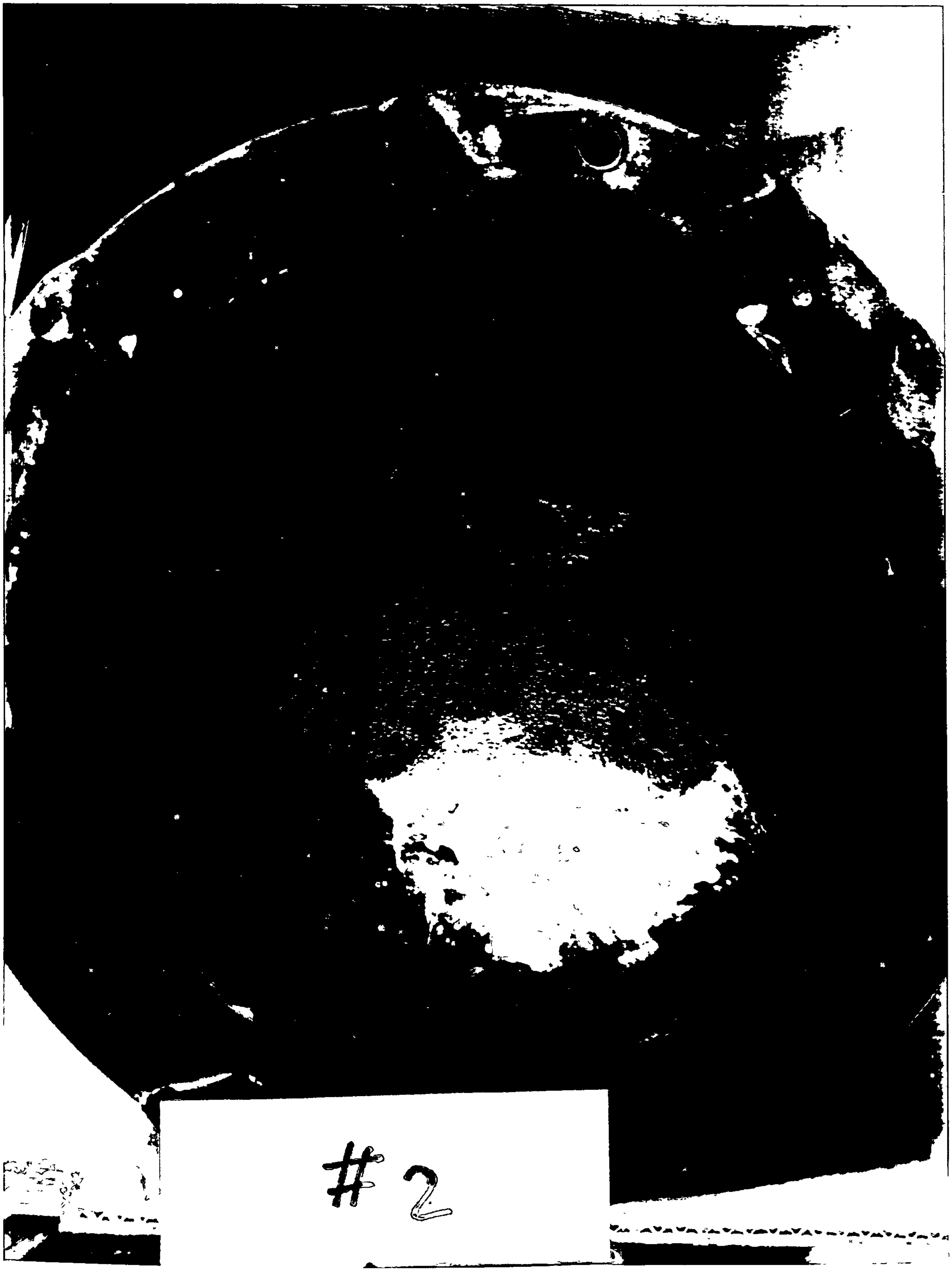
LSC-6 Engine Test Results (July 1993)
(All data reported for 400 rev/min)

