P P & L

ANALYSIS OF "B" & "C" 1989 DIESELS FAILURE

SEA-GW-037

PP&L

ENGINEERING STUDIES, ANALYSES, AND EVALUATIONS COVERSHEET

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ROOT CAUSE FAILURE ANALYSIS OF "B" AND "C" DIESELS: 1989

I. ABSTRACT

In September and October of 1989, SSES "B" and "C" diesels suffered crankcase overpressure events that resulted from excessive heating of pistons from friction. This report provides a comprehensive discussion of the findings related to these failures. It reviews previous failures at SSES and in the nuclear industry, describes various inspection findings related to visible damage and wear from normal operating conditions and collates all the pertinent information into a comprehensive analysis and an outline of future actions to prevent such events from occurring.

We have done sufficient reviews to identify the precursors to failure and have identified appropriate corrective actions to prevent recurrence.

While no single root cause could be determined from the information collected, the possible contributing factors are:

(1) The quality of the oil used for lubrication.

(2) The surveillance procedures required by the Technical Specifications which may strain the engines to their limit.

(3) The piston pin/bushing and their interface.

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ROOT CAUSE FAILURE ANALYSIS OF "B" AND "C" DIESELS: 1989

II. EXECUTIVE SUMMARY

As a result of the engine overpressurization events of September 16, 1989 and October 7, 1989, a Nuclear Department Diesel Generator Group was established. The objective of this group is to develop and implement a plan that assures all diesel generators continue to maintain a reliable status. The plan must address both the options of finding the root cause and not finding the root cause. The plan must also address the long and short term corrective actions to mitigate recurrence.

All available data was gathered on these machines in terms of analyzer runs, operating times, number of demands, total numbers and types of failures. Reviews of maintenance and operating practices for Susquehanna versus the manufacturer's recommendations were conducted. Reviews of Susquehanna maintenance practices versus the rest of the Cooper users is still to be completed. Other utility experience with Cooper KSV engines was reviewed. Information sources such as NPRDS and the Cooper Users Group was searched to find whatever information was available. Extensive inspections, reviews, and comparisons of failed versus "good" components were conducted. Many detailed metallurgical analyses, including elemental analyses, were conducted on failed and non-failed components. Extensive oil analysis, including foaming tests, were conducted. Leading experts in the field, as well as the manufacturer, were contacted and requested to provide analyses.

PP&L has already taken definitive actions (see Section VIII) to insure reliable operation which, as a minimum, consist of:

(1) Based on the metallurgical findings, Diesel Generators "A," "B," "C," and "D" were inspected. New pistons and liners were installed if the existing components were conservatively judged to warrant replacement. This refurbishment of the diesels placed them into an acceptable condition.

(2) We continue to test the diesels in accordance with the Technical Specifications. During some of this testing, we administratively extend the run time to enable bearing "self-healing" properties to occur and provide added assurance of load carrying capabilities.

Several general conclusions drawn from the body of this document can be stated:

(1) The engines will fulfill their current design intent and are reliable.

(2) Current maintenance procedures conform to manufacturer's recommendations.

(3) There is no evidence of general fatigue or wear out of the engines. The only abnormal wear identified is the compression rings, pin-to-bushing interface and the tin transfer from the non-thrust side of the piston to the liner. These appear to be unique to Susquehanna.

(4) Current Operating Procedures and Technical Specifications that require/allow rapid loading of the engine after starting and that do not permit sufficient thermal equilibrium of the engine to be reached could be contributing to failures and run counter to good reliability principles.

(5) We have done sufficient reviews to identify the precursors and have identified appropriate corrective actions to prevent recurrence.

(6) Removal of the current piston pin end cap and lower oil ring will improve lubrication to the piston liner area and not introduce other failure mechanisms.

(7) A significant amount of distress between the piston pin/bushing area is evident in a fairly large sample. Determination of root cause is in progress.

(8) Gulf lube oil is currently being used by PP&L but not by other nuclear utilities with Cooper engines. However, this oil passes all required tests and meets Cooper requirements.

(9) The oil sample point is not in the optimum location.

In addition to the positive actions already taken by PP&L, the following actions have been identified to meet the objective of enhancing diesel generator reliability. Definitive and detailed schedules will be developed and are subject to availability of spare parts, critically skilled resources, and the impact of planned refueling outages.

For the "A" through "D" engines:

(1) Replace all piston rings with correct size/type.

(2) Inspect all cylinders for tinning.

(3) "Blue check" all piston-to-pin assemblies for proper contact. Check flatness and surface finish on pins.

(4) Inspect all liners for correct chrome surface porosity.

(5) Implement the following design changes:

- o Lube oil sampling point change.
- o Removal of the lower oil ring and end cap.

For the "A" through "E" engines:

(1) Replace the Gulf lube oil with another Cooper recommended oil.

(2) Submit Technical Specification changes.

(3) Revise Operating Procedures to minimize stress on the engines consistent with Technical Specification allowances.

(4) Implement a Reliability Monitoring Program that includes routine engine analysis, routine engine oil analysis, vibration analysis and trending.

(5) Schedule remaining analysis work, assessments by Cooper, perform a study to determine the feasibility of adding a remote emergency shutdown switch, and examinations of distressed piston pins.

III. INTRODUCTION

1.0 Engine Description

1.1 Engine

1.1.1 The emergency diesel generators at Susquehanna are four V-16 cylinder and one V-20 cylinder Cooper Bessemer KSV engines. They are 4-cycle, turbo-charged engines. The 16-cylinder engines have a continuous rating of 4,000 KW and the 20-cylinder engine is rated at 5,000 KW continuous. Since the 20-cylinder is basically the same design as the 16-cylinder engine except for the additional four cylinders, turbocharger size and size of the cooling system, the following discussions apply to both engines.

1.1.2 The 16-cylinder engines have eight cranks on the crankshaft and the 20-cylinder engines have ten cranks on the crankshaft. Hence, opposite cylinders (right/left) share a common crank. This is accomplished with a main connecting rod for the right bank rod and an articulated connecting rod for the left bank. Refer to Attachment 1 for a sketch of this design. The main rod connects directly to the crankshaft while the articulated rod uses a pin connection to the main rod. The articulated rod bolts to this pin and the pin pivots in a bearing integral to the main rod. Each rod is bolted to the piston pin. The piston pin then pivots in a bushing inserted into the piston. End caps are placed over the holes for the piston pin bushing in the piston. These caps, held in place by an interference fit, serve to keep the oil lubricating the piston pin confined to the inside of the piston and prevents blow-by gases from forcing oil out of the pin/bushing interface. Otherwise, this oil would fill the annular space between the piston and liner in the area from the lower oil ring to the middle oil ring. (See Section 1.2, Piston, for placement of the oil and compression rings.)

1.1.3 Torque (power) is produced when the rapid burning of fuel above the piston generates a high pressure and forces the piston down as it passes top dead center (TDC). Once the piston is past TDC the connecting rod is at an angle relative to the downward movement of piston. This angle rotates the crankshaft and also pushes obliquely against the piston. This obliqueness causes the piston to thrust against the liner. The side of the piston which reacts to this lateral load during the power stroke is called the thrust side and the side opposite it is the non-thrust side. Note: These sides are perpendicular to the axis of the piston pin. During the other portions of the cycle, the non-thrust side is also loaded. However, the magnitude of the force is less than the power-stroke side thrust.

1.1.4 The next section discusses the lubrication path relevant to the crankshaft, connecting rods, piston pins and pistons/liners only.

The bottom of the engine serves as the lube oil reservoir for the engine. It holds approximately 1,000 gallons of 40-weight lube oil. The main oil pump is driven directly from the crankshaft and takes suction from a perforated pipe running the length of the engine. From the main oil pump, the oil is filtered and cooled and then returns to the engine. For the components of interest, the oil is supplied to them by a header in the oil sump which has hose connections to each journal bearing. Some of the oil supplied to each journal bearing lubricates it and the remainder of the oil flows through drilled passages in the crankshaft to the crank bearings. Here again, some of the oil lubricates and the remainder goes through a drilled passage in the main connecting rod splitting into the articulated rod and the main rod. The oil flows through these rods and supplies lubrication to the piston pin with the majority of it going to cool the crown of the piston.

The piston/liner interface is lubricated by oil splash. The splash is provided by the oil escaping from the crankshaft journal bearing, crank bearing articulated rod pin bearing, and the oil draining from the interior head of the piston.

1.2 Piston

The pistons are made of cast iron. They have a 2-3 mil thick tin plating on their exterior from the middle oil ring on down to the bottom of the piston. The tin plating provides a relatively soft surface which allows the piston surface to conform to the liner and allows it to absorb some small dirt particles which may otherwise cause abrasion. The top of the piston (crown) is dished so that the sides are higher than the center. This construction provides a heat dam in an effort to contain the combustion heat to the piston, thus providing a heat shield for the liner.

The top section of the piston from the middle oil ring up is of a lesser diameter than the remainder of the piston. This relief is provided to allow for thermal expansion in this section since it will be hotter than the lower section. See Attachment 2.

1.3 Rings

The piston has two types of rings:

(1) Compression rings to provide sealing of the combustion gas so that they do not "blow-by" the top portion piston and overheat the lower portions of piston/liner.

(2) Oil rings to control the amount of oil on the liner (cylinder wall). The oil rings may also be called oil control rings or scraper rings.

There are four compression rings. The two upper rings are made of a nodular iron and their radial width is less than that of the two lower compression rings. The two lower compression rings are made of a flake cast iron. All four compression rings have a one to two degree taper on their face (see Attachment 3) as to form a knife edge on the bottom.

There are three oil rings below the compression rings, two are above the piston pin and one on the lower piston skirt below the piston pin. These oil rings control the amount of oil on the cylinder wall so that the engine does not consume too much lube oil. There are holes drilled in through the piston wall behind the bottom two oil rings allowing the excess oil to drain back to the sump.

1.4 Cylinder Liner

The cylinders in our engines are replaceable and called cylinder liners. These liners fit into the block and are cooled on their exterior by jacket water. The liners are surfaced on the ID with a thick (approximately 6 mils) porous chrome plating to provide a hard wear surface and to retain oil for lubrication. The chrome surface is made porous by an electropolish-etching or roughened by a diamond lapping for oil retention purposes. All piston liners manufactured after 1987 are manufactured using the lapping process.

There are about 12 to 17 mils diametrical clearance between the piston and liner.

2.0 Engine Failure History at SSES

The SSES engines have had a total of six crankcase explosions, two on "B," two on "C" and two on "D."

The crankcase explosion on the "C" engine occurred in Cooper's shop during the factory testing of the engine in 1977. The cause of this explosion was identified by the Bechtel shop inspector to be overheating of a piston pin on the 4R cylinder.

The next crankcase explosion occurred on November 29, 1981 on the "D" engine. SSES was in the start-up testing mode and an improper oil flush caused the main lube oil pump bearing to be oil starved. The bearing overheated and caused the crankcase explosion.

The "D" engine had another explosion on January 14, 1984. The crankcase explosion was initiated by heavy scoring on the 2L piston and liner. When the failed cylinder was disassembled, a broken fuel oil drain hose

was discovered. This hose drains the fuel oil from the injection nozzle to the base of the fuel injection pump, thus allowing the fuel oil to drain into the crankcase. This failure was described as being caused by fuel oil dilution of the lube oil. PP&L is now considering this failure to be similar to the "B" and "C" engine failures.

On January 18, 1986, a loose piston pin bolt on the "B" engine, 5L piston, allowed lube oil to spill from the piston pin to connecting rod joint, starving the pin and piston of oil. This explosion was caused by an overheated pin.

The next crankcase explosions occurred on September 16, 1989 and October 17, 1989 on the "B" and "C" engines respectively, and these explosions are the subject of this report.

3.0 Other Utility Experience With KSV C-B Engines

All the other (eight without SSES) Cooper users were contacted concerning their experiences with crankcase explosions. Attachment 4 summarizes the design and operating information obtained.

Five stations have reported crankcase explosions or crankcase overpressurizations. A summary follows:

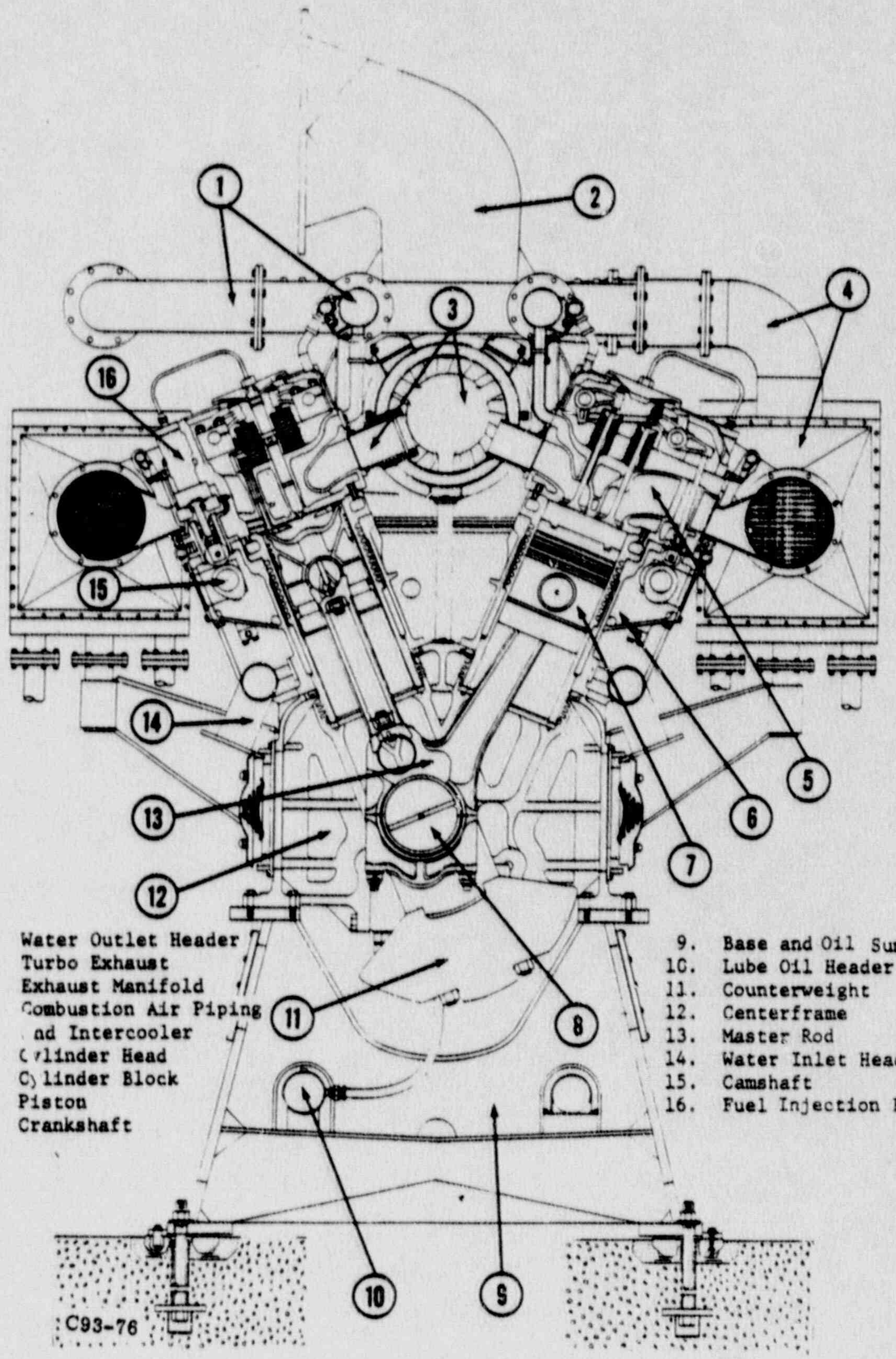
Palo Verde - The cause of their explosion was determined to be insufficient contact area between the piston bushing and piston pin. This contact area is to be checked (by a "blue check") during assembly per the Cooper IOM; however, this utility missed this step.

Braidwood - The crankcase explosion at this station was due to metal cuttings and chips in the piston pin oil groove. They feel the debris was left in the engine from the factory. This explosion occurred during their preoperational tests.

Byron - Byron experienced a crankcase explosion, the cause of which was not conclusively determined. The two possible causes are high firing pressure and dirt impeding lube oil flow.

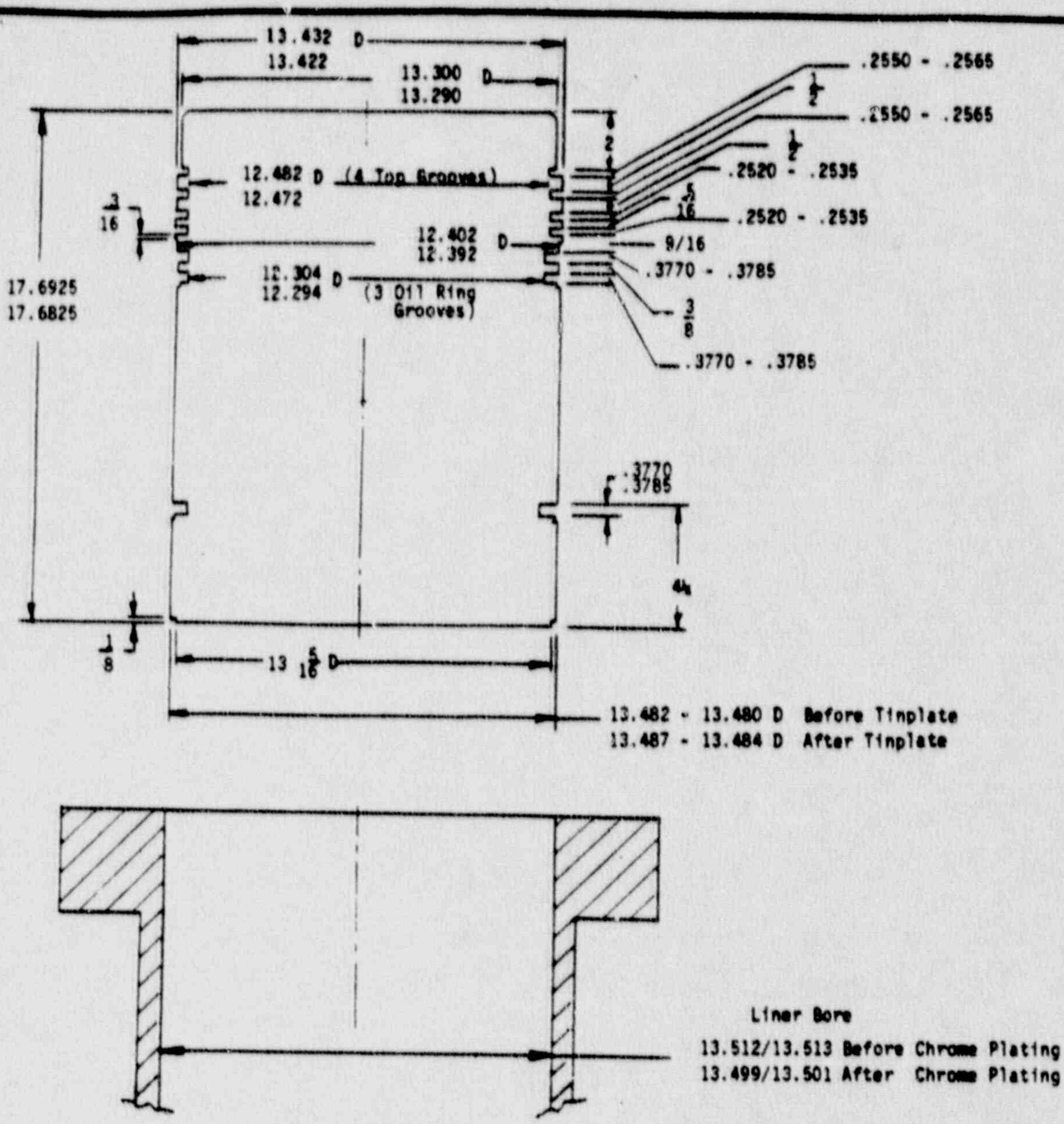
Zion - The Zion station also reported a crankcase explosion due to a piston bushing failure.

Cooper Nuclear Station - This station had three "crankcase overpressurizations" which blew open the relief doors on the crankcase. They are thought to be caused by water leaking into the lube oil, rapidly expanding and causing an overpressurization in the crankcase.



1. Water Outlet Header
2. Turbo Exhaust
3. Exhaust Manifold
4. Combustion Air Piping and Intercooler
5. Cylinder Head
6. Cylinder Block
7. Piston
8. Crankshaft
9. Base and Oil Sump
10. Lube Oil Header
11. Counterweight
12. Centerframe
13. Master Rod
14. Water Inlet Header
15. Camshaft
16. Fuel Injection Pump

KSV Diesel Engine for Nuclear Power Plant Service



Ref Power Piston KSV-5-A
Liner-Power 618-9-12A#2

KSV PISTON/LINER - DETAIL

DRWN. BY	BMcL
CHKD. BY	
APPR. BY	
SCALE	
SHEET 1 OF 1 SHEETS	

COOPER ENERGY SERVICES

A COOPER-BESSEMER PRODUCT

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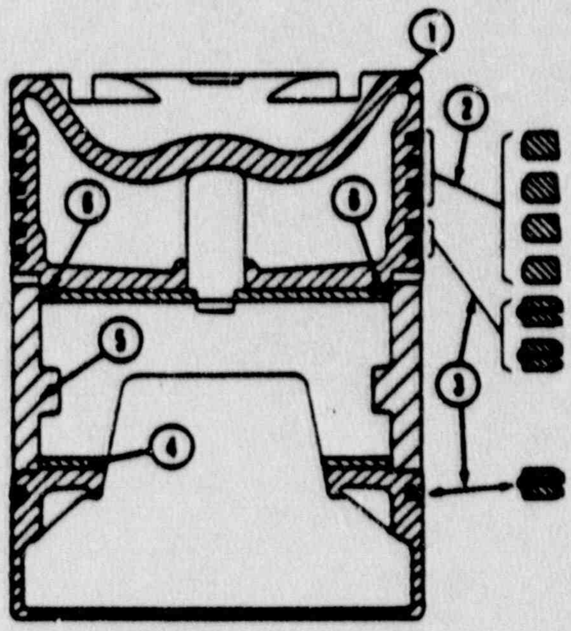
Chapter III
Attachment 3



1. Piston
2. Compression Rings
3. Oil Rings
4. Pin Bushing
5. Pin Cap
6. Screw Dowel

Piston and Rings

DATA FROM OTHER C-B USERS
UTILITY/STATION

Parameter	N P PD Cooper Nuclear	Commonwealth Edison Zion Station	Commonwealth Edison Byron Station	Commonwealth Edison Braidwood	ANPP Palo Verde	HP&L South Texas Project	LP&L Waterford 3	Niagara Mohawk Nine Mile Point
FULL LOAD KW	3600	4000	5500	5500	5400	5500	4400	4400
OVER LOAD KW	4400 (5000 for 1 hr)	4400	6050	6050	Not obtained	6000	4840	4840
# OF CYLINDERS	16	16	20	20	20	20	16	16
LUBE OIL USED	Mobil Guard 412	Mobil Guard 450	Mobil Delvac 1340	Mobil Delvac 1340	Shell Rotella T40	Chevron Delo 400	Mobil Devac 1340	Mobil Devac 1340
MONTHLY TESTS	Start, run 5-15 min., syn. load in 1000 KW increments every 10-15 minutes to full load	Same as Braidwood except loads are different	Start, run 5 min. syn. load to full load in 5 minutes	Start, syn. load in 1000 KW every 3-5 min. at full load in 15 minutes	Start, syn. 2-5 min. to full load	Start, syn. load to full load in 10 minutes	Start, syn. load to full load in 5 minutes	Start, syn. load in 500 KW increments (about 1 minute apart)
6 MONTH TEST	--		Start, run 5 min. syn. and full load in 60 seconds		Start, syn. 60 sec. to full load		Start, syn. load to full load in 176 sec. no warm up	
OUTAGE TEST	Sequence loads on then do 24 hour run - these are not T.S. requirements	Same as Braidwood except loads are different	24 hour run test without any specific warm up period, then they do sequence tests (LOCA/LOOP tests)	Slowly load to 110% (30 minutes) and do 24 hour test, after this they do their sequence test (LOCA/LOOP test)		Sequence test is done then overload test. Engine is load from zero to overload in 10 min.	Sequence test is done first which lasts 30-45 min. and they they do 24 hour run	24 run test loaded same as monthly
QUARTLY					Start, run 15-20 min. then load to 110% load			

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IV. METALLURGICAL ANALYSIS OF THE "B" DIESEL 7L CYLINDER FAILURE

1.0 Observations of the Failed Parts

1.1 Visual Condition After the Failure

1.1.1 The piston was heavily oxidized on the surface with evidence of the tin plating being melted in various areas. The blackened areas were highly scuffed with deep scratches and much smeared metal. The skirt of the piston below the pin covers appeared to be the location of the highest temperature seen by the piston and could be described as the most heavily damaged of any area seen. Some cracking was evident in the skirt area that went thru the wall. The top of the piston was covered with a crusty deposit that was very adherent but could be flaked off with a knife. The deposit was initially wet with oil and appeared black in color. The compression and oil rings were heavily worn and the lower two oil rings were so heavily caked with debris that they could not be released easily from the grooves. End caps cover the piston pin holes and these were heavily scuffed and blackened by heat.

1.1.2 When the end caps were removed, the inside surface of both caps were locally blackened by heat which oxidized the surface and decomposed the oil present there. One cap was easily removed, but the second cap required a force of two tons to remove it. After the cap came loose, the piston pin was pushed out with a relatively low force of about 200-300 lbs. Normally, the pins should be able to be removed by hand or at most with light tapping with a rubber mallet. Dark areas near both ends of the pin bearing surface indicated local heating. The bushing was blackened by oxidized oil and heat near both ends with the heaviest blackening in the region 90° from top center on both sides.

1.1.3 The cylinder liner showed a large amount of scuffing and wear in many areas where the piston rubbed against the surface. The worst areas appeared to be where the pin end caps would contact the surface of the liner. These areas showed long longitudinal scrapes and much deformed metal. Blackened surface areas occurred in various locations around the liner along with smeared metal in a tiger-striped pattern.

2.0 Metallurgical Investigation Findings on Specific Parts

2.1 Piston

2.1.1 Figure B-1 shows the lower section of the piston skirt after it was cut from the rest of the piston. One can see that there is much scraping and scoring at the level of the lowest

oil ring. A cross-section of a selected area from this region (Figure B-2) shows that the surface is coated with some very heavily deformed metal. Higher magnification pictures in Figure B-2 shows that just below the layer of deformed metal the cast iron has a martensitic microstructure. Since the normal structure is perlitic surrounding graphite flakes, it is evident that temperatures in excess of 1350°F had to be generated locally. Figure B-3 shows a high magnification micrograph of the surface layer revealing that it is layered with different phases.

2.1.2 Scrapings of deposits were taken from the surface of the piston to determine what elements they contained. Figures B-4 and B-5 give the results of the analysis taken on the Energy Dispersive Spectrograph (EDS). The particle shown in Figure B-4 was non-magnetic and contained P, Zn, S, Ca, and a small amount of Fe. The other particle in Figure B-4 contained the elements Zn, Ba, P, S, Fe, and Ca. All of these elements are in the additives in the oil except iron, but the iron could have been included from oxidation of the piston head material.

2.1.3 Figures B-6 and B-7 show another area of the piston skirt where a crack was found along with the layering phenomena. Figure B-6B is an enlarged portion of the lower left portion of Figure B-6A. Area "Q" was analyzed by the EDS technique to determine the elemental components here. Figure B-8 shows the spectrum obtained, which include high peaks for both iron and chromium and a smaller peak for silicon. Figure B-7 is an enlarged area of the surface of Figure B-6B and area "P" was analyzed as shown in Figure B-8 to contain iron and chromium with a small amount of tin mixed in. Finally both light grey and dark grey areas of Figure B-7 layers were analyzed and were found to contain mostly chromium with only a small amount of iron mixed in. These curves are shown in Figure B-9. Since the only source of chromium is from the cylinder liner plating, some transfer of this element had to occur to the piston skirt from the liner during the failure process.

2.1.4 A section of the piston head material was removed in the thinnest section near a valve depression to determine if excessive overheating had occurred. Figure B-10a shows the microstructure in a thick section of the piston head, and Figure B-10b shows the structure in the thinnest section of the same piston head. Both microstructures are perlitic and do not show any overheating. It would take temperatures above 1000°F before any significant changes would be detectable visually.

2.2 Piston Rings

2.2.1 Oil Ring

2.2.1.1 The uppermost oil ring was removed inspected and photographed. It was the only one of the three oil rings that was loose and easily removed without force. Figure B-11 shows some metallic debris that was caught between the wiper surfaces. Analysis of this debris showed that it consisted primarily of iron, tin and chromium elements with some minor elements coming from the oil additives.

2.2.1.2 A cross-section of the lower oil ring was made to show the profile of the wiper surfaces and this is displayed in Figures B-12 and B-13. The surface that is in contact with the liner was deformed from the high friction developed when the lubrication became inadequate. Figure B-13 shows a 50 times magnified image of the cross-section which reveals the deformed tip more clearly. From this photo it was possible to measure the length of the scraper finger which according to the drawing should be between 0.065" and 0.075" from the support. In this case, we determined that the finger was only 0.0575" long so the amount of wear is 0.0075" minimum.

2.2.1.3 Debris from the coil spring behind the lower oil ring was sampled and analyzed for elemental composition. The elements Fe, Cr, P, S, Sn, Ca, Mn and Si were found in the loose debris and also on the surface of the metal itself as shown in Figure B-14. The elements P, S, and Ca are commonly found in the nonmetallic debris in these engines from oil decomposition.

2.2.2 Compression Rings

2.2.2.1 The fourth compression ring from the top of the piston was examined to show the condition of the surface in contact with the liner. Figure B-15 shows that there was some unknown material embedded in the surface identified as spot "A." It was found to be mostly chromium with some Fe, Sn and Si mixed in. Spot "B" in the same figure was identified to be the base material which consists mostly of iron. Figure B-16 shows the EDS analysis curves of both of these spots.

2.2.2.2 Figure B-17 shows another unusual spot on the surface of the fourth compression ring. This spot was identified by the EDS technique to be an inclusion consisting mostly of copper along with tin, lead, iron and nickel. The source of this unusual combination of materials is a mystery. One will observe that this defect is associated with surface cracking so it could have been rubbed into the surface as a particle of debris. It is possible, however, for this defect to have been an inclusion in the casting at the time it was manufactured. A source of copper from any of the components in the cylinder area has not been identified at this time. Tin and lead are elements which could be on the piston either as a part of the plating or from repair of a casting defect in the foundry.

2.3. End Caps on Pin Openings

2.3.1 Piston 7L End Caps

2.3.1.1 Figure B-18 shows both sides of both end caps taken from the 7L piston. Both caps show heavy wear and scuffing on the outside surface that would contact the cylinder liner. The scuffing was so heavy that oxidation of the surface and deposition of deposits occurred. On the inside surface, oxidation was most noticeable on the "A" cap indicating that it may have been the first to experience overheating. The deposits on both of these caps was analyzed on the EDS and the elemental compositions found are shown in Figures B-19 and B-20. Cap "A" deposits were mostly iron and chromium, with minor amounts of Si, P, Sn and Ca. The minor elements were from the oil and the Cr must have come from metal transferred from the cylinder liner. On cap "B" the deposit analysis showed only chromium and a small amount of iron.

2.3.2 Piston 7R End Caps

2.3.2.1 At the time the 7L piston failed, the 7R piston was removed from service and inspected. The only appreciable degradation noticed was on the pin end caps. These were removed and made available for further inspection. The piston was then refurbished with new rings, the liner inspected for degradation, and then reassembled. No pictures were taken nor were any parts kept for analysis except the end caps. Figure B-21 shows the inside and the outside

views of the end caps. The "A" cap was the most severely worn on the surface with evidence of heat being generated sufficiently to produce blackening by oil decomposition and oxidation of the metal. The heat did not appear to progress deep enough to cause any darkening on the inside surface like that found on the 7L piston caps. A close-up photograph of this end cap shows evidence that some metal loss occurred in the rubbed areas as shown in Figure B-22. No analysis of this surface was performed to determine if chromium transferred to it from the liner.

2.3.2.2 On the 7R-B cap, there was evidence of some burnishing of the surface caused by contact with the liner, but not enough heat was generated to cause any blackening nor was there any apparent metal transfer from the rubbing surface. Figure B-23 shows two close-up photographs of the burnished surface. It does not show even enough wear to remove the machining marks which are the heavier parallel lines. The very small vertical scratches are abrasion marks from where the liner rubbed against the surface.

2.4 Piston Pin

2.4.1 The piston pin was easily removed from the bushing once the end cap was released. It did not appear to have frozen into place against the bushing surface. We assume, therefore, that the interface still had some lubrication present. The pin did not have heavy heating marks, but there was some slight oxidation towards both ends just beyond the circumferential oil grooves. No hardness measurements have been taken yet on this pin to determine the extent of overheating experienced over the bearing surface.

2.5 Pin Bushing

2.5.1 The heat pattern generated on the surface of the pin bushing can be seen in Figure B-24. Both ends of the bushing were obviously heat tinted and blackened with decomposed oil products. No metallurgical sectioning of this bushing was undertaken.

2.6 Cylinder Liner

2.6.1 The 7L cylinder liner was heavily damaged by metal wear, transfer and heat generation in most areas. The liner was sectioned on the thrust and non-thrust sides as well as the side which contacted the most heavily damaged end cap. The

following sections describe the findings of this investigation.

2.6.1.1 The thrust side of the liner was labeled as "B-1" and the section obtained from this is shown in Figure B-25. One can describe the damage here as being "tiger striped" in appearance with relatively clean unscraped areas separated by heavily worn, overheated and deposited areas. Cross-sections of the liner were taken in an effort to determine the condition of the chromium layer and identify the elemental deposits in the pits and attached to the surface. Figure B-26 shows the approximate locations of the metallurgical samples taken from this piece of the liner.

2.6.1.2 Piece "A" was cut from a damaged area 22 7/8" from the bottom of the liner and the polished surface examined was perpendicular to piston travel (the transverse section to the cylinder axis). Figure B-27 shows that the chromium layer is cracked and some of the pores are filled with a metal. The metal in this case was not specifically identified.

2.6.1.3 A second piece labeled "B" was cut from the liner about 11.5" from the bottom with the viewed surface oriented perpendicular to piston travel (longitudinal to the liner axis). Figure B-28 shows typical areas where debris has attached itself to the chromium surface. It is very tenacious and does not exhibit typical metallurgical microstructures. It appears to be smeared metal or metal that has experienced a lot of deformation. We will later show analysis taken from some typical areas to show the elemental components present.

2.6.1.4 Section labeled "C" was also taken about 11.5" from the bottom of the liner, but the viewed surface was taken parallel to the piston travel. Figure B-29 shows that the chromium surface has a rough appearance and does appear to contain many small cracks. This is an unetched polished surface, so the fine lines in the chrome layer are actually fine cracks and not etched grain boundaries. The formation of such cracks could take place with friction from the rings as they traversed over this area.

2.6.1.5 Sections D and E were taken from the lower end of the liner 7.5" from the bottom and the viewed surface was perpendicular to piston travel in both

cases. The pieces were adjacent to each other. Typical wear areas are shown in Figures B-30 and B-31. These areas were heavily scraped by something that made valleys in the chromium and in one case cracked the base cast iron core. The amount of wear was so great that the original pores in the chromium are not longer evident and in the case of Section E, the groove in the chromium plate was filled with another metal.

2.6.1.6 Section B-2 was taken from the side of the piston where the pin cover would have scraped the surface. Figure B-32 shows what the scraped surface looked like before it was further sectioned. Figure B-33 shows the locations of the pieces taken from this liner piece for metallurgical examination.

2.6.1.7 Sections A, B, and C were taken from an area 5 1/8" to 6 1/2" from the bottom of liner part B-2. Sections A and B were polished to show the surface parallel to piston travel while Section C was taken perpendicular to piston travel. Figure B-34 shows and area of Section A which had a defect in the chromium plate (possibly damage formed during the failure) which traverses the plate at an angle and goes all the way to the base metal. The light phase in the crack was determined by EDS to be mainly tin. This tin most likely worked its way into the defect by capillary action when the heat generated during the failure process melted it from the piston surface and it got spread over the liner surface by the rings.

2.6.1.8 Section C was photographed in the Scanning Electron Microscope (SEM) and pictures of the condition of the chromium plate are shown in Figure B-35. EDS analysis of the layer of deposits found are shown in Figures B-36 and B-37. The major elements found at each level analyzed are shown in the following table:

Area	Elements (Greatest to Least)
"R" Base of pit	Cr, Fe, and Sn
"Q" Top of pit	Fe, Sn, and Cr
"P" Center layer	Fe, Cr, and Sn (Trace)
"P" Topmost layer	Fe, Cr, and Si

2.6.1.9 Section F was taken 24.625" from the bottom of liner part B-2 and sectioned to produce a surface parallel to the piston travel direction. Figure B-38 shows

one area which did not show deposits on the surface of the chromium, but did show several cracks. Another area of Section F, shown in Figure B-39, is an enlarged portion of the surface of the chromium coating which is coated with a layer of metal. An EDS analysis of the lighter section of this coating shows it to contain both chromium and iron in approximately equal proportions. Both iron and chromium were found in the rest of the coating also.

2.6.1.10 Section H taken from liner part B-2 was oriented such that the polished surface inspected was perpendicular the piston travel. The surface of the chromium plating was also coated with a layer of foreign metal and this layer is shown as a dark layer in Figure B-40. An enlarged section of the layer is also shown and appears to be composed of several layers. The material in the pore, labeled "U" was found to be high in chromium and a lesser amount of iron. The outermost layer "W" was found to be highest in iron with a lesser amount of chromium. Location "Z" contains just iron since this is the liner casting material. Location "Y" was identified as containing just chromium and location "X" was identified as being mainly chromium with a small amount of iron mixed with it. The EDS analysis results are displayed in Figures B-41 and B-42. The following table gives the elemental contents of the various layers in each location.

Layer	Elements in Descending Order
Z - Base metal casting	Fe
Y - Plating layer	Cr
U - Content of pore	Cr, Fe, Sn (Trace)
X - Thick dark layer	Cr, Fe, Si
W - Top layer	Fe, Cr, Sn (Trace)

2.6.1.12 Section B-3 is the non-thrust side of the liner seen in Figure B-43, and it was sectioned in various areas as shown in Figure B-45 to determine if the metallic layers and wear patterns had any unique characteristics which could tell us the sequence of failure taking place in this liner. Section A was taken from the lower end of the liner between 7 5/8" and 8 3/8" from the bottom and oriented such that the observed surface was along the direction of piston travel. The various metallic layers were found on this section attached to the chromium plating and these are identified in Figure B-46. The following table summarizes the findings of the EDS analysis from Figures B-47 and B-48.

Layer	Elements in Descending Order
A - First layer on Cr	Fe, Cr, Mn, Si, S
B - Light layer	Fe
C - Dark layer	Fe, Cr, Mn
C - Outermost dark layer	Fe, S, P (Trace)

Figure B-49 shows and enlargement of area "C" which was observed to have both a dark grey and a light grey component C1 and C2, respectively. The compositions of these phases as shown in Figure B-50 were different as shown next.

C1 = Fe, Mn, S, Si and Cr (Trace)
C2 = Fe, Mn (Trace) and Cr (Trace)

2.6.1.13 In the lower right portion of Figure B-46 is a chromium pore which contains metal and this area was investigated further to show what metal was first laid down on the chromium surface. The elements found to be present in this pore were Fe, Cr (trace), Mn (trace) and S (trace).

2.6.1.14 One area of Section A contained a rather thick layer of added debris to the surface and this is shown in Figure B-51. The thickness of this deposit was measured to be 0.012" compared to the 0.004" of chromium in this area.

2.6.1.15 Section B was identified for investigation and it was located 16" up from the bottom of the liner. The section was cut and polished to show the surface perpendicular to the piston travel direction. Figure B-52 shows various damage areas found on this section. Much wear, gouging, pore filling and cracking are observed on this chromium surface.

2.6.1.16 Section C was taken at a distance 16 1/2" to 17 1/4" from the bottom of the liner and oriented such that the surface observed was in the direction of piston travel. Figure B-53 shows typical light microscope pictures of the surface condition. Figures B-54, B-55, B-56, B-57, and B-58 show various locations on the surface of this section as seen on the SEM and in some cases, the elements in the various areas are identified. In the cracks in the chromium plating we found particles of tin predominant. In the pores themselves, tin was occasionally present, but usually associated with a large particle of iron. Chromium was sometimes present as a single identifiable particle or was mixed up with iron and

other elements. In some cases, iron, chromium, tin, manganese, sulfur, and silicon were all found together.

3.0 Discussion

3.1 This cylinder showed much physical damage, heating and generation of debris. The lower two oil rings showed heavy wear and deformation on the sliding surfaces. Molten and resolidified metal was found and was so heavy that the rings were difficult to remove. Debris analyzed on the surfaces of the rings showed the presence of all of the possible elements found in the materials of construction of the various parts and elements that would only come from oil decomposition products.

3.2 The lower piston skirt was heavily scuffed all the way around indicating that the piston had overheated so much that it expanded and completely filled the gap normally open between the piston and the liner. With a clearance of about 0.014" between the OD of the piston and the ID of the liner, a thermal expansion coefficient of 7.6E-06/°F, an. a diameter of 13.5", a temperature difference of only 136°F is needed to close the gap. Since tin has been melted off the surface of the piston during the failure, and it melts at 350°F, it is not unreasonable to expect that a temperature difference of this magnitude existed at the time of failure.

3.3 Compression ring damage was in the form of wear, scraping and metal pickup. The top ring showed the most wear and the least wear was on the bottom ring as evidenced by the amount of ring surface contacting liner. A normal ring should only show an estimated 5-10 percent of its outer surface polished by contact with the liner. In this case, the upper ring showed 100 percent contact. Ring wear can be caused by a number of different mechanisms discussed below.

3.3.1 Ring wear can be initiated by debris falling down from the head of the piston just above the top ring. Heavy deposits of debris in this area above the top ring on the non-thrust side has been noticed on most of the pistons inspected to date. Vertical scrape markings on these deposits indicate that loose debris can be generated by rubbing against the liner whereupon it can and will fall onto the ring. Since there is a small gap between the upper side of the ring and the liner wall when the ring is new, it will enter this area and abrade or polish the wall until it is washed down by the oil.

3.3.2 Another means of initiating ring wear is to reduce the oil thickness or lubrication on the cylinder wall. Oil is smeared over the surface of the cylinder wall via a splash mechanism which deposits the oil on the lower half of the

liner. As the piston goes down and back up through its normal cycle, the lower oil ring spreads the oil film uniformly over the surface and removes the excess. On the up stroke, the remaining oil film is drawn upwards and spread farther up the liner surface. As the piston completes many cycles, the oil film is gradually spread over the entire length of the cylinder wall to the uppermost location of the top compression ring. The oil is retained on the liner wall by pores in the liner surface deliberately put there for that reason. If there is some reason that these pores no longer retain oil, then the upper wall of the liner will gradually become starved of oil, friction will increase, ring wear will increase and heat will be generated that has to be dissipated. Lubricating oil can be diminished by the following causes.

3.3.2.1 Filling the pores with wear debris from the rings.

3.3.2.2 Filling the pores with carbonized combustion products and oil decomposition compounds.

3.3.2.3 Filling the pores with tin transferred from the piston surface.

3.3.2.4 Scraping of the surface of the liner by some moving part. Two parts which should not normally remove oil are the pin end caps and the top of the piston between the first and second compression rings.

3.3.2.5 Excessive pressure from the rings being forced against the liner can come from various sources.

A. High firing pressures in the cylinder.

B. Crud buildup behind the ring which would prevent the ring from moving freely back into the ring slot when the piston is moving towards the liner.

C. Insufficient ring end gap.

D. Wrong ring radial width. We have investigated the possibility of some of the ring having the wrong radial dimension and did find some in other cylinders and engines that were slightly oversize by a few mils. This reduces the gap behind the ring for debris to build up in before it becomes a problem. However, we have no evidence that indicates the gap was never so small that the ring would be forced against the liner wall by cylinder movement (without debris buildup).

3.3.2.6 Fuel oil dilution of lube oil was considered as a cause of the failure, but an investigation of the spray nozzle showed that it was spraying efficiently and we concluded that lube oil dilution was unlikely.

3.3.3 If the wrong material was used to make the rings, rings may be expected to wear abnormally. We have examined many sets of rings from different pistons and new ones from the warehouse, and have not found any that had the wrong material or hardness including the ones from the 7L cylinder piston.

3.3.4 That the rings can undergo excessive wear independent of other degradation in the engine has recently been made evident by an inspection of pistons and liners from the "C" diesel during the week of December 10, 1989. Pistons 1R, 2R, and 8L were removed from this engine. The following visual inspection results were reported.

Cylinder ID	Liner Damage	Ring/Liner Contact	Evidence of Piston Wear	Pin	Bushing
1R	None	100%	None	OK	OK
2R	None	90%	None	OK	OK
8L	Slight tin smear non-thrust	100%	"Sn" burnish	Scratched	Scratched

It is evident from this data that ring wear does not have to be associated with any other obvious wear or damage process.

3.4 Pin End Cap Damage Discussion

3.4.1 Pin end cap scraping appears to be a significant source of damage and debris with this failure. Both of the end caps of the 7L piston did show a high degree of scuffing, scratching, metal transfer and overheating. Tin, chromium, and iron were found on the surface of the liner in the area of contact with the end cap indicating transfer of metal back and forth between the moving cap and the liner. The end cap also showed a layer of chromium indicating that metal transfer occurred from the liner to the cap. Once metal transfer starts, the rubbing surfaces become rough and this increases friction between these parts and the oil rings that move over the liner in the same area. To produce this type of metal transfer, either a lot of pressure was needed to overcome the hydraulic action of the oil film, or the film was not present and metal-to-metal contact occurred. Metal transfer takes place on an atomic level where a clean, oxide-free surface contacts a similar surface and local welding occurs. On relative motion, the weaker surface breaks and a small amount of metal is transferred to the stronger surface.