Purpose of Test	CWS Test: 100 hour Test Preparation	
Date	15 July 93	17 June 93 (listed for comparison to July 15)
CWS Duration (hr)	3.0	12.3
CWS Fuel	CQI plus Triton x114 (approx 1.8% ash; 50% solids)	Same
Injection Configuration	Sapphire insert nozzle tip (new); 18x0.633mm dia holes 140 degree spray angle 36mm injection pump; fast rate cam (LSC-16-1C)	Same
Injection Timing	Nom. 18 BTC port closure	Same
Pilot Configuration	Twin DF2 pilot ($120 mm^3$/stroke/injector)	Same
Load (psi bmep)	150	150
Brake Power (bhp)	1887	1887
Peak Cylinder Pressure (psi)	NA	960
Cylinder Exhaust Temperature (°F)	890	850
CWS Fuel Rate (lb/hr)/DF2 Fuel Rate (lb/hr)	1774/53	1746/59
BSFC (Btu/hp•hr)	7450 (LHV) 7800 (HHV)	7250 7600
NO_x / CO / O_2 Engine out	500 ppm (1.4 lb/MMBtu) 280 ppm 11.5%	470 (1.3 lb/MMBtu) 270 12.0%
NO_x (lb/MBtu) After Emission Control System	0.31 (78% reduction)	-
Comments	• Engine performance after inspection and rebuild matches previous performance results.	• Listed for comparison to 15 July 1993.

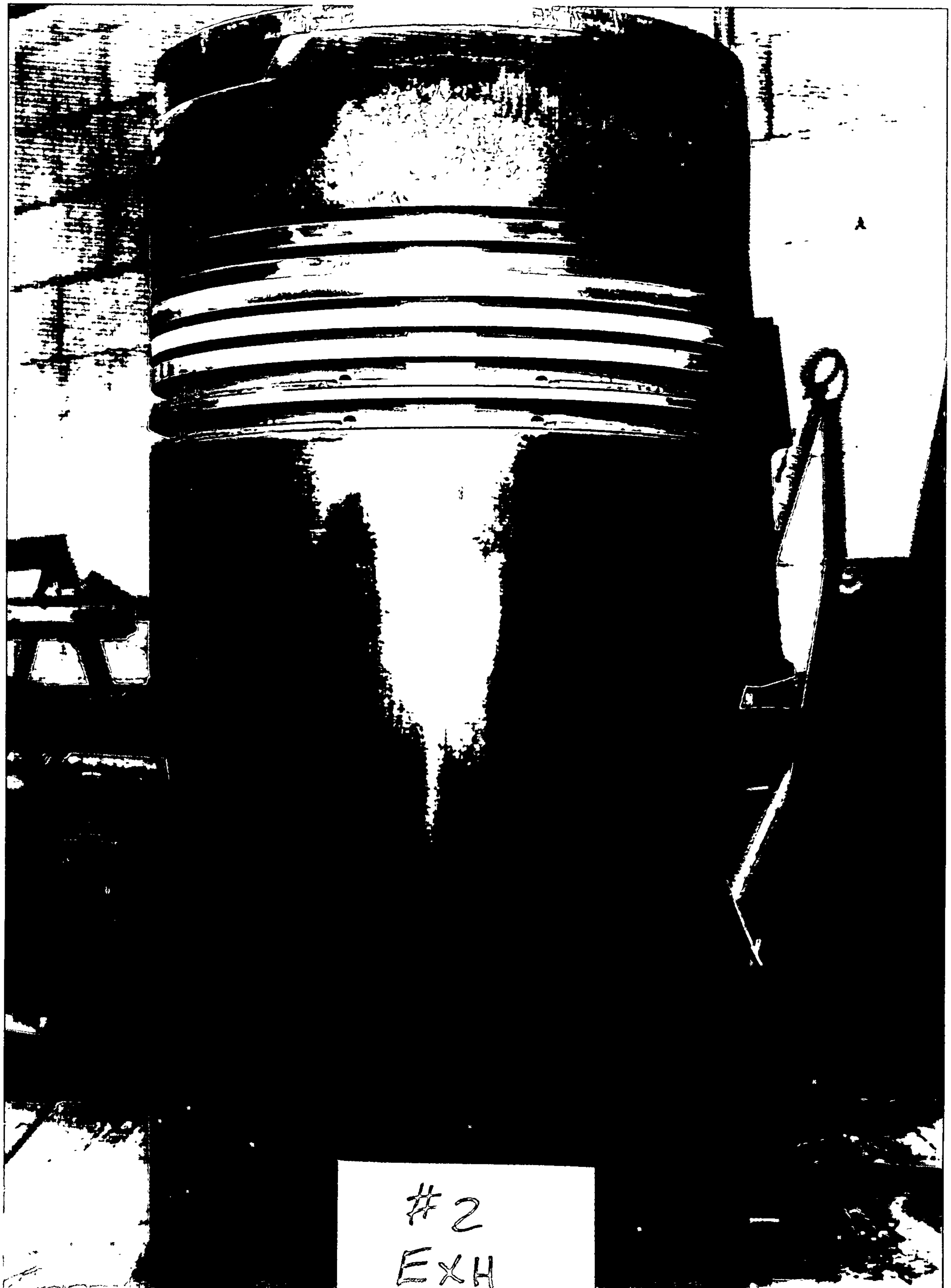
Appendix C

Photographs of Cooper-Bessemer LSC-6 Engine and Turbocharger Components After 100-Hour Endurance Test