3.4.2 In the absence of sufficient pressure, the oil film must have been scraped away. The end cap does have sharp edges and we know that it has moved outwards from its assembled position to contact the liner. If the first contact was at a slight angle so that the edge could act as a scraper on either the upward or downward stroke of the piston, the oil would be scraped away and further contact would be between the metal parts of the piston and the liner without the benefit of the oil film.

3.4.3 Pressure can be applied to the end caps in the following manners:

3.4.3.1 The end caps are inserted into holes that are machined to be 0.0005" to 0.0025" smaller than the cap. The cap is tapped into place with a hammer until it seats against the bottom of the hole and the bushing. In its normal position, as designed and installed, the cap would reside at least 1/16" away from the liner. During operation of the engine, there should not be any movement of the end cap, but the cap is not locked into place in a positive manner so that vibration, thermal expansion forces and other forces can act to move the cap outwards. Once pushed out, it would be difficult to force it back in by just contact with the liner. In the "out" position, there would be a tendency for the cap to bow because the force fit would now only be acting on part of the circumference area of the cap. In addition, any added heating on the cap due to friction with the liner would expand the cap and lock it firmer into place. Cooling is minimal in this area since there is no cooling oil flow to take away the heat.

3.4.3.2 Movement of the cap outwards can also take place by actions taken during installation. Before the piston is inserted into the liner, the connecting rod is connected to the piston pin in the bushing and the end caps are inserted in their place. While handling this assembly, it is possible to move the pin back and forth far enough to touch and apply pressure to the inside surface of the pin end caps. Until the lower end of the rod is connected to the crank, nothing prevents the pin from pushing the end caps out.

3.4.3.3 Another method of pushing the end cap out against the liner during operation of the engine is to heat up and expand the bushing. At installation, there is very little gap between the end of the bushing

and the back of the end cap. Excessive heating has been observed in the bushing surface of this piston and temperatures up to the melting point can be rationalized. The bushing is made of a bronze material designated as C93700. One phase of this alloy melts at a temperature of 361°F (183°C). In the area of the piston pin that contacts the bushing, temperatures in excess of 1200°F have been estimated to exist as evidenced by softening of the hardened surface of the pin. The expansion coefficient for the bronze is 10.22E-6/°F (18.4E-6/°C) so that over the length of the bushing (12") and considering a normal temperature of 180°F, the added length of the bushing at 361°F would calculate to be 0.022". This is not enough expansion to push the end cap out to the liner, but could move it part of the way.

3.4.3.4 Expansion of the piston pin can be greater than its surroundings since we said above that local heating was estimated to be upwards of 1200°F and higher. The expansion coefficient for steel is about 8E-6/°F, so that if the pin reached a uniform temperature of 1200°F from 180°F, the total length change would be 0.098". Of course, we do not believe that the entire pin reached this temperature, but a value of half of this would not be out of the question, so that 0.049" would not be unreasonable.

3.4.3.5 Another source of pressure to move the end cap outwards could be pressure from oil building up behind the cap and not being relieved. There are two drain holes for oil leaking out of the ends of the piston pin which are small enough so one could imagine that they could become plugged with debris. No evidence has ever been collected from all our disassembly operations over the years that any debris was found blocking any oil drain hole in any piston from any of our engines.

3.5 The piston itself can contact the liner between the two upper compression rings if it tips slightly on its upward stroke. We have seen evidence of this action since the pistons have a moon-shaped wear pattern in this area on the non-thrust side of the piston. The only way this wear pattern can be produced is if the piston tips. The effect of this scraping is twofold. First, it removes any oil film that may be present and abrades the surface of the liner. Second, the abrasion can remove any oxide film that would prevent welding of the piston to the liner. The clean metal surface can now pick up any metal that it contacts including compression rings,

piston iron and piston tin coating. Both iron and tin have been found embedded in pores of the non-thrust side of the liner (Figure B-44) and the moon-shaped pattern with heavy scraping and metal removal has been observed on the piston. There is no doubt that this mechanism is operating to transfer metal back and forth between the moving parts of the piston and the liner.

3.6 Pin Lubrication and Overheating

3.6.1 We have noticed on this 7L piston that overheating has occurred on the pin and the bushing in which it moves. Lubricating oil is applied to the interface between these two parts by gravity and inertial forces. The oil is pressure fed up the connecting rod, through the pin and into the upper cavity of the piston above the pin. Oil that sprays up to the head falls down and fills a pool of oil that covers the exposed portion of the pin and bushing. A groove in the top of the pin allows oil to flow along the length of the pin and through circumferentially oriented grooves near the end of the pin. Rotation of the pin in the bushing is a back and forth motion which distributes the oil in the groove over the mating surfaces. How can the bushing be starved of oil in an area that is flooded with oil at all times? Some of the things that could affect the lubricity of the surfaces include the following:

A. Oil Viscosity - Oil properties have been always within specifications.

B. Oil Foaming Characteristics - This quality has not been a normal test for our engines. However, recent oil foaming tests, described in Section 1.4.1 of Chapter VI, did not reveal any abnormal foaming characteristics in oil taken from all of our engines.

C. Clearances Between the Pin and Bushing - Fit-up or "blue checking" of the parts has not been performed per manufacturers specifications until recently.

D. Distortion of the Pin or Bushing - Several piston pins have shown bending up to 0.005" after years of operation usually with the fact that overheating had occurred. However in one case, bending of 0.005" was noticed on a pin from the "D" engine which did not show any evidence of overheating. This is being investigated further.

E. Debris Causing Friction Between the Pin and Bushing Surfaces - Debris has been observed to be present on occasion as evidenced by scratching of the surface of the bushing. No relation to the overheating could be found however.

F. Debris Filling Up the Grooves in the Pin and Not Allowing Oil to Flow Down Them - We have never found debris or deposits of such a volume that they were clogging up any oil passages.

G. Debris Plugging Up the End Cap Drain Holes - Never observed.

H. Air Pockets Preventing Oil to Flow Down the Grooves in the Pin - Since the pin/bushing lubrication is only by gravity feed and not force fed by a pump, an air bubble could conceivably block free oil flow into the bearing area. Once starved of oil, local overheating could start in high friction areas at raised parts of the bushing. At the boiling point of the oil, gas volume would increase dramatically and prevent any further ingress of oil into the area. At this point, nothing would stop the process of severe overheating except shutting down the engine.

4.0 Conclusions

4.1 Heavy damage was found on the cylinder liner surface, piston, compressions rings, oil rings, pin end caps, piston pin and pin bushing of the "B" diesel 7L cylinder.

4.2 Metal was found to be transferred back and forth between the rings, piston and chromium liner during the failure process.

4.3 Wear products from the piston surface and rings were generated and deposited in the liner pores early in the failure process.

4.4 A subjective overview of the damage to the piston assembly after the failure showed the heaviest damage located in the area of the piston pin end covers and lower on the piston skirt. End cap cover damage leads us to the conclusion that they may have initiated the failure process. This is supported by the fact that the largest amount of damage was found on the surface of the end caps in the 7L piston and that the only damage found in the 7R piston was end cap damage of a similar nature but lesser extent. This leads us to believe that the 7R cylinder was undergoing the initial stages of failure by end cap rubbing on the liner.

4.5 The following sequence of events appears to be one of the possible failure scenarios that could logically describe the failure event:

A. For one of several reasons described in this report, the end caps move out and press against the wall of the liner generating wear products and heat.

B. The wear particles attach themselves to the wall of liner, are pressed into the chromium liner pores or find their way down to the oil rings where they can cause more friction and wear to occur on the oil rings.

C. The buildup of debris on the liner surface causes accelerated wear on the compression rings and oil rings as they pass over the damaged liner surface.

D. Wear products from all the parts get mixed up and continuously pass back and forth between the wearing surfaces as the failure process proceeds.

E. Eventually localized heating becomes so great in some areas that the cooling oil cannot take it away fast enough to prevent oil decomposition and oxidation of the heated parts.

F. The heat continues to build up in the piston so that it expands to fill the gap between the piston and the liner.

G. Once the gap is closed, frictional heat raises the temperature of the parts to red heat at which time oil vapor and air in the region are raised to their ignition point and a crankcase overpressurization occurs.

Chapter IV

5.0 Figures

5.1 Figure Captions

5.2 Figures B-1 to B-58

Figure B-1. Picture of the skirt of the "B" diesel 7L piston showing heavy wear and metal distortion in the lower ring area.

Figure B-2. Cross-section of the "B" diesel 7L piston skirt area showing deposited metal in the upper micrograph and martensite formation in the microstructure in the lower micrograph. The latter indicates temperatures in excess of 1350°F were experienced by the material locally.

Figure B-3. Enlarged section of the deposits on the skirt of the "D" diesel 7L cylinder showing the layering evident there.

Figure B-4. Micrograph and EDS analysis results of non-magnetic debris taken from the head of the "B" diesel 7L piston.

Figure B-5. Micrograph and EDS analysis results of non-magnetic debris taken from the head of the "B" diesel 7L piston on another particle from that analyzed in the Figure B-4.

Figure B-6. Cross-section of a cracked area of the "B" diesel 7L piston skirt showing metal deposits on the surface.

Figure B-7. Enlarged area of the bottom right portion of Figure B-6 showing where analysis "P" was taken.

Figure B-8. EDS analysis of areas "P" and "Q" in Figures B-7 and B-6.

Figure B-9. Spectra of the light and dark phases in Figure B-7.

Figure B-10. Microstructure of the head of the "B" diesel 7L piston showing a normal perlitic, graphite flake condition
a. Thick section b. Thinnest section

Figure B-11. Portions of the top wiper ring of the "B" diesel, 7L piston showing the pickup of metal and debris on the surfaces touching the liner.

Figure B-12. Cross-section of the lower oil ring from the "B" diesel 7L piston showing the wear and deformation on the two wiper surfaces.

Figure B-13. Picture of oil ring profile at high magnification.
Magnified image of oil ring finger from "B" diesel, 7L piston lower oil ring showing structure deformation and size of the finger.

Figure B-14. Debris EDS analysis from the spring behind the lower oil ring from the "B" diesel 7L piston.

Figure B-15. Smeared metallic deposits found on the #4 compression ring of the "B" diesel, 7L piston are shown here as the dark gray area labeled "A." EDS analysis of area "A" shows it to contain Cr, Fe, Si (trace) and Sn (trace). Area "B" shows only Fe with a trace of Cr.

Figure B-16. EDS analysis of the #4 compression ring from the "B" diesel 7L piston in areas "A" and "B" of Figure B-15.

Figure B-17. An unusual spot of foreign material found embedded in the #4 compression ring from the "B" diesel 7L piston. EDS analysis shows it to contain Cu, Fe, Sn, Pb, and Ni.

Figure B-18. Pictures of the end caps as found on the "B" diesel 7L piston showing the scraping and local heating experienced by these parts during the failure sequence.

Figure B-19. Picture and EDS analysis of a piece of smeared metal removed from end cap "A" from the "B" diesel 7L piston.

Figure B-20. Particle of smeared metal removed from pin end cap "B" from the "B" diesel 7L piston. The EDS analysis shows that it consists primarily of Cr with some Fe and a trace of Ca and Sn.

Figure B-21. Abrasion and wear found on the "B" diesel 7R piston pin end caps.

Figure B-22. Photograph of the "B" diesel 7R piston end cap which showed the worst amount of wear, heating and metal transfer.

Figure B-23. "B" diesel 7R end cap showing the least amount of wear, but does show that the cap was touching the surface of the liner as evidenced from the light scratch marks on the surface perpendicular to the machining marks.

Figure B-24. Photographs of both ends of the "B" diesel 7L piston bronze bushing showing the heat tinting and decomposed oil products near the ends 90° to the top of the bushing.

Figure B-25. Full-length view of the "B" diesel 7L liner surface on the thrust side of the liner (identified as piece B-1).

Figure B-26. Sketch of the thrust side of the "B" diesel 7L liner showing the location and orientation of the sections removed for metallurgical cross-sectioning (part labeled B-1).

Figure B-27. Transverse cross-sectional views of the thrust side of the "B" diesel 7L liner 22 7/8" from the bottom (piece "A"). The chromium plating is cracked and some of the pores are filled with metallic particles.

Figure B-28. Longitudinal view of the "B" diesel 7L liner on the thrust side 11 1/2" from the bottom showing metallic deposits, filled pores and cracking in the Cr plating (piece labeled "B").

Figure B-29. Longitudinal cross-section (labeled "C") taken 12" from the bottom of the "B" diesel 7L cylinder liner showing a rough fractured surface with pieces missing and some debris in the pores of the chromium plating layer.

Figure B-30. Transverse cross-section (labeled area "A") taken 7 1/2" from the bottom of the "B" diesel 7L liner showing depressions in the Cr plating surface, cracking of the base metal casting and deposited metal.

Figure B-31. Transverse cross-section (labeled area "E") of the thrust side of the "B" diesel 7L liner taken 7 1/2" from the bottom showing depressions in the Cr plating and deposited metal.

Figure B-32. Views of the length of the "B" diesel 7L cylinder liner in the area of pin end cap scraping (part B-2) showing heavy longitudinal marks and overheating over 20.5" of the center section.

Figure B-33. Sketch of the end cap side of the "B" diesel 7L liner showing the locations of the metallurgical samples taken for cross-sectional examinations (part labeled B-2).

Figure B-34. Longitudinal section of "B" diesel 7L liner (labeled "A" of part B-2) showing cracking in the Cr plating surface and the presence of Sn at the Cr-Fe base metal interface.

Figure B-35. Transverse cross-section of the "B" diesel 7L liner (labeled "C" of part B-2) showing areas of metal buildup and filling of pores with Cr, Sn and Fe.

Figure B-36. EDS analysis of areas "Q" and "R" shown in Figure B-35.

Figure B-37. EDS analysis of area "P" and the dark layer above "P" in Figure B-35.

Figure B-38. Longitudinal section "F" from "B" diesel liner 7L piece B-2 taken 24 5/8" above the bottom. This section does not show any pores filled or otherwise, but does show several cracks through the thickness of the Cr plate.

Figure B-39. Longitudinal section of "B" diesel 7L liner taken from the end cap side 24.625" from the bottom showing the layer of deposited metal and the EDS analysis of the lighter grey area.

Figure B-40. End cap side of the "B" diesel 7L liner taken from transverse Section H showing damage to the chromium liner and deposited metal. EDS analysis of the various layers are shown in Figures B-41 and B-42.

Figure B-41. EDS analysis of areas "U" and "W" shown in Figure B-40.

Figure B-42. EDS analysis of areas "X" and "Y" shown in Figure B-41.

Figure B-43. General surface condition of the non-thrust side of the "B" diesel, 7L liner.

Figure B-44. Magnified image of the surface condition of the non-thrust side of the "B" diesel 7L liner near the upper position of the top compression ring. Both iron and tin are seen embedded in the chromium liner pores.

Figure B-45. Sketch of the non-thrust side of the "B" diesel 7L liner (designated B-3), showing the location and orientation of the metallurgical Sections A, B, C, D, E, and F. Section E was for surface examinations, while the others were for cross-sections.

Figure B-46. Longitudinal section of the "B" diesel 7L liner taken from an area 7 5/8 to 8 3/8" from the bottom. EDS analysis results are shown in Figures B-47 and B-48.

Figure B-47. EDS analysis of areas "A" and "B" in Figure B-46.

Figure B-48. EDS analysis of areas "C" and "D" in Figure B-46.

Figure B-49. Enlargement of area "C" in Figure B-46 showing light and dark phases present. EDS analysis of these areas are shown in Figure B-50.

Figure B-50. Analysis of the pore from Figure B-46 center right, showing that it contains mainly iron with only a trace of the elements Cr, S and Mn.

Figure B-51. Longitudinal cross-section of the "B" diesel 7L liner taken 7 5/8" from the bottom showing a very thick layer of metallic deposits covering the chromium plating.

Figure B-52. Transverse cross-section of a typical area of the non-thrust side of "B" diesel 7L liner 16" from the bottom (B-3 area "B") showing the filled pores and the deposited metal from this location.

Figure B-53. Longitudinal cross-section of the non-thrust side of the "B" diesel 7L liner taken 16.5" from the bottom of the liner. Typical examples of the metal deposited from the piston and rings on the Cr plating.

Figure B-54. SEM micrographs of longitudinal cross-section "C" part B-3 of the "B" diesel liner at a level 16.5" from the bottom. Elements shown were identified as being present in the pores indicated.

Figure B-55. Another area of pore deposits and elements found present in longitudinal cross-section "C" part B-3 of the "B" diesel liner 16.5" from the bottom.

Figure B-56. Various elements and their location found in a pore from the "B" diesel liner 7L, part B-3, area "C."

Figure B-57. Elemental contents of two pores filled with metal in the "B" diesel 7L liner 16.5" from the bottom of part B-3, area "C."

Figure B-58. Additional areas of filled pores found in area "C" of part B-3 of the "B" diesel 7L cylinder.

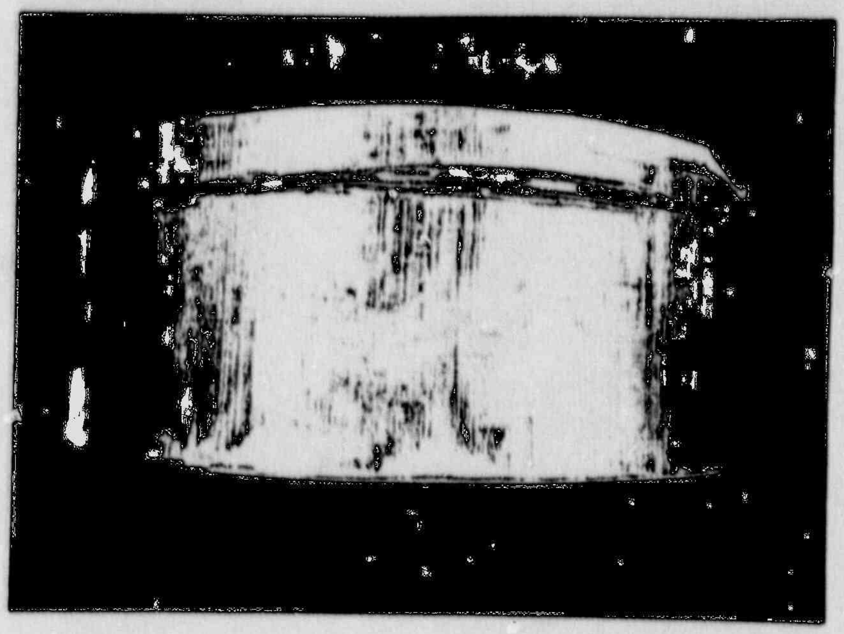


Figure B-1 Picture of the skirt of the B diesel 7L piston showing heavy wear and metal distortion in the lower ring area.

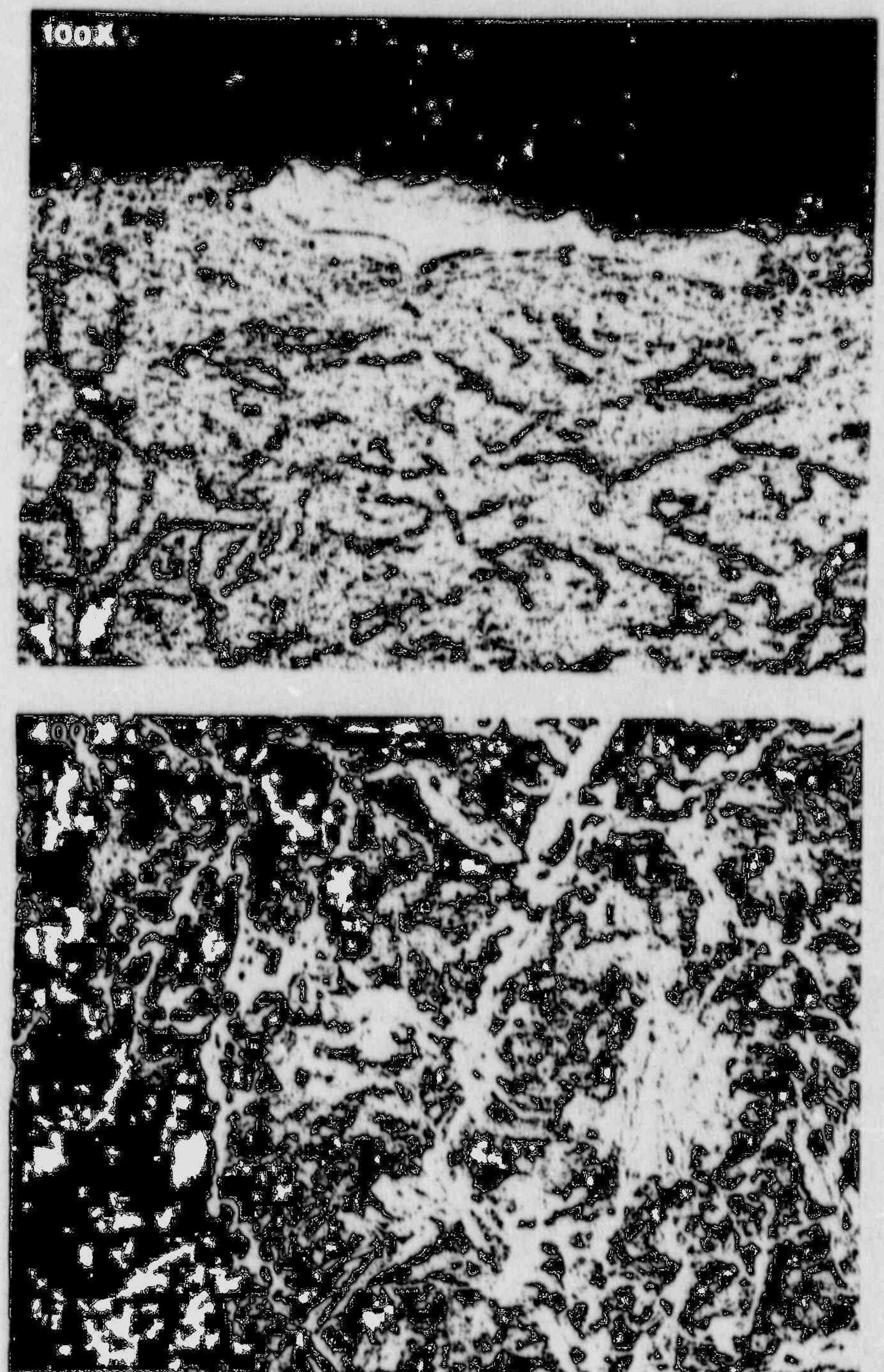
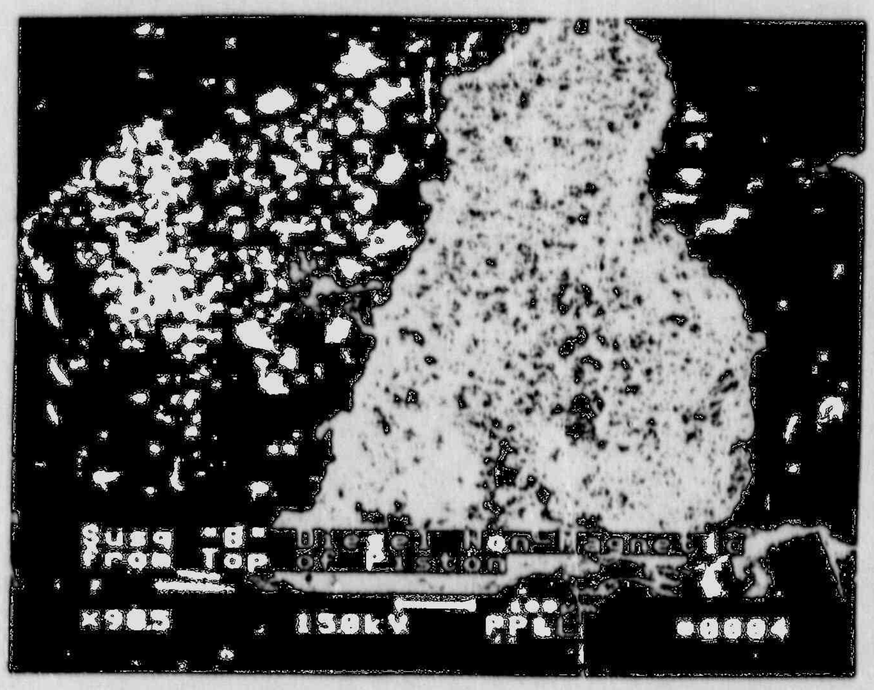


Figure B-2. Cross-section of the B diesel 7L piston skirt area showing deposited metal in the upper micrograph and martensite formation in the microstructure in the lower micrograph. The latter indicates temperatures in excess of 1350 deg F were experienced by the material locally.

Figure B-3. Enlarged section of the deposits on the skirt of the D diesel 7L cylinder showing the layering evident there.



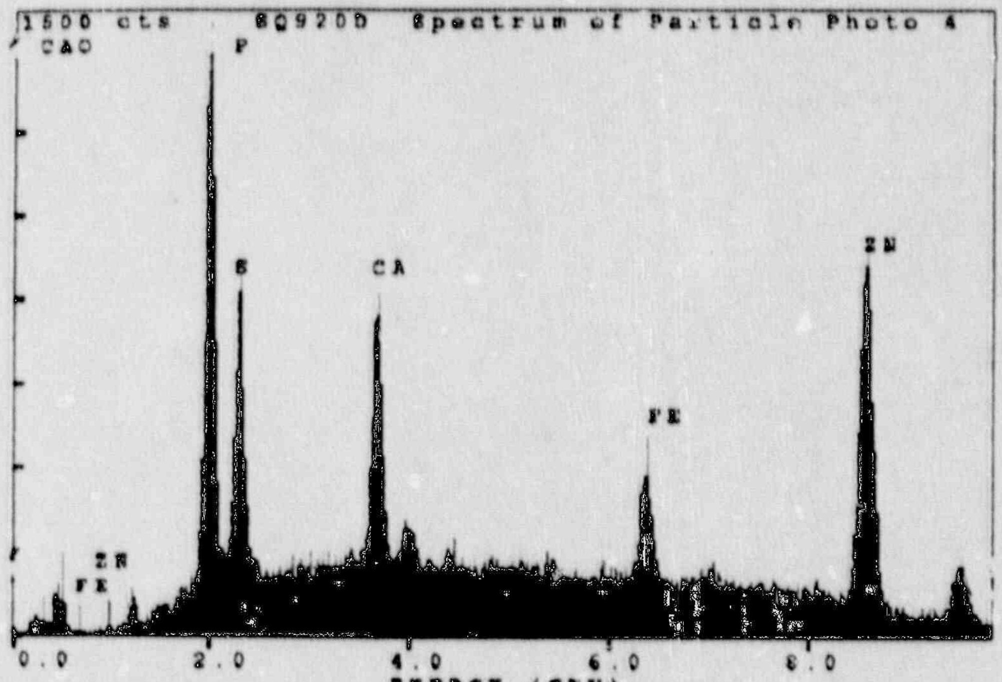
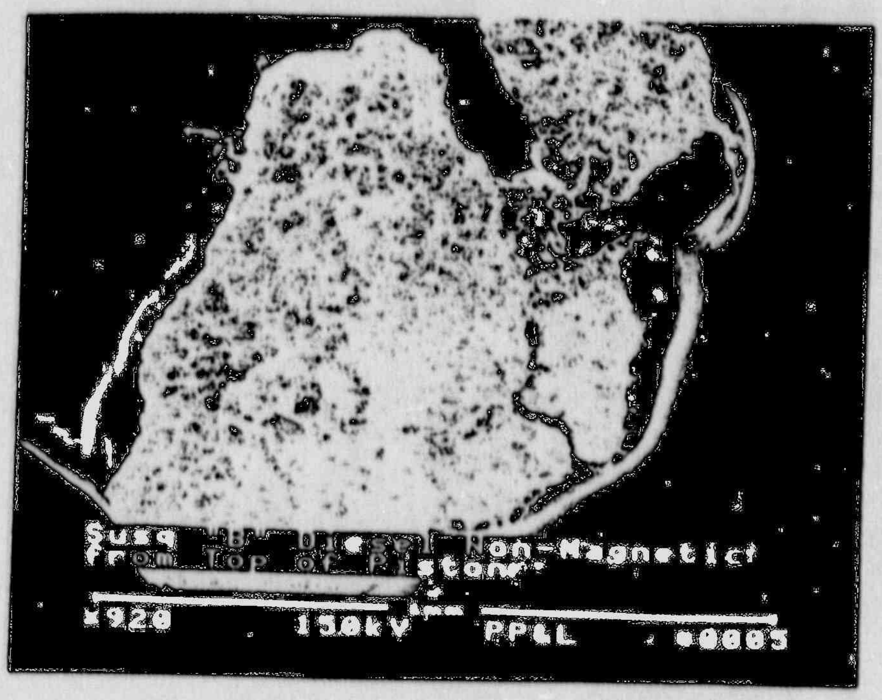


Figure B-4. Micrograph and EDS analysis results of non-magnetic debris taken from the head of the B diesel 7L piston.



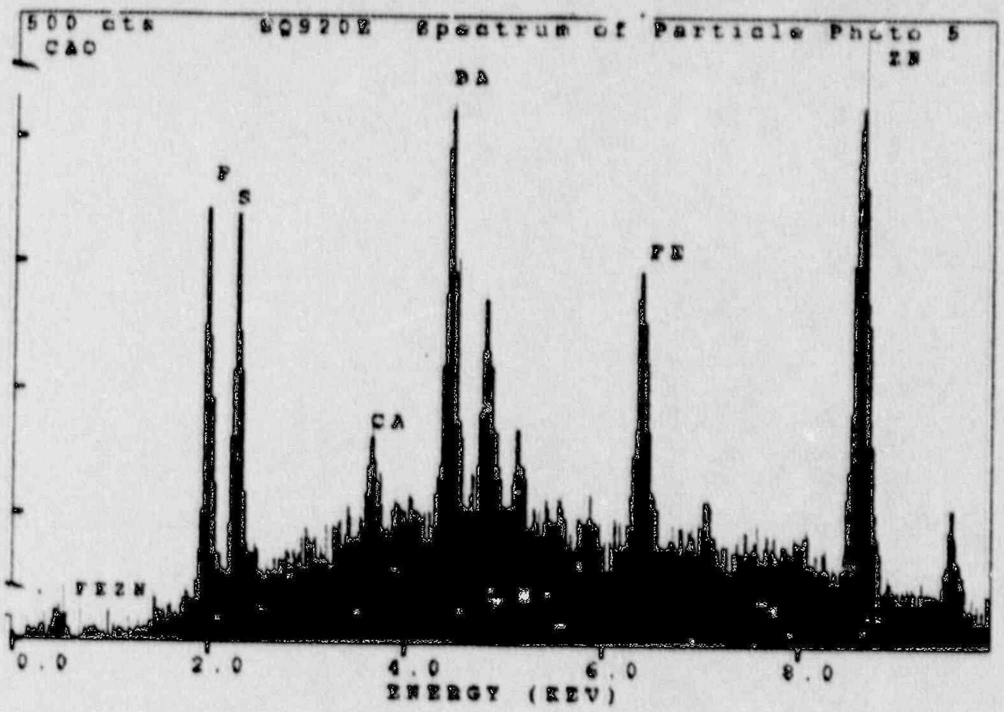


Figure B-5. Micrograph and EDS analysis results of non-magnetic debris taken from the head of the B diesel 7L piston on another particle from that analyzed in the figure B-4.

A

B

Figure B-6. Cross-section of a cracked area of the B diesel 7L piston skirt showing metal deposits on the surface.

Figure B-7. Enlarged area of the bottom right portion of figure B-6 showing where analysis 'P' was taken.

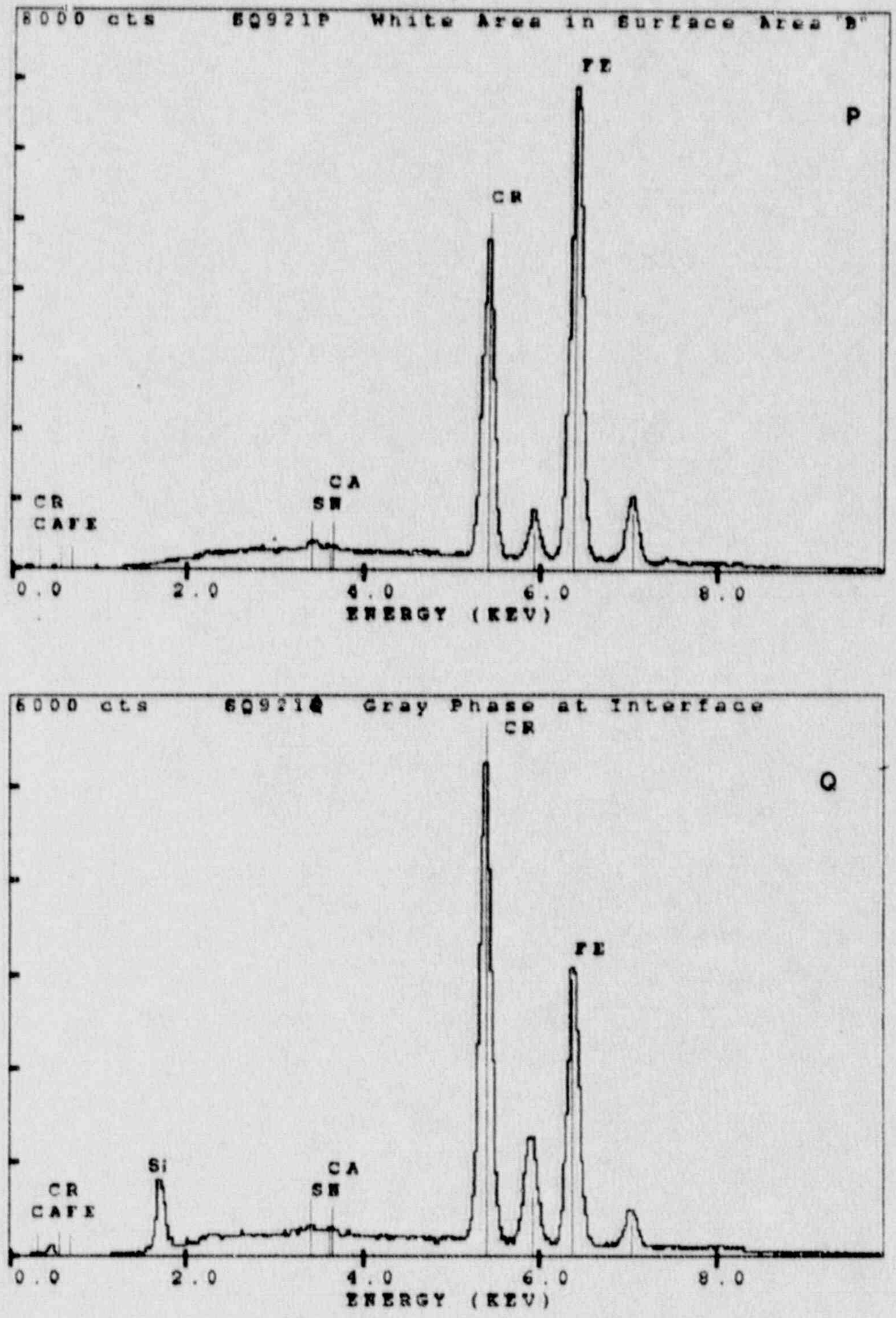
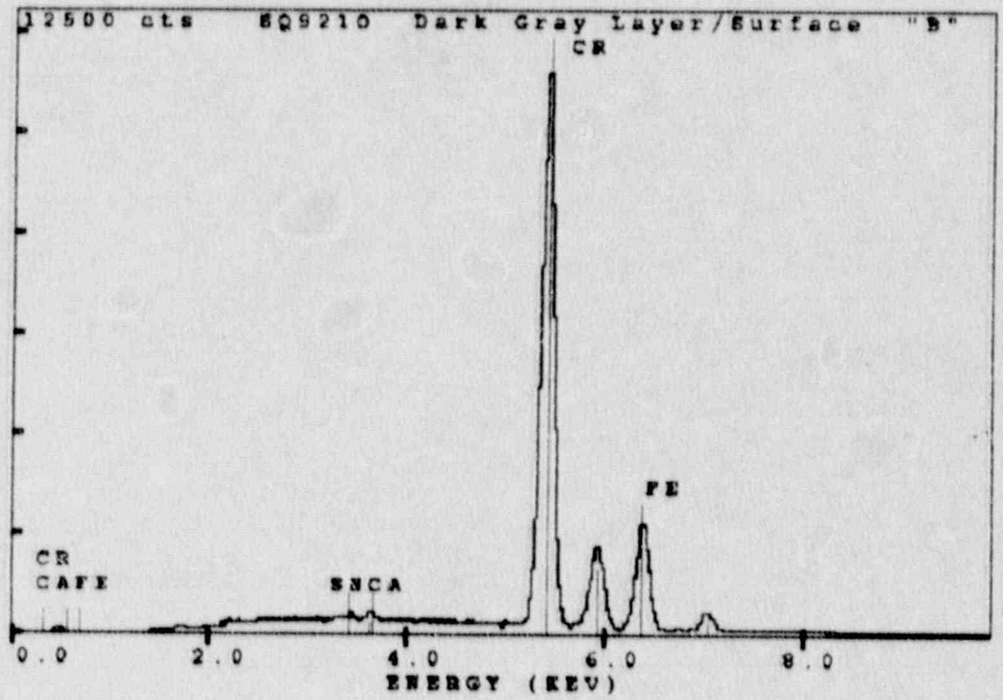


Figure B-8. EDS analysis of areas 'P' and 'Q' in figures B-7 and B-6.



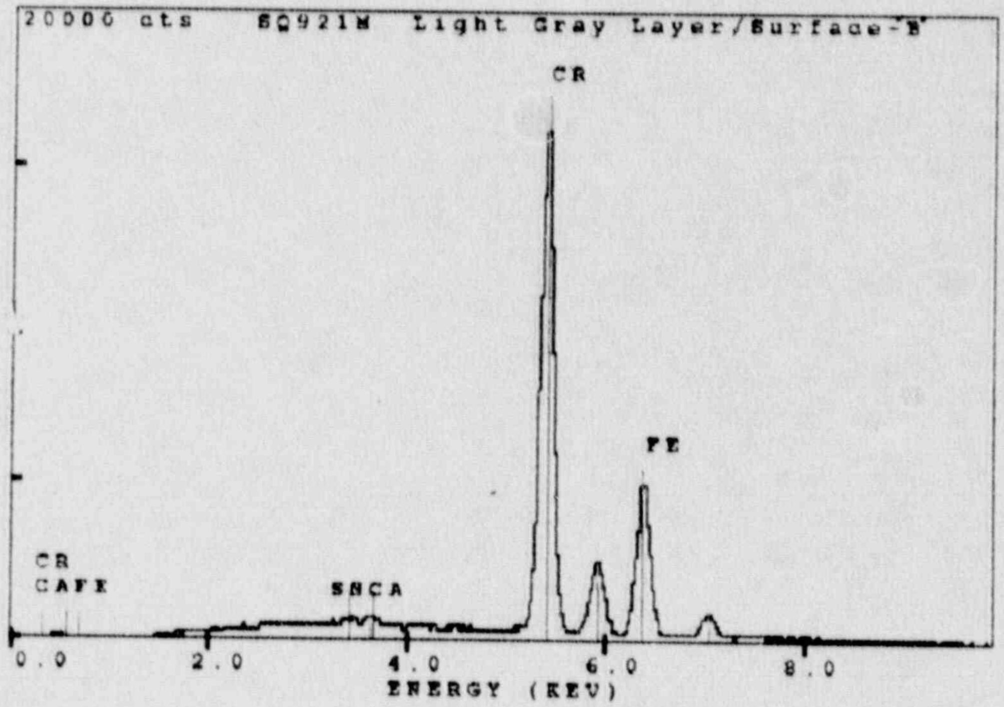


Figure B-9. Spectra of the light and dark phases in figure B-7.

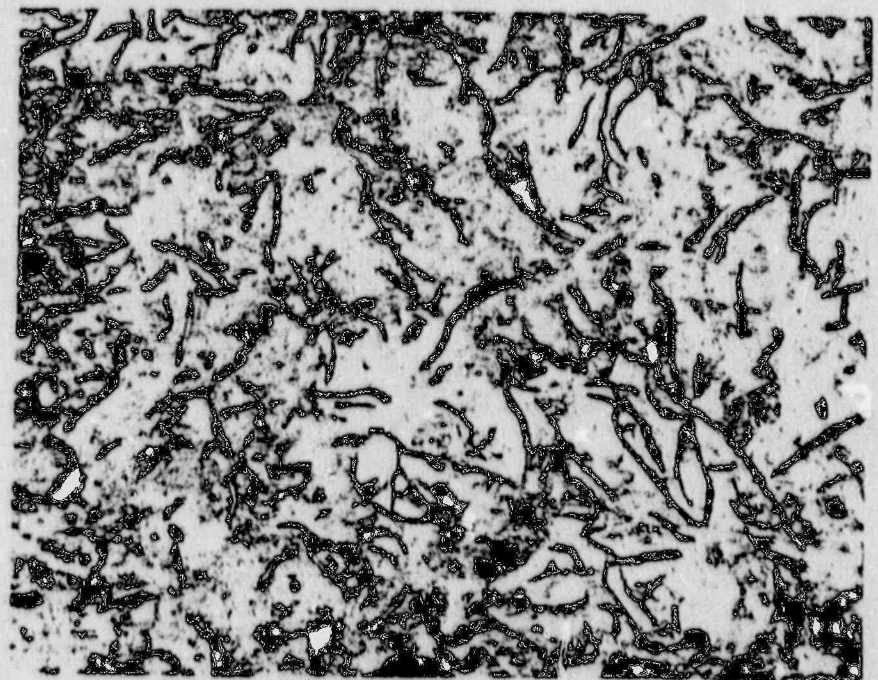


Figure B-10. Microstructure of the head of the B diesel 7L piston showing a normal pearlitic, graphite flake condition
a. Thick section b. Thinnest section

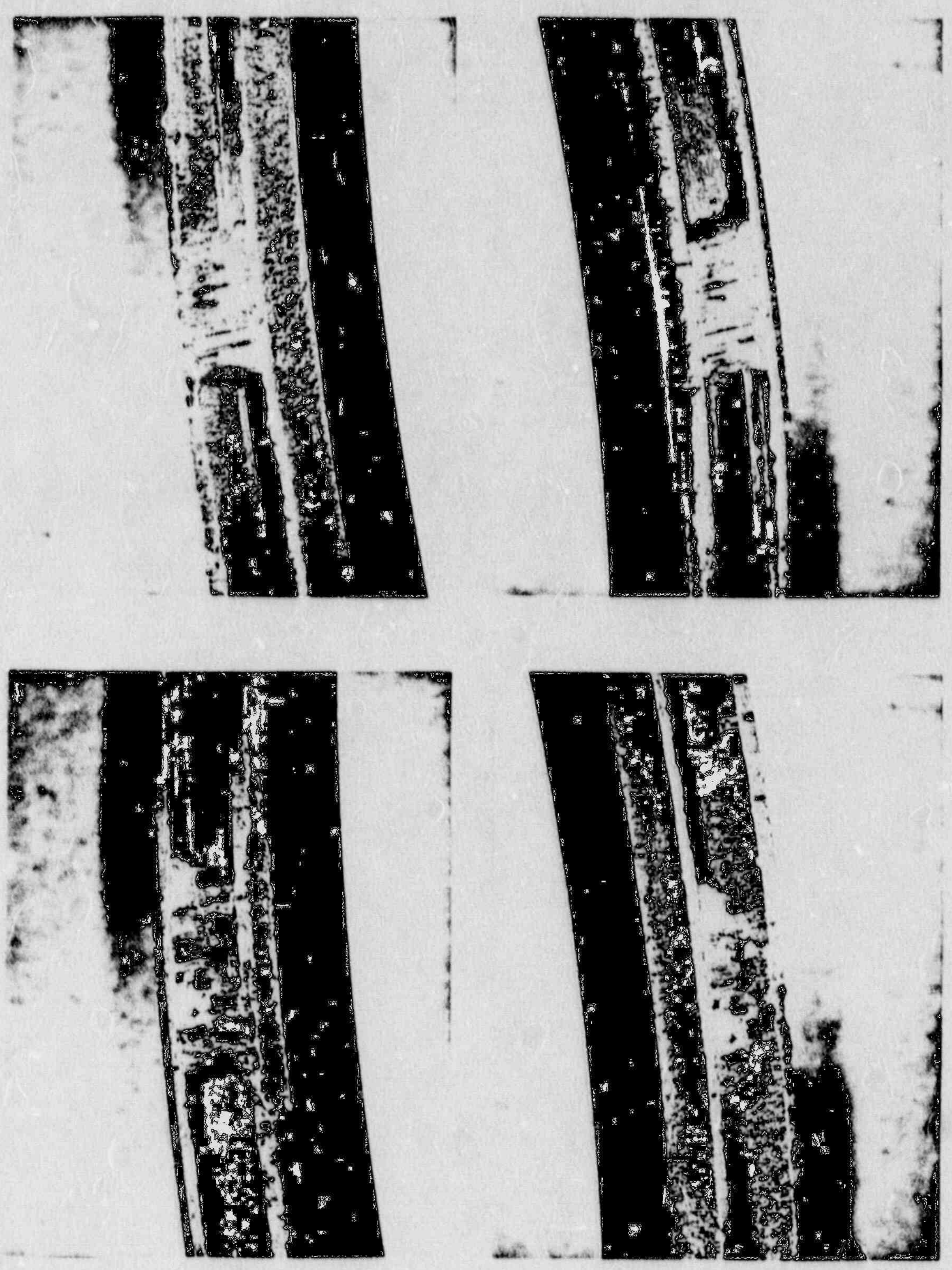


Figure B-11. Portions of the top wiper ring of the B diesel, 7L piston showing the pickup of metal and debris on the surfaces touching the liner.