Photograph of Cooper-Bessemer LSC Engine Cylinder Head After 100-Hour Endurance Test

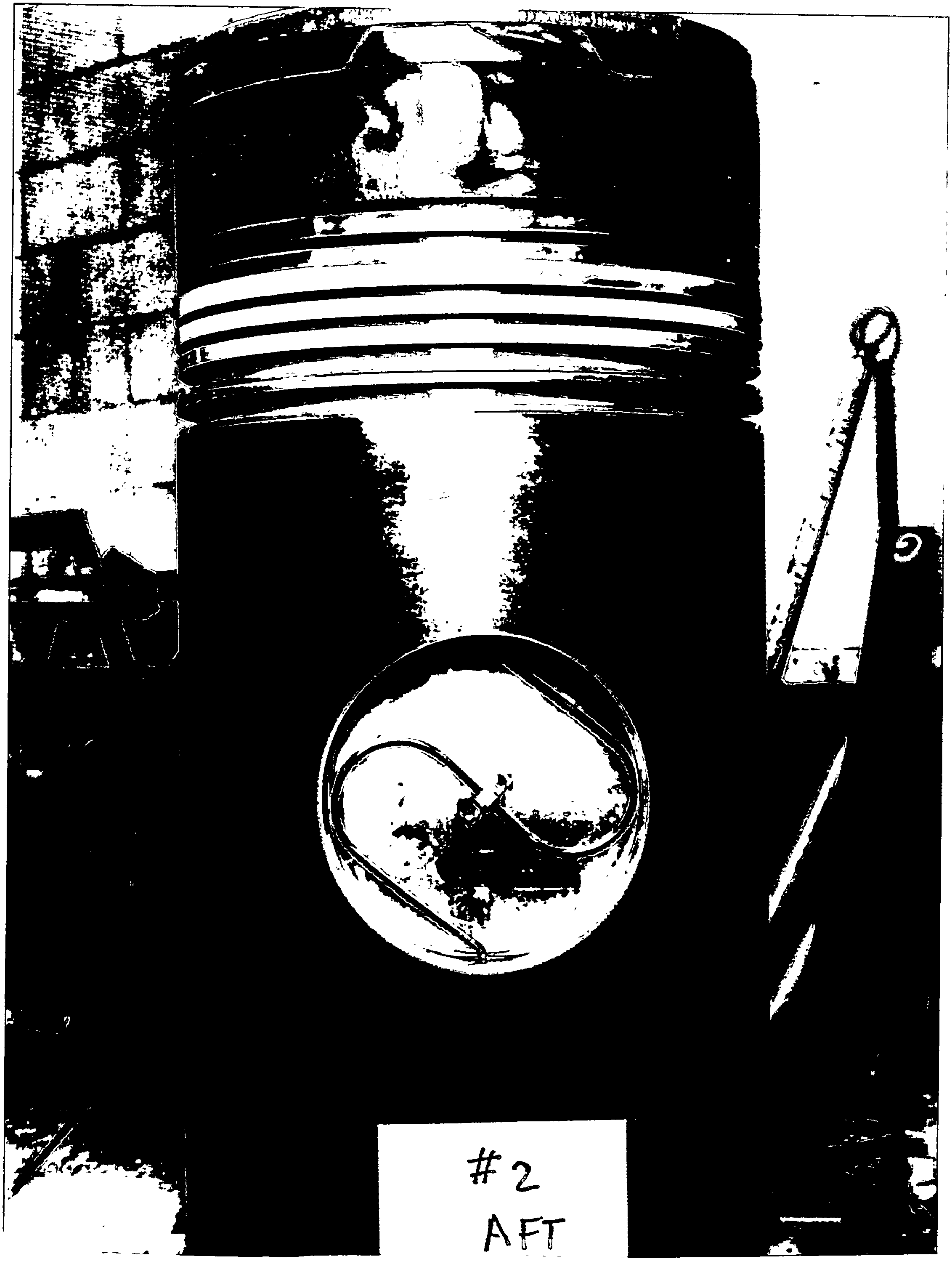


Photograph of Cooper-Bessemer LSC Engine Piston After 100-Hour Endurance Test

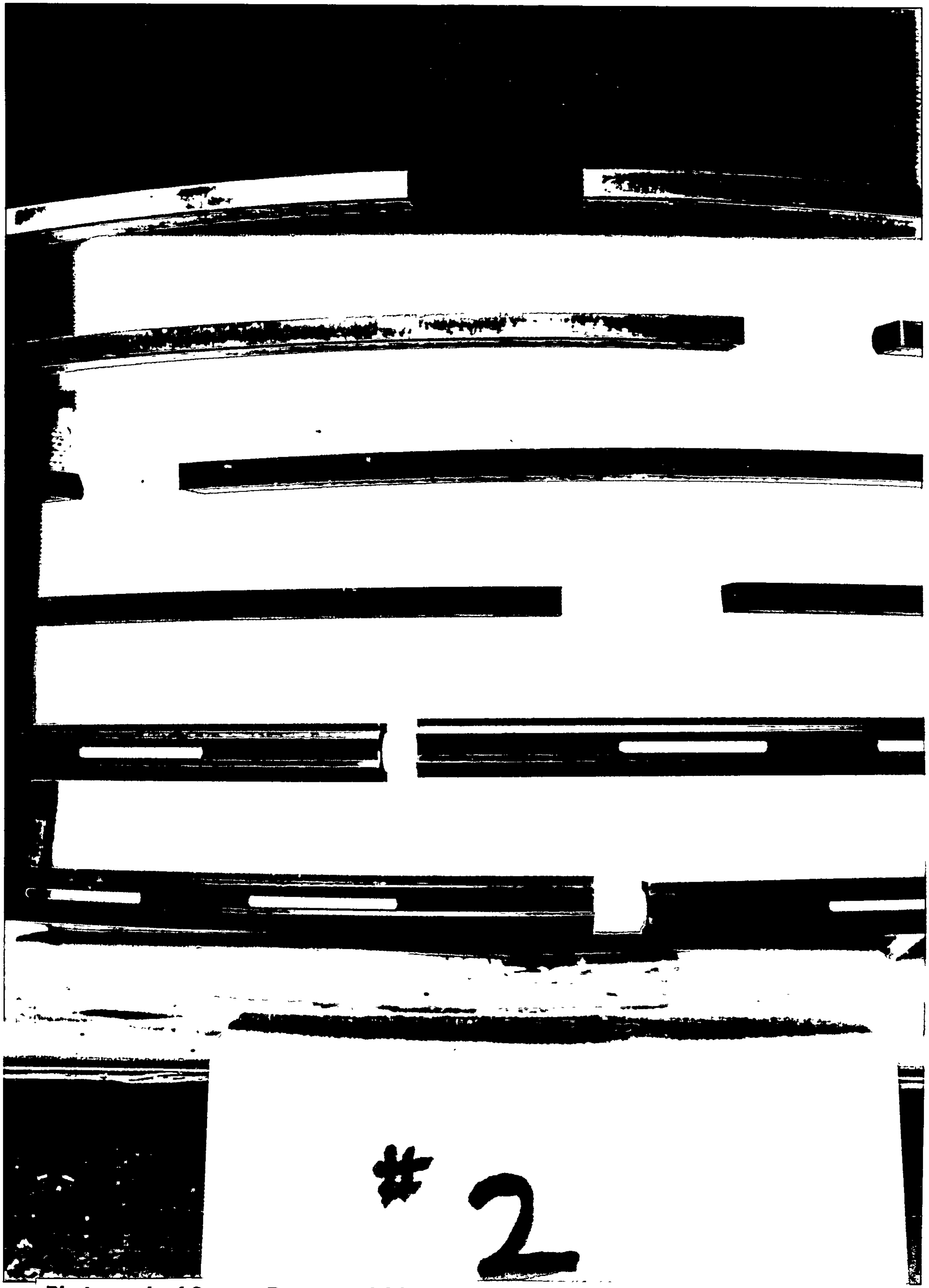


Photograph of Cooper-Bessemer LSC Engine Piston (Thrust Face) Showing Ring Grooves After 100-Hour Endurance Test

C-4



Photograph of Cooper-Bessemer LSC Engine Piston Bowl After 100-Hour Endurance Test



Photograph of Cooper-Bessemer LSC Engine Ring Pack After 100-Hour Endurance Test

C-6

Photograph of Radial-Flow Turbocharger Nozzle Blades After 100-Hour Endurance Test

C-7

Photograph of Radial-Flow Turbocharger Rotor Blades After 100-Hour Endurance Test