Figure B-12. Cross section of the lower oil ring from the B diesel 7L piston showing the wear and deformation on the two wiper surfaces.

Figure B-13. Picture of oil ring profile at high magnification. Magnified image of oil ring finger from B diesel, 7L piston lower oil ring showing structure deformation and size of the finger.

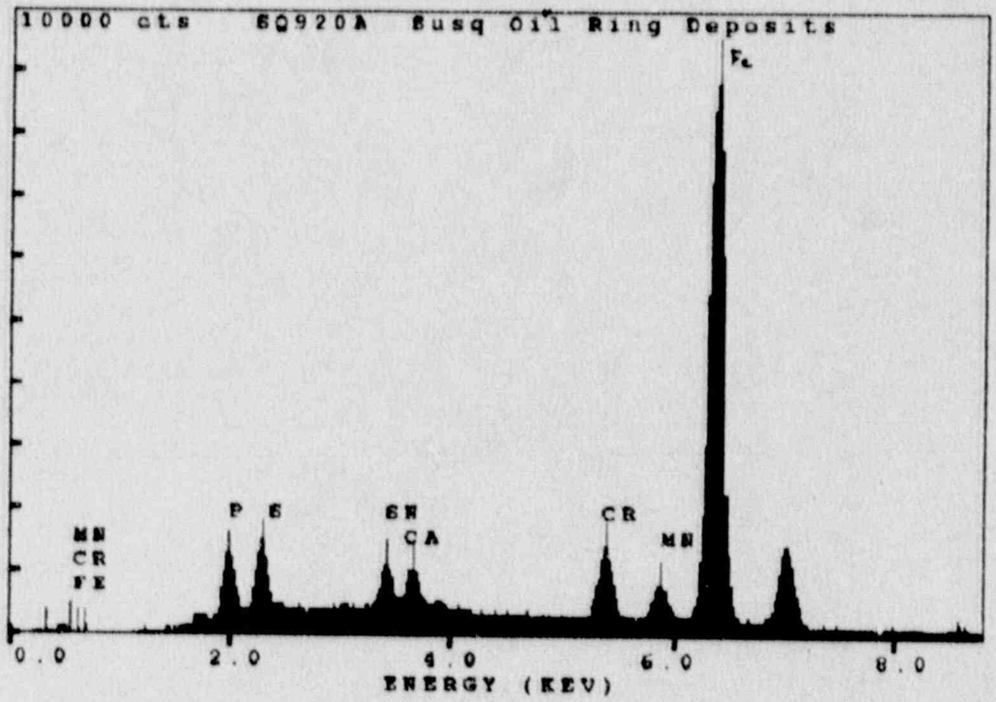


Figure B-14. Debris EDS analysis from the spring behind the lower oil ring from the B diesel 7L piston.

Figure B-15. Smeared metallic deposits found on the #4 compression ring of the B diesel, 7L piston are shown here as the dark gray area labeled 'A'. EDS analysis of area 'A' shows it to contain Cr, Fe, Si(trace) and Sn(trace). Area 'B' shows only Fe with a trace of Cr.

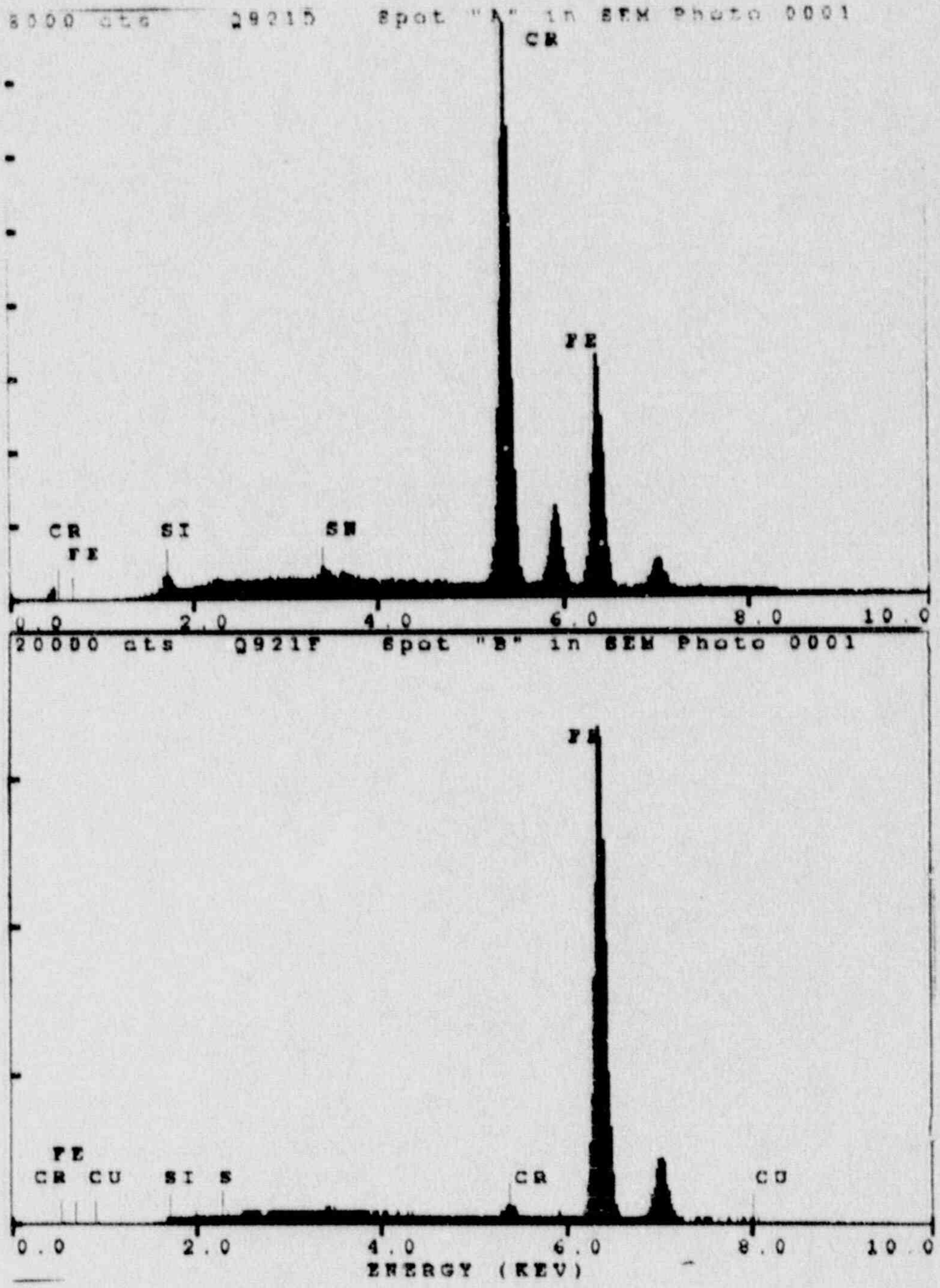


Figure B-16. EDS analysis of the #4 compression ring from the B diesel 7L piston in areas 'A' and 'B' of Figure B-15.

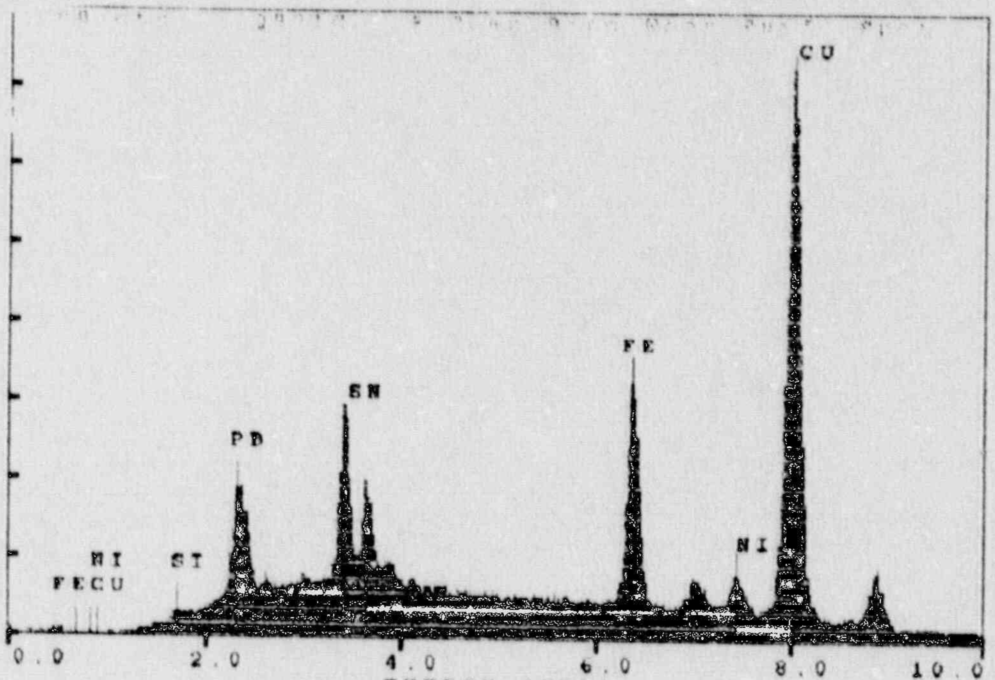


Figure B-17. An unusual spot of foreign material found embedded in the #4 compression ring from the B diesel 7L piston. EDS analysis shows it to contain Cu, Fe, Sn, Pb, and Ni.

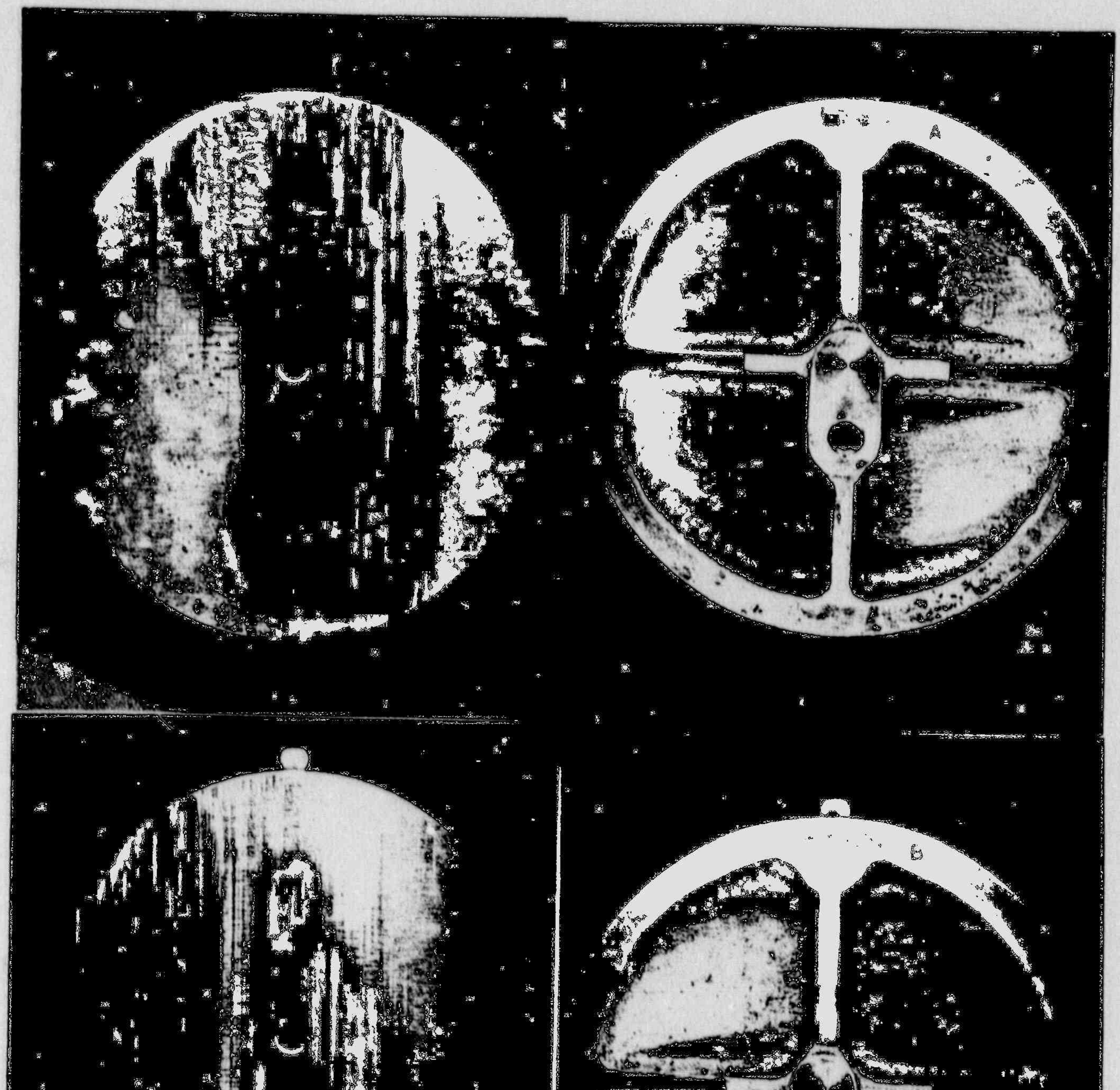


Figure B-18. Pictures of the end caps as found on the B diesel 7L piston showing the scraping and local heating experienced by these parts during the failure sequence.

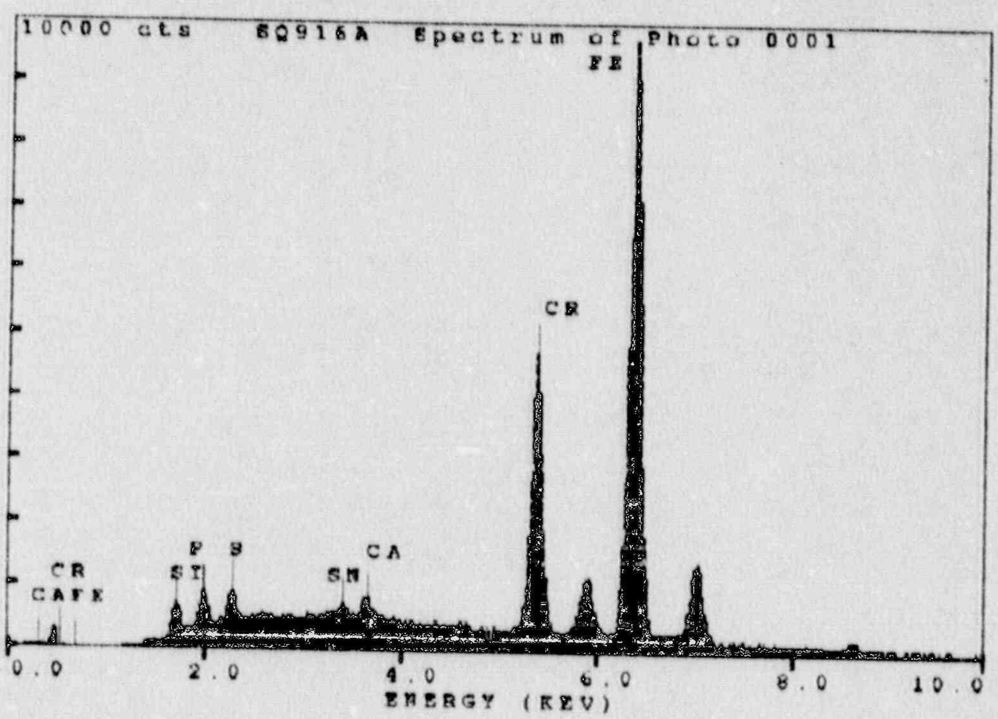
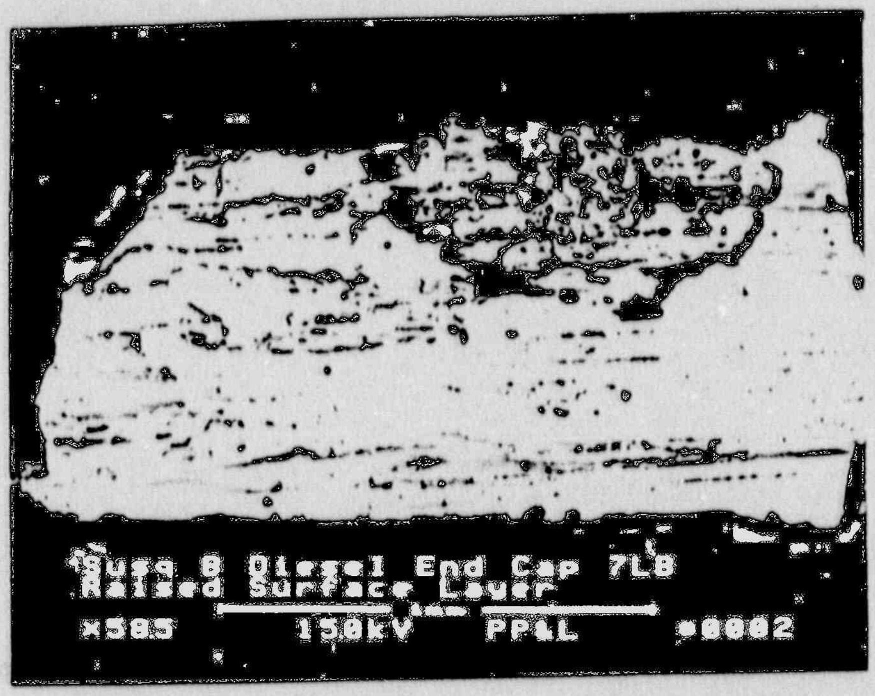


Figure B-19. Picture and EDS analysis of a piece of smeared metal removed from end cap 'A' from the B diesel 7L piston.



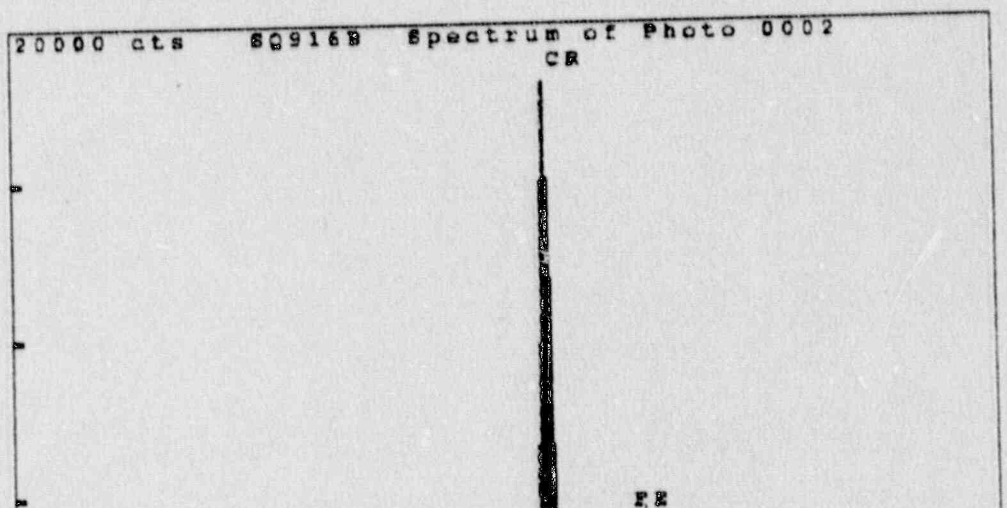


Figure B-20. Particle of smeared metal removed from pin end cap 'B' from the B diesel 7L piston. The EDS analysis shows that it consists primarily of Cr with some Fe and a trace of Ca and Sn.

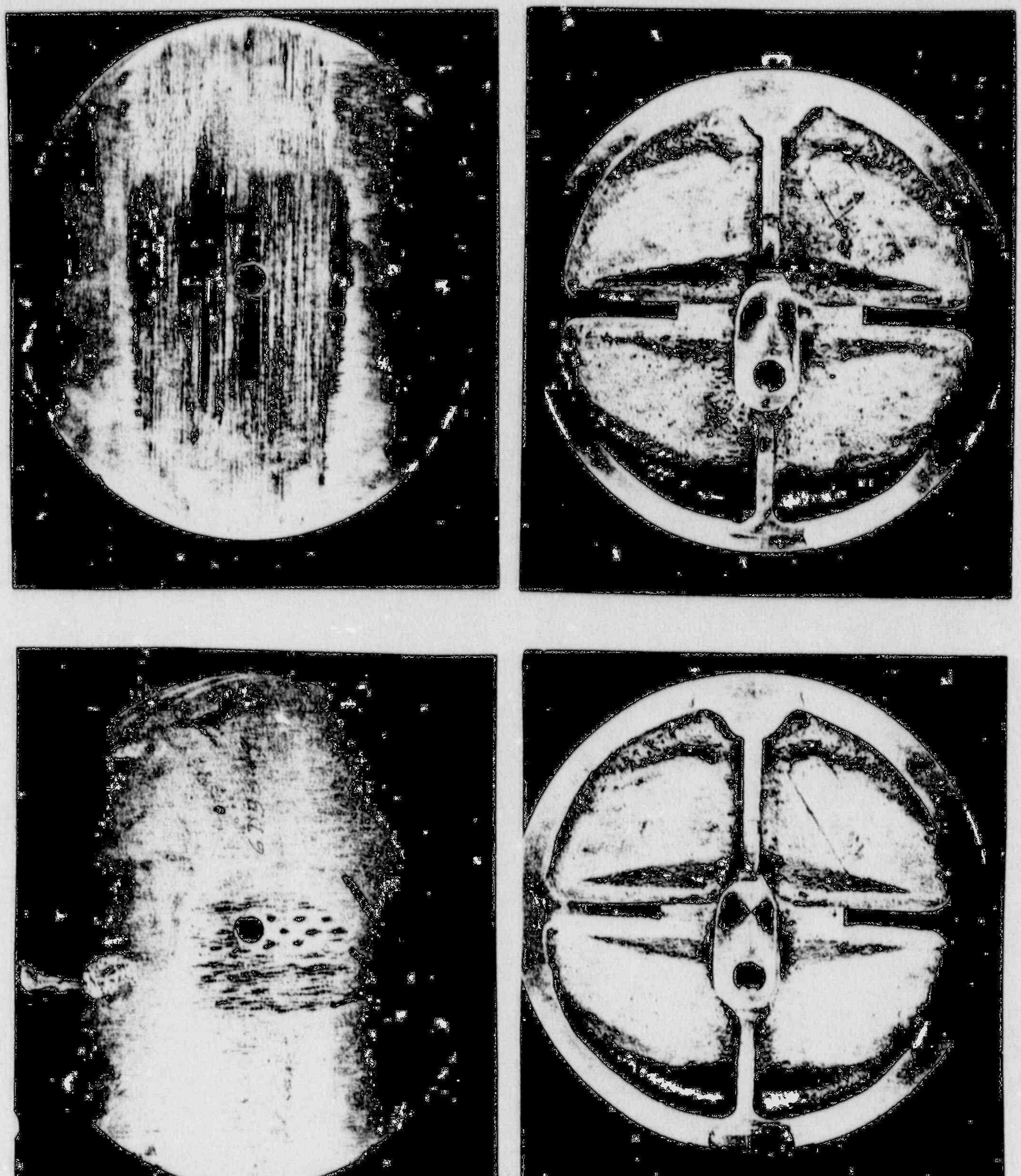


Figure B-21. Abrasion and wear found on the B diesel 7R piston pin end caps.

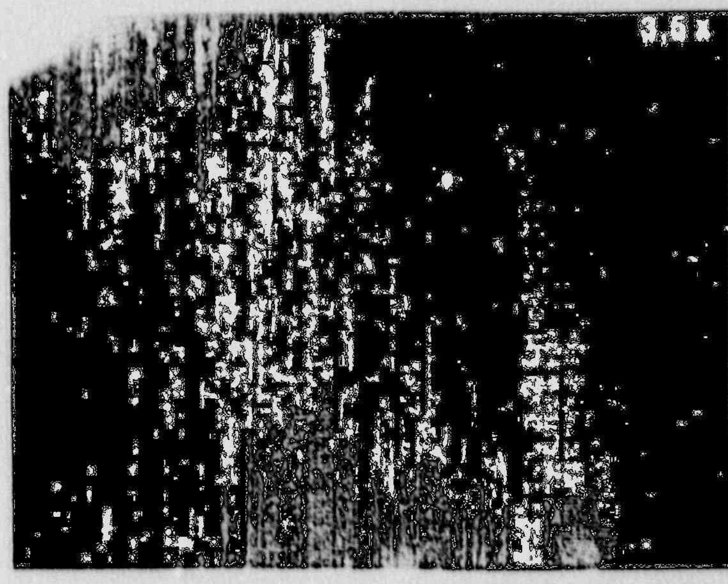
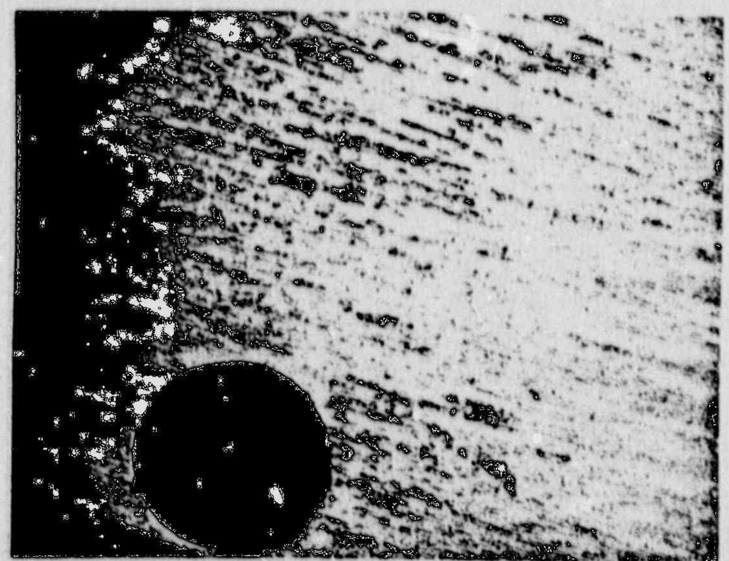
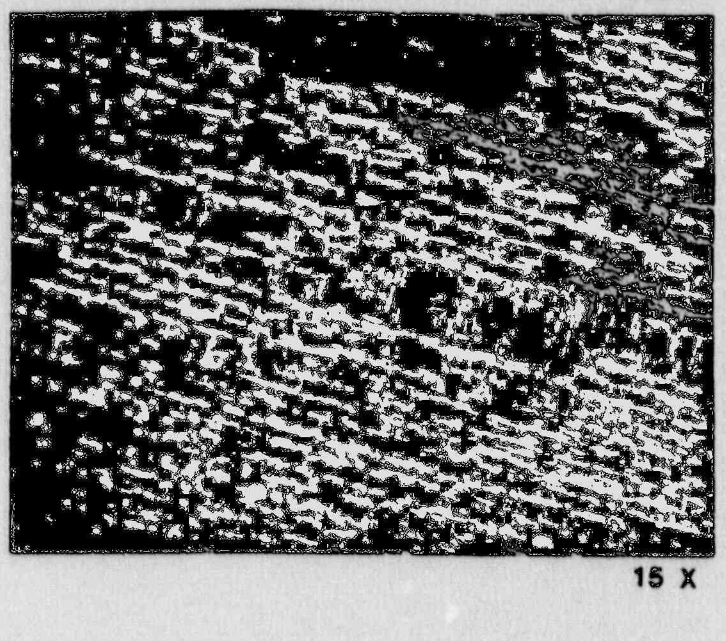


Figure B-22. Photograph of the B diesel 7R piston end cap which showed the worst amount of wear, heating and metal transfer.



3.5 X



15 X

Figure B-23. B diesel 7R end cap showing the least amount of wear, but does show that the cap was touching the surface of the liner as evidenced from the light scratch marks on the surface perpendicular to the machining marks.

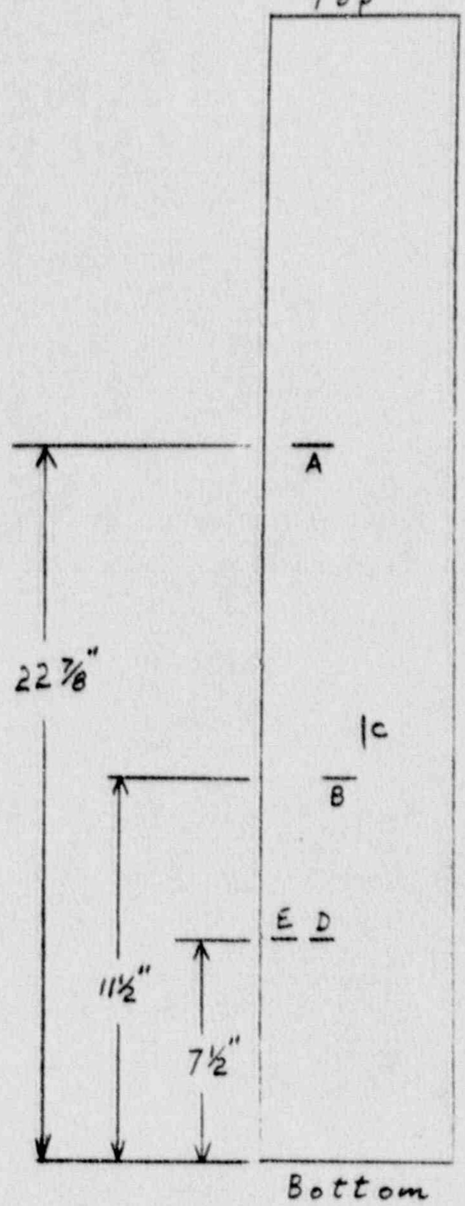


Figure B-26. Sketch of the thrust side of the B diesel 7L liner showing the location and orientation of the sections removed for metallurgical cross-sectioning(part labeled B-1).

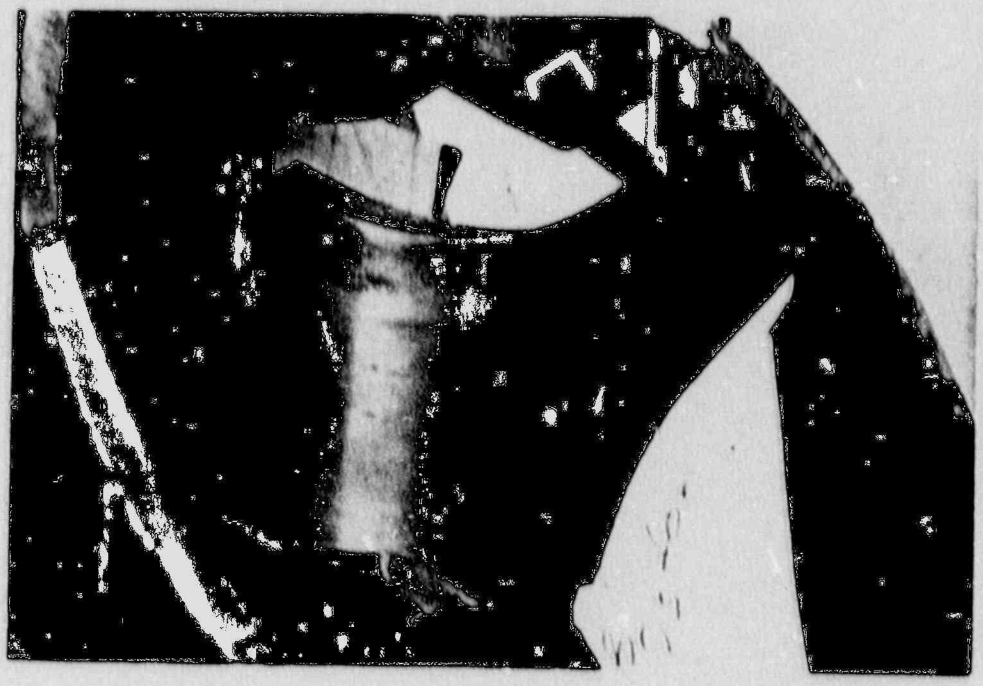


Figure B-24. Photographs of both ends of the B diesel 7L piston bronze bushing showing the heat tinting and decomposed oil products near the ends 90 degrees to the top of the bushing.

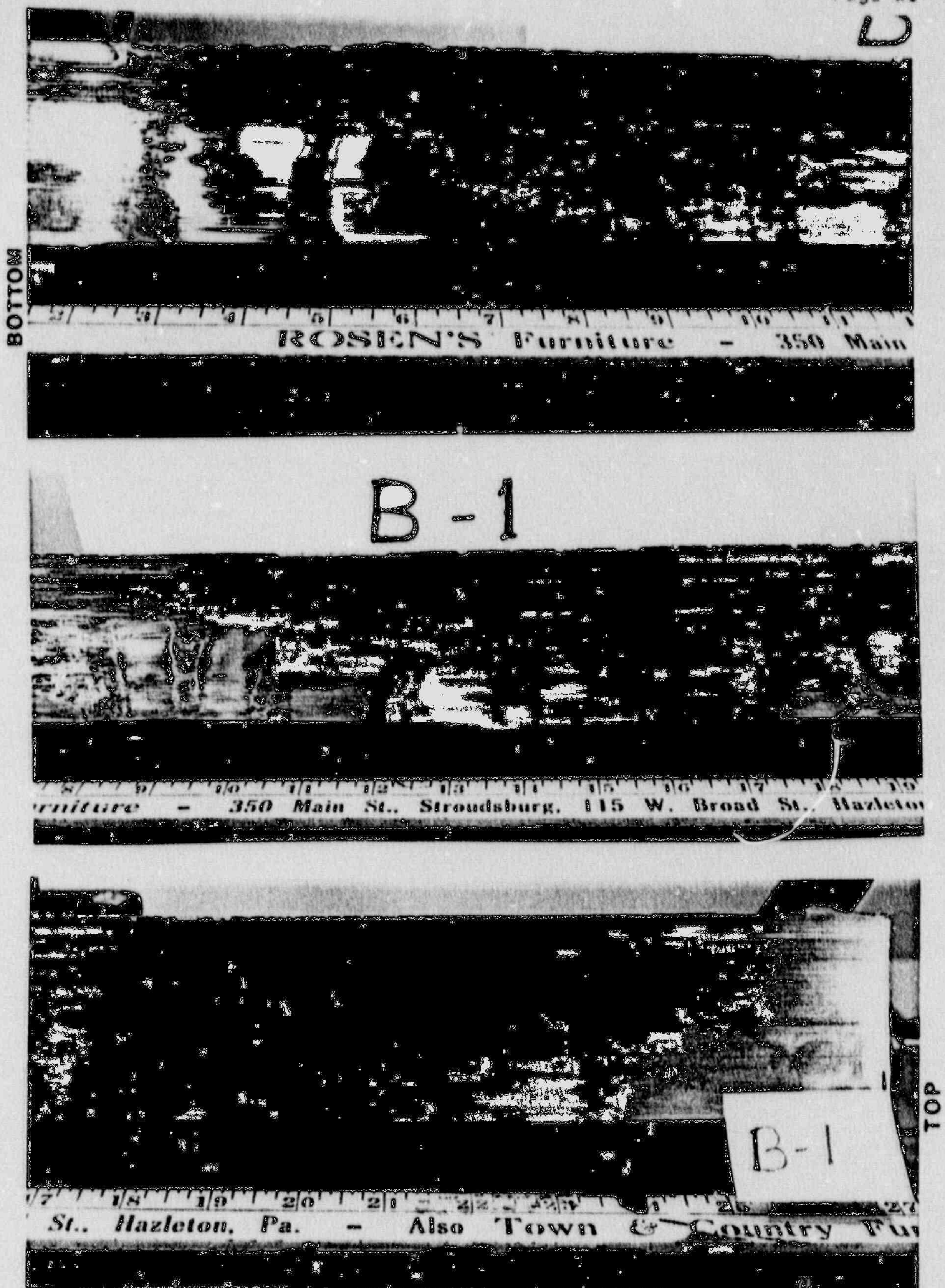
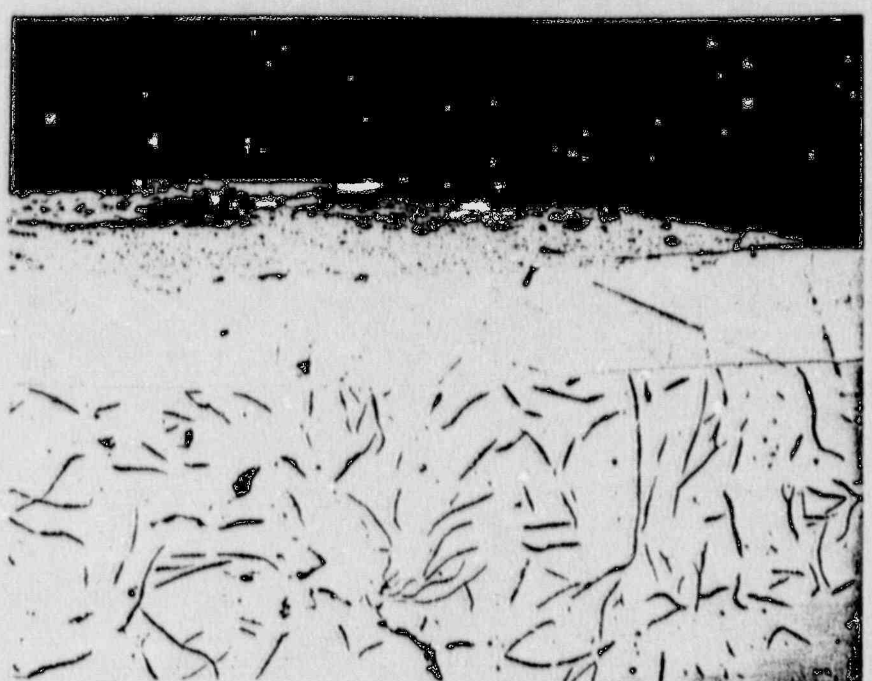
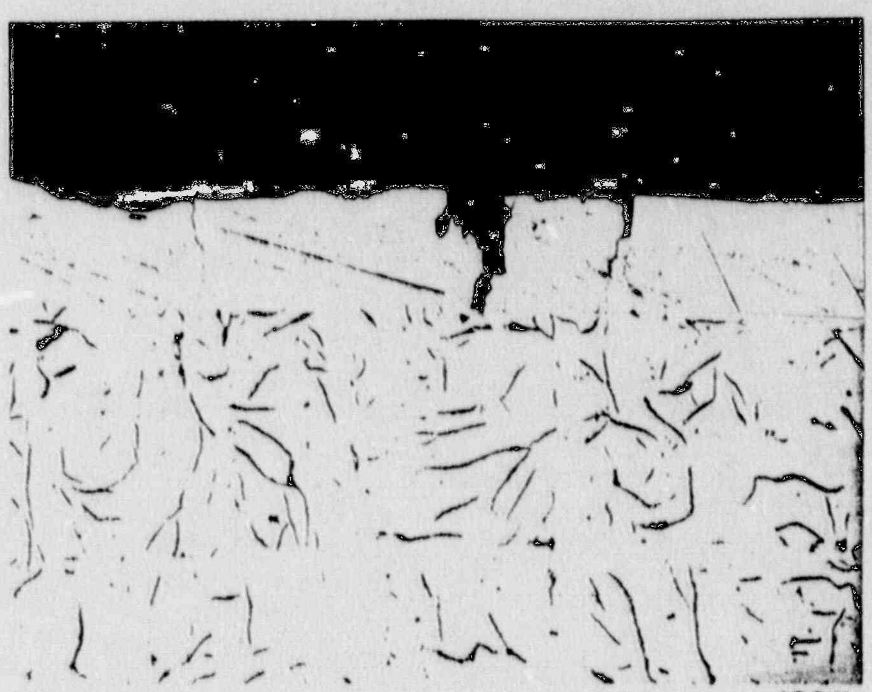


Figure B-25. Full length view of the B diesel 7L liner surface on the thrust side of the liner(identified as piece B-1)

Figure B-27. Transverse cross-sectional views of the thrust side of the B diesel 7L liner 22 7/8 inches from the bottom(piece 'A'). The chromium plating is cracked and some of the pores are filled with metallic particles.



100X



100X

Figure B-28. Longitudinal view of the B diesel 7L liner on the thrust side 11 1/2 inches from the bottom showing metallic deposits, filled pores and cracking in the Cr plating(piece labeled 'B').

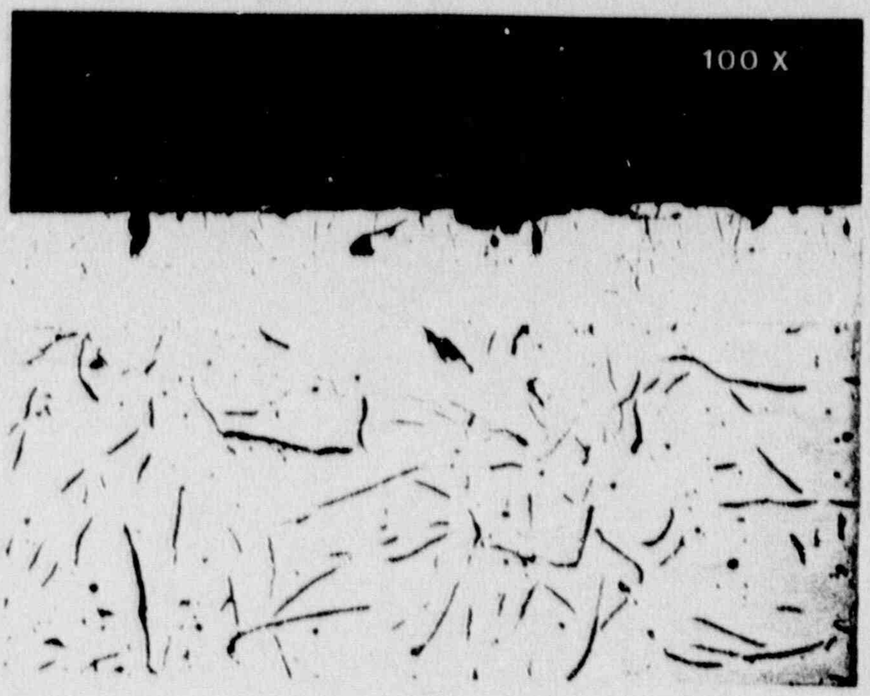
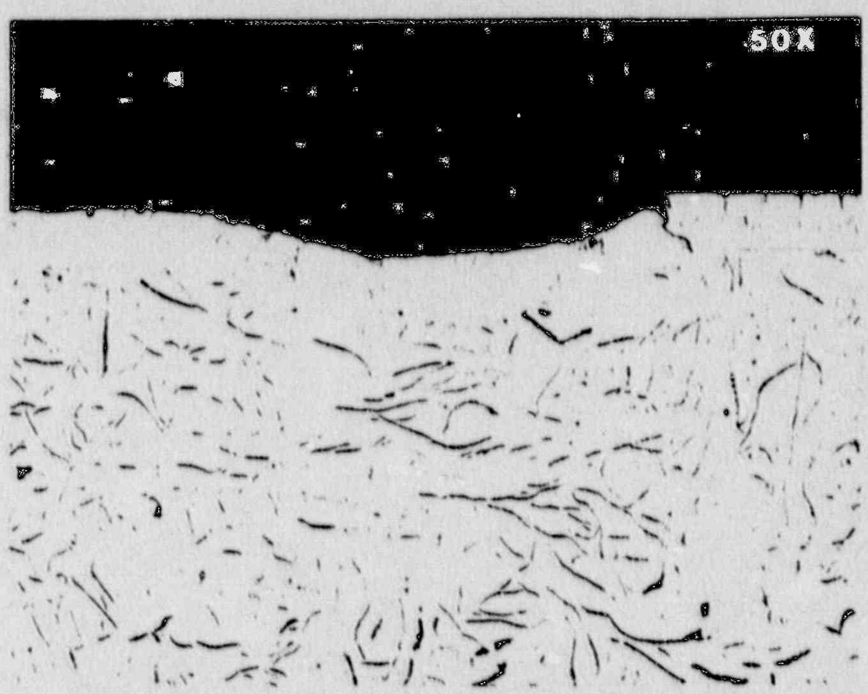


Figure B-29. Longitudinal cross-section (labeled 'C') taken 12 inches from the bottom of the B diesel 7L cylinder liner showing a rough fractured surface with pieces missing and some debris in the pores of the chromium plating layer.

Figure B-30. Transverse cross-section(labled area 'D') taken 7 1/2 inches from the bottom of the B diesel 7L liner showing depressions in the Cr plating surface, cracking of the base metal casting and deposited metal.



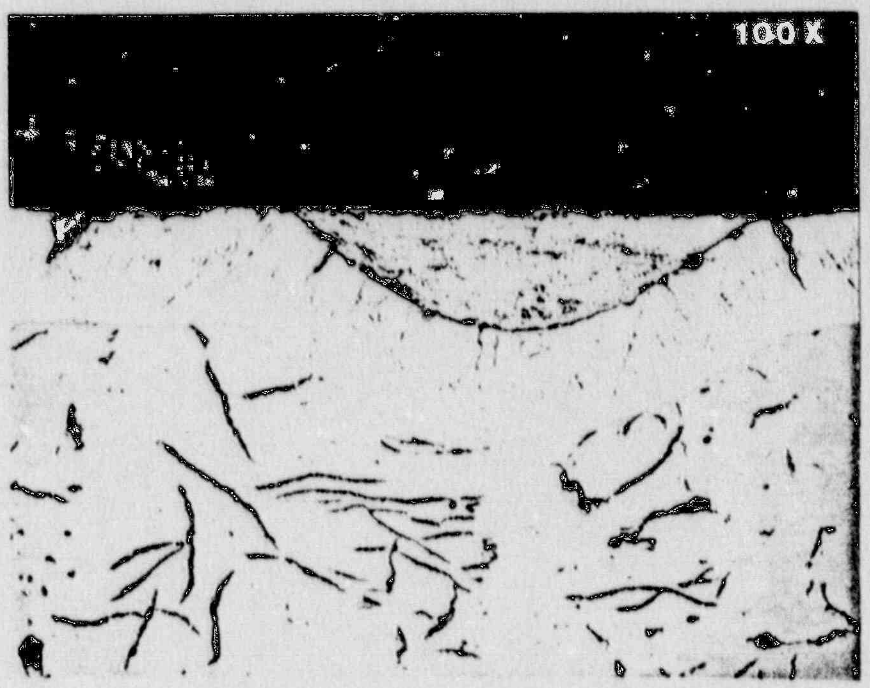


Figure B-31. Transverse cross-section (Labeled area 'E') of the thrust side of the B diesel 7L liner taken 7 1/2 inches from the bottom showing depressions in the Cr plating and deposited metal.

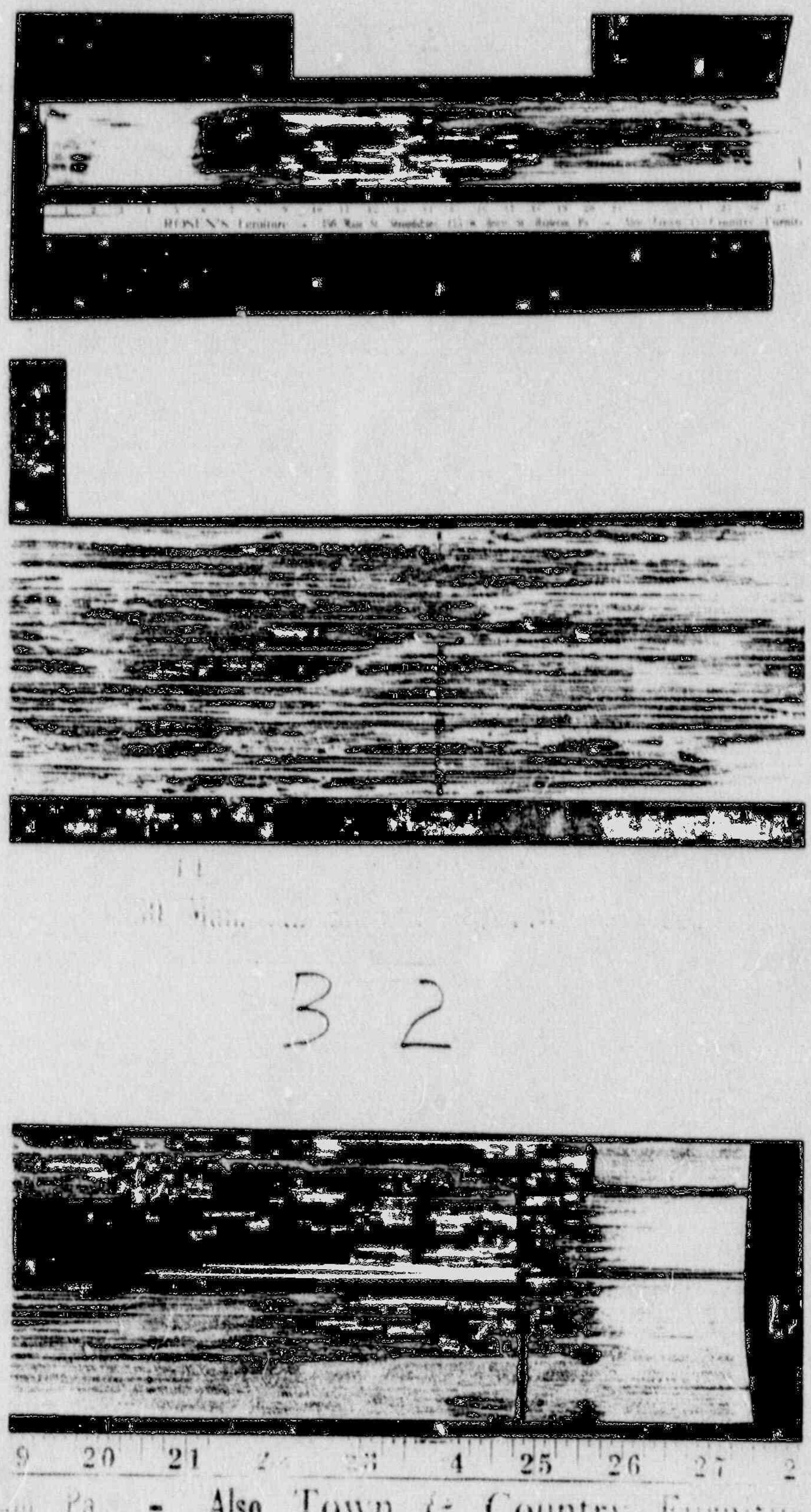


Figure B-32. Views of the length of the B diesel 7L cylinder liner in the area of pin end cap scraping (part B-2) showing heavy longitudinal marks and overheating over 20.5 inches of the center section.

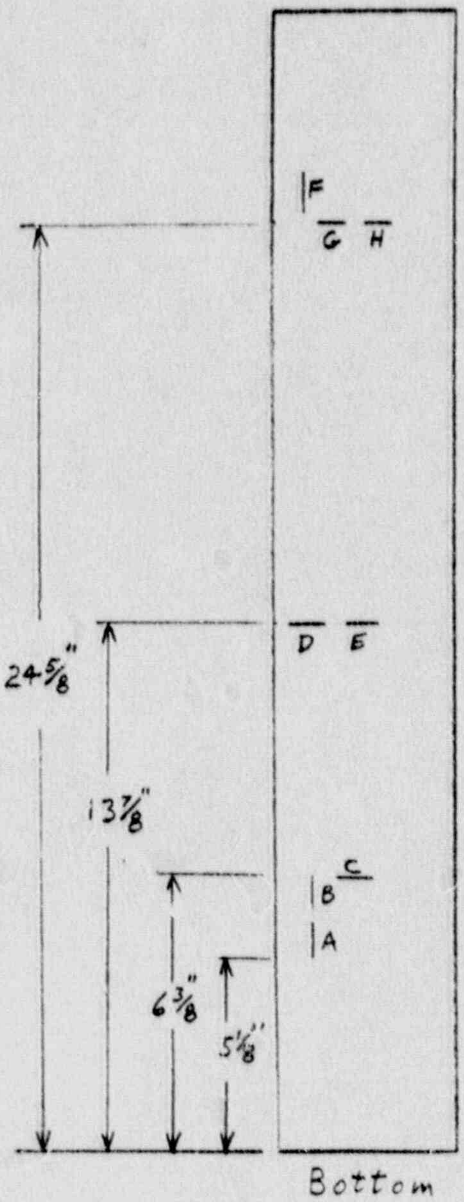
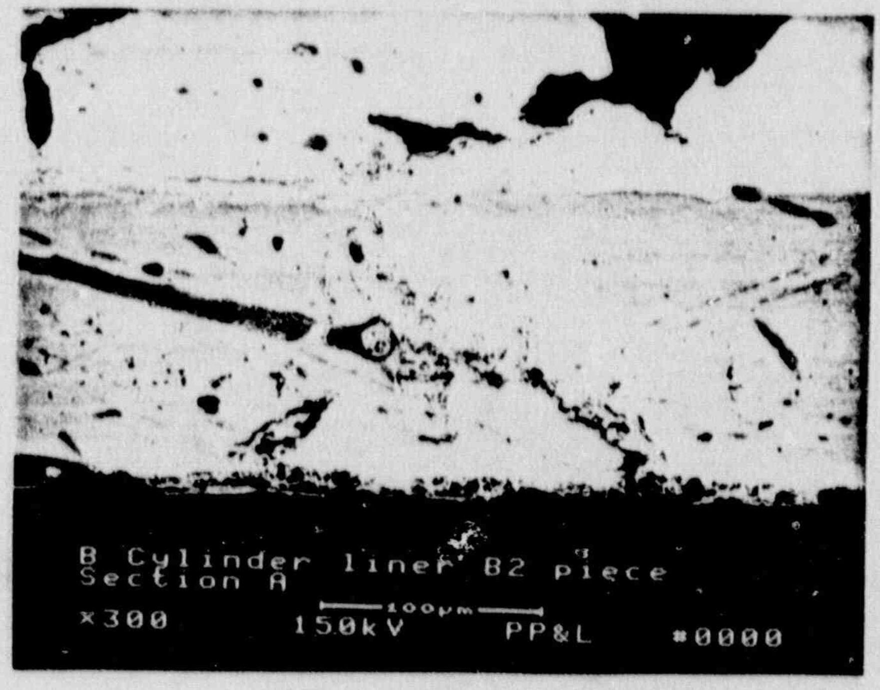


Figure B-33. Sketch of the end cap side of the 'B' diesel 7L liner showing the locations of the metallurgical samples taken for cross-sectional examinations(part labeled B-2).



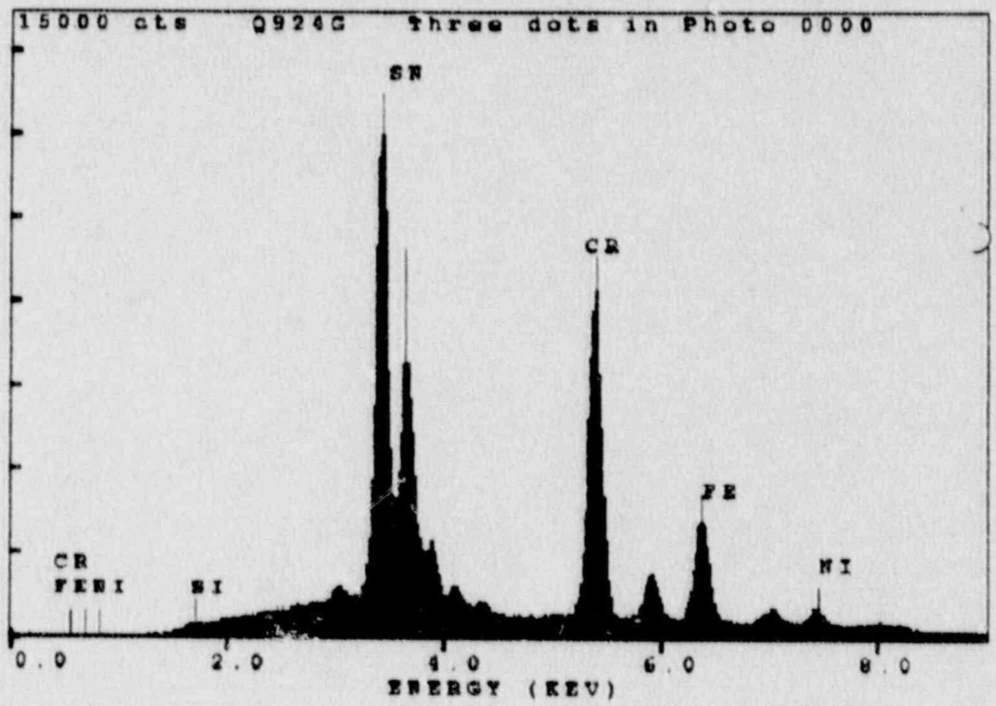
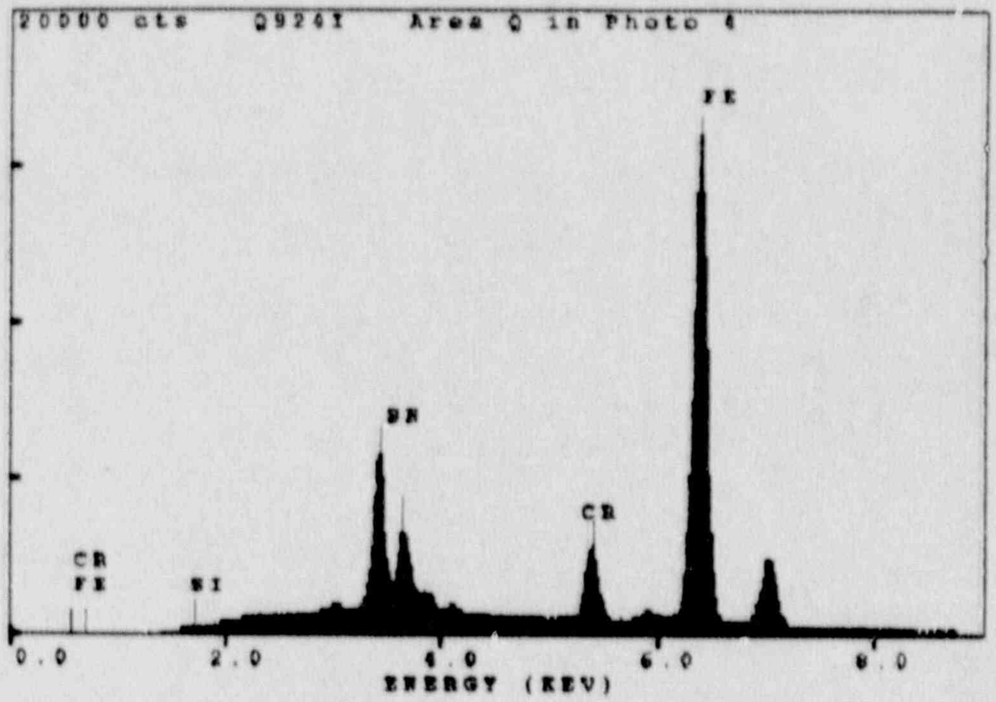


Figure B-34. Longitudinal section of B diesel 7L liner (labeled 'A' of part B-2) showing cracking in the Cr plating surface and the presence of Sn at the Cr-Fe base metal interface.

Area	Elements(greatest to least)
'R' Base of pit	Cr, Fe, and Sn
'Q' Top of pit	Fe, Sn, and Cr
'P' Center layer	Fe, Cr, and (Sn)
'P' Topmost layer	Fe, cr, and Si

Figure B-35. Transverse cross-section of the B diesel 7L liner (labeled 'C' of part B-2) showing areas of metal buildup and filling of pores with Cr,Sn and Fe.



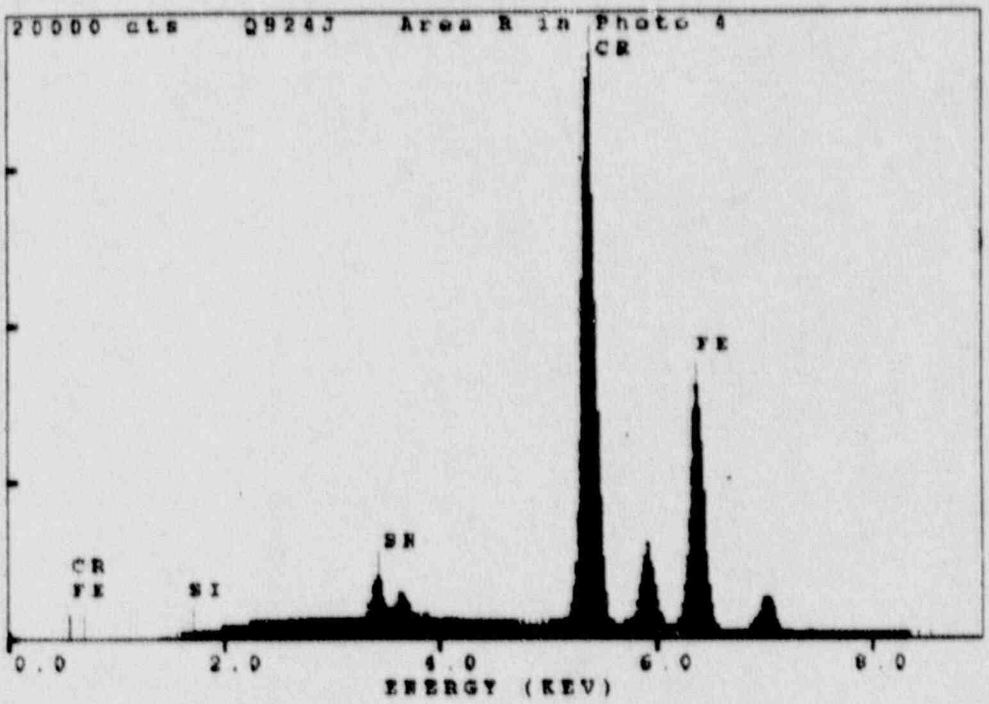


Figure B-36. EDS analysis of areas Q and R shown in figure B-35.

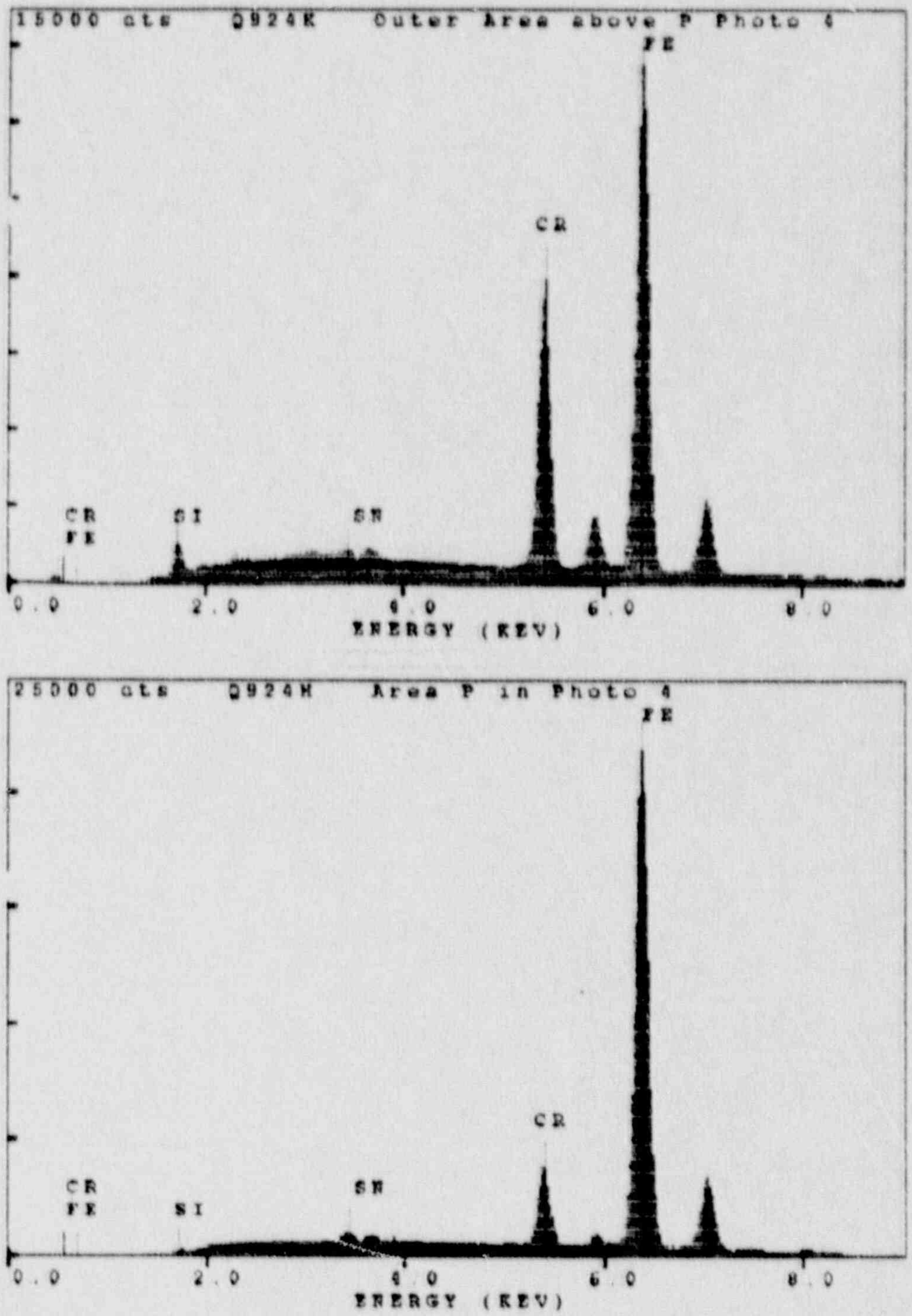
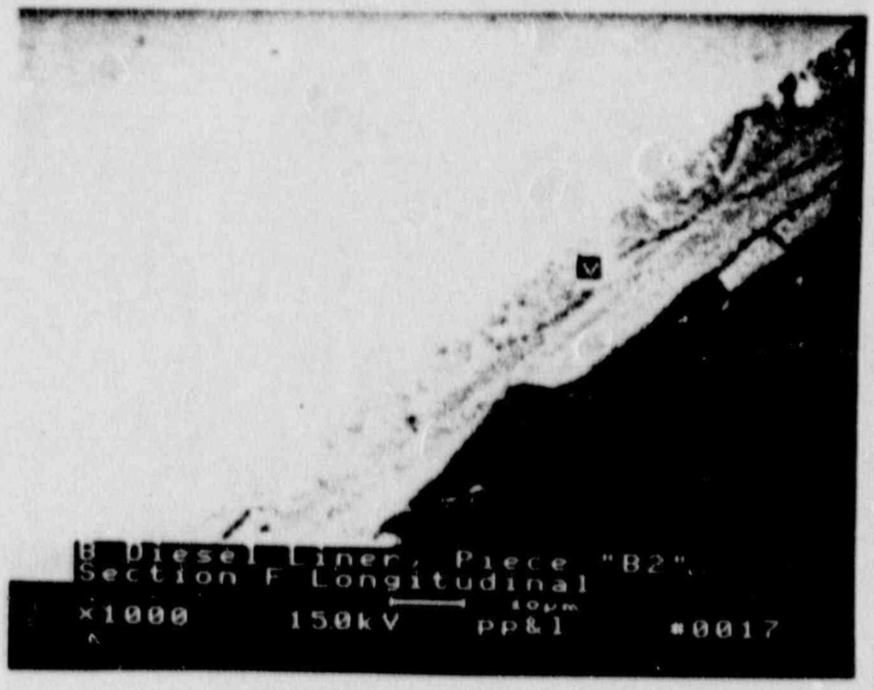


Figure B-37. EDS analysis of area P and the Dark layer above P in figure B-35.

Figure B-38. Longitudinal section 'F' from B diesel liner 7L piece B-2 taken 24 5/8 inches above the bottom. This section does not show any pores filled or otherwise, but does show several cracks through the thickness of the Cr plate.



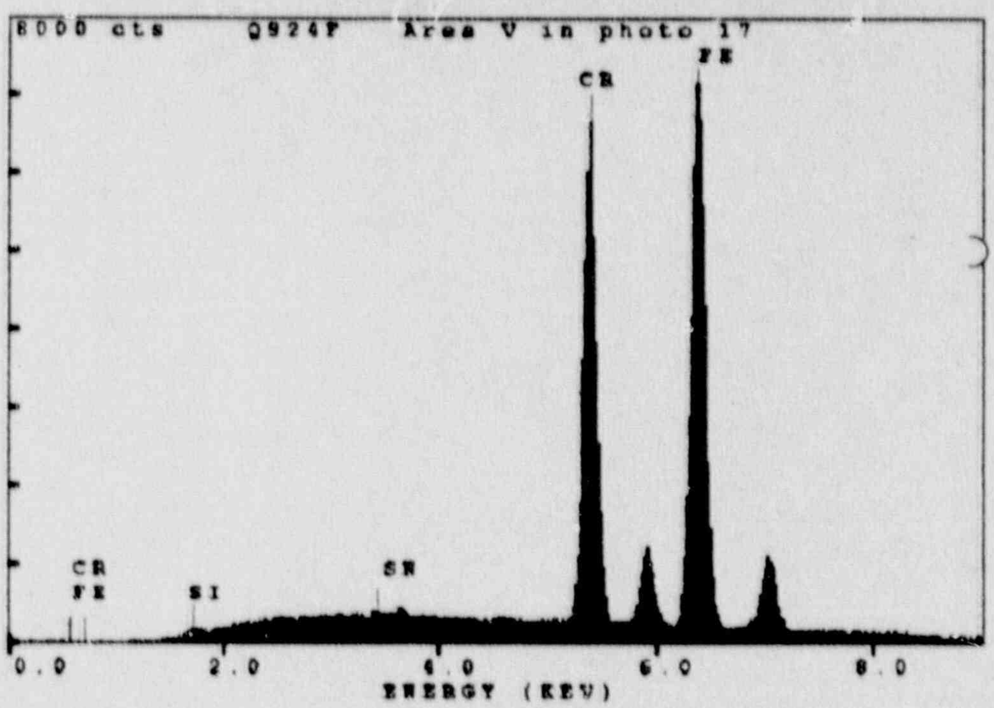
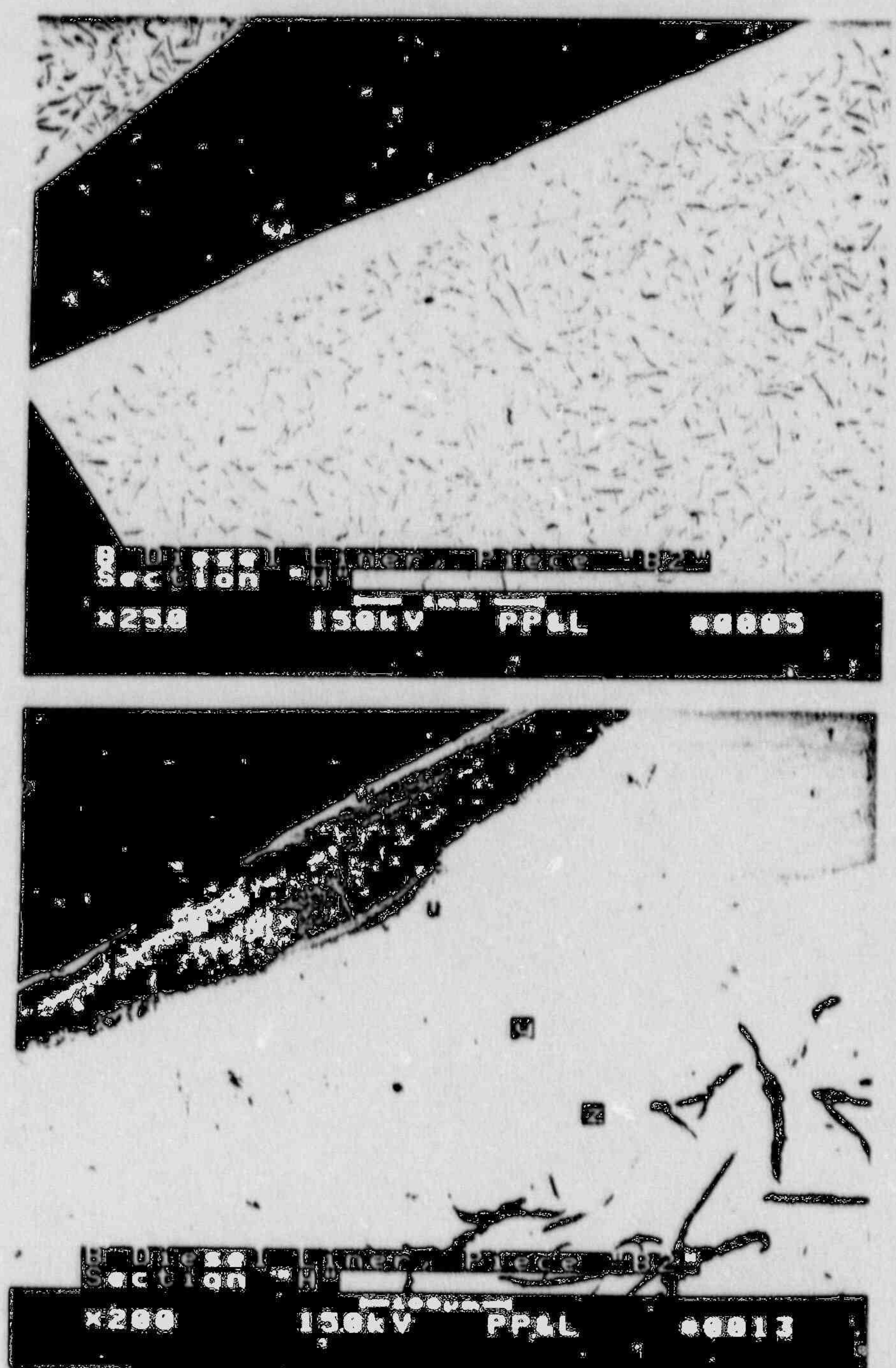
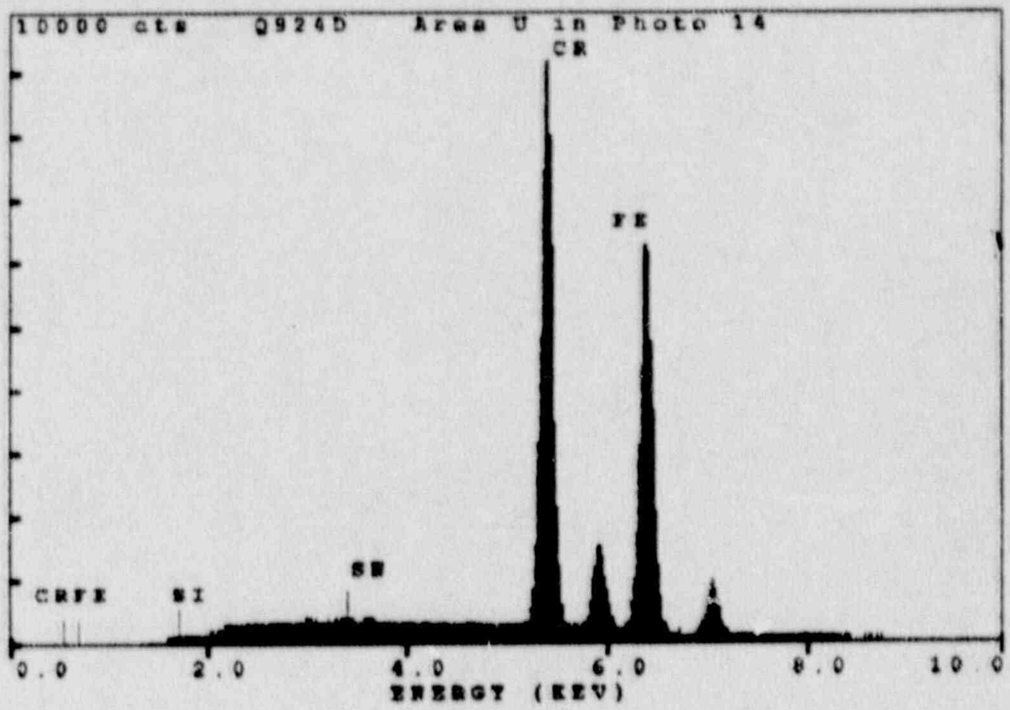


Figure B-39. Longitudinal section of B diesel 7L liner taken from the end cap side 24.625 inches from the bottom showing the layer of deposited metal and the EDS analysis of the lighter grey area.



Layer		Elements in decending order.
Z	Base metal casting	Fe
Y	Plating layer	Cr
U	Content of pore	Cr, Fe, Sn(Trace)
X	Thick dark layer	Cr, Fe, Si
W	Top layer	Fe, Cr, Sn(Trace)

Figure B-40. End cap side of the B diesel 7L liner taken from transverse section H showing damage to the chromium liner and deposited metal. EDS analysis of the various layers are shown in figures B-41 and B-42.



file name: /usr/tmp/resprEAAa01622

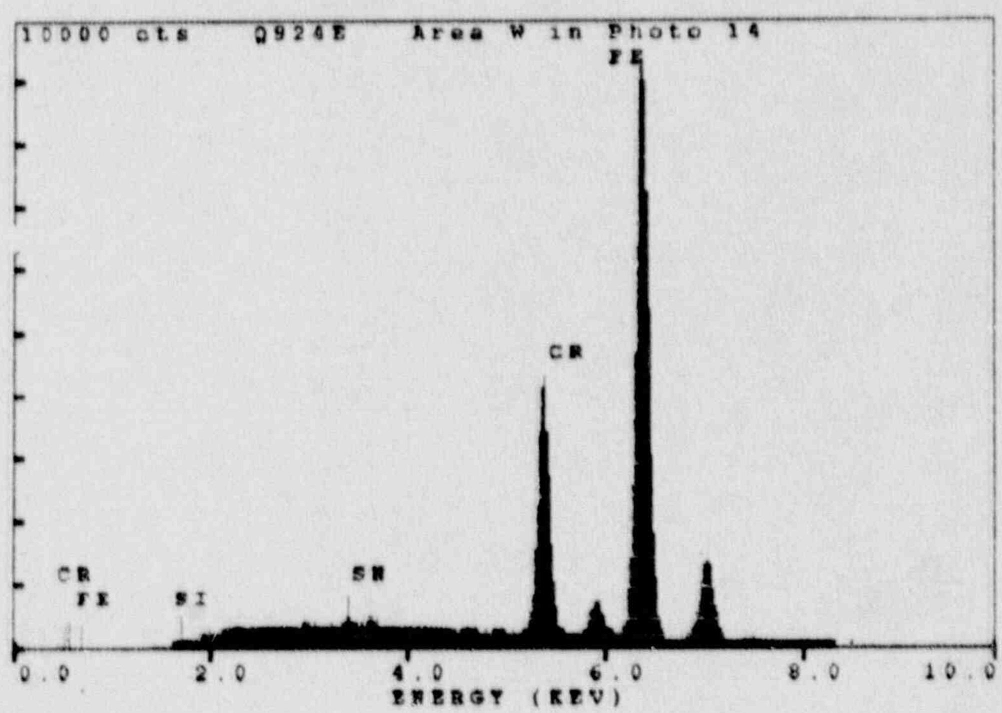
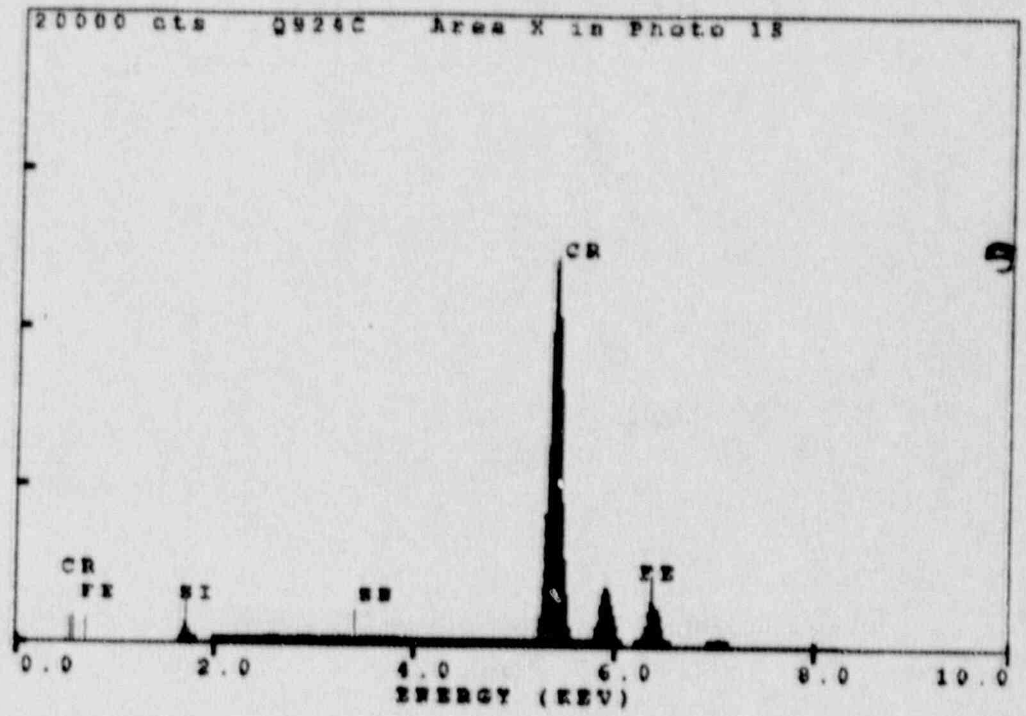


Figure B-41. EDS analysis of areas U and W shown in figure B-40.



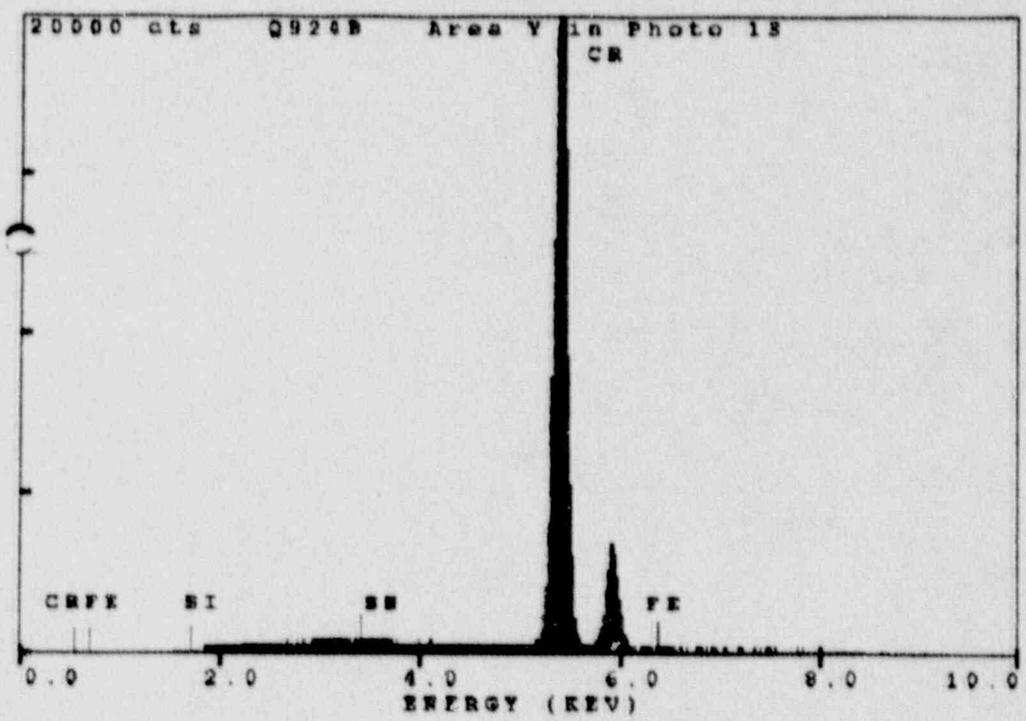


Figure B-42. EDS analysis of areas x and y shown in figure B-41.

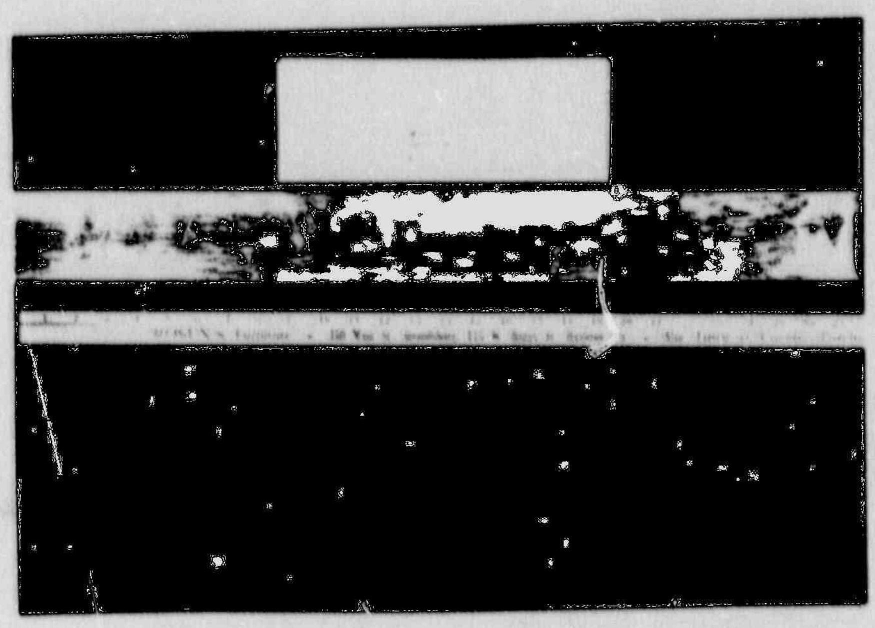


Figure B-43. General surface condition of the non-thrust side of the B diesel, 7L liner.

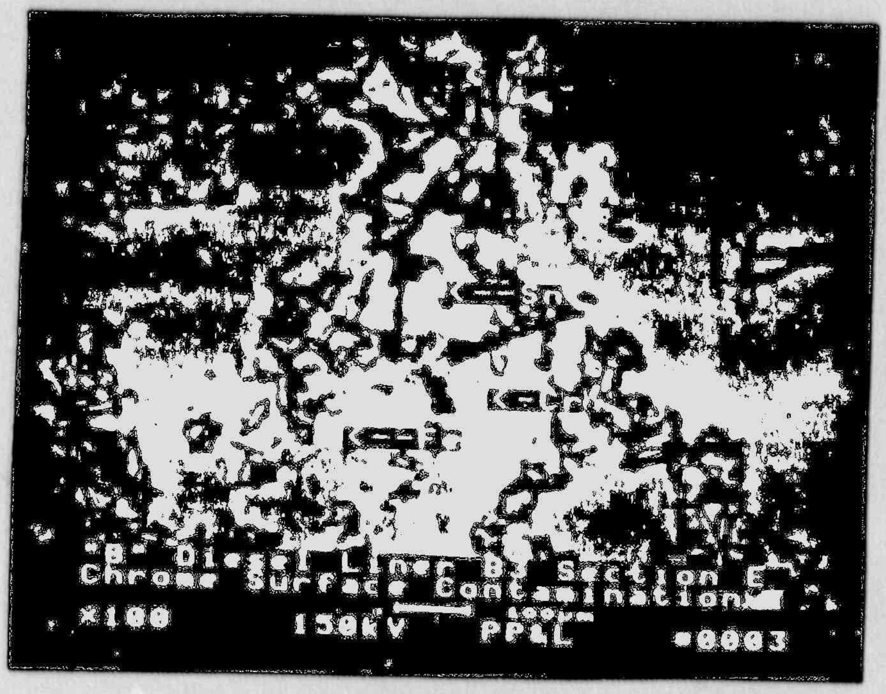


Figure B-44. Magnified image of the surface condition of the non-thrust side of the B diesel 7L liner near the upper position of the top compression ring. Both iron and tin are seen embedded in the chromium liner pores.

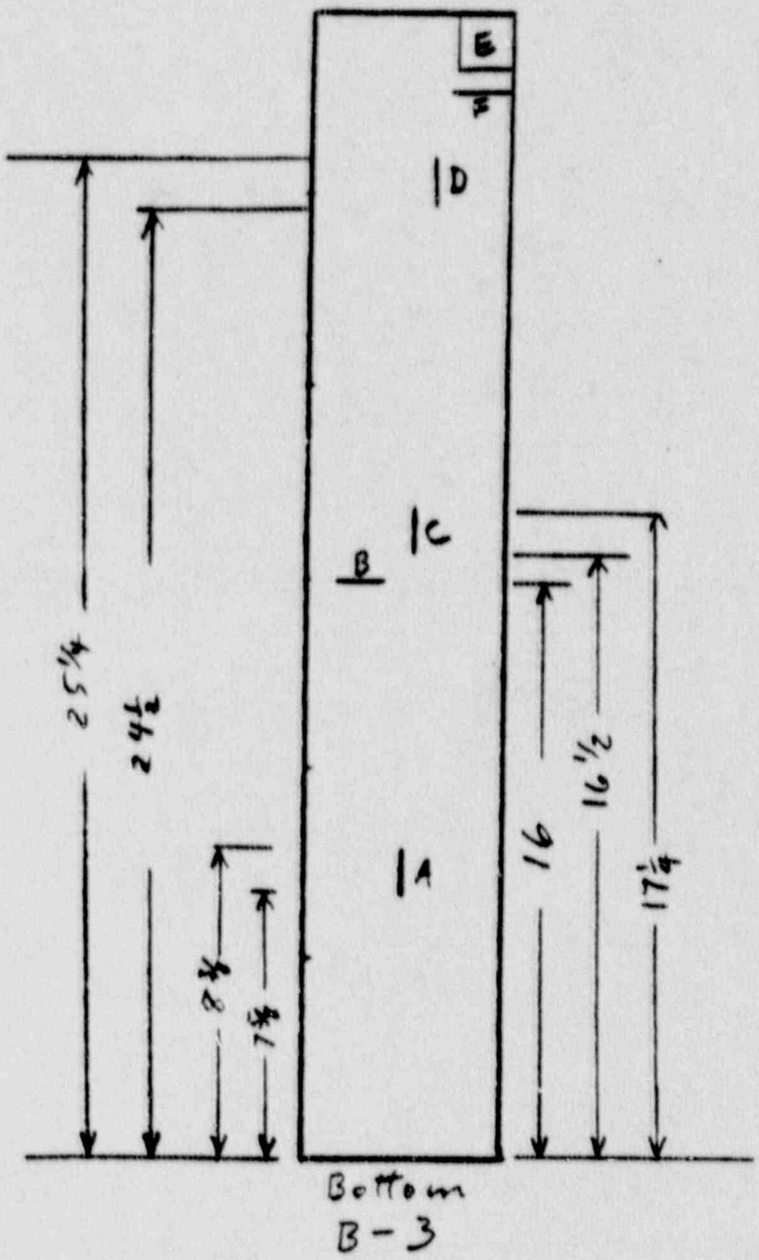


Figure B-45. Sketch of the non-thrust side of the B diesel 7L liner(designated B-3), showing the location and orientation of the metallurgical sections A, B, C, D, E, and F. Section E was for surface examinations, while the others were for cross-sections.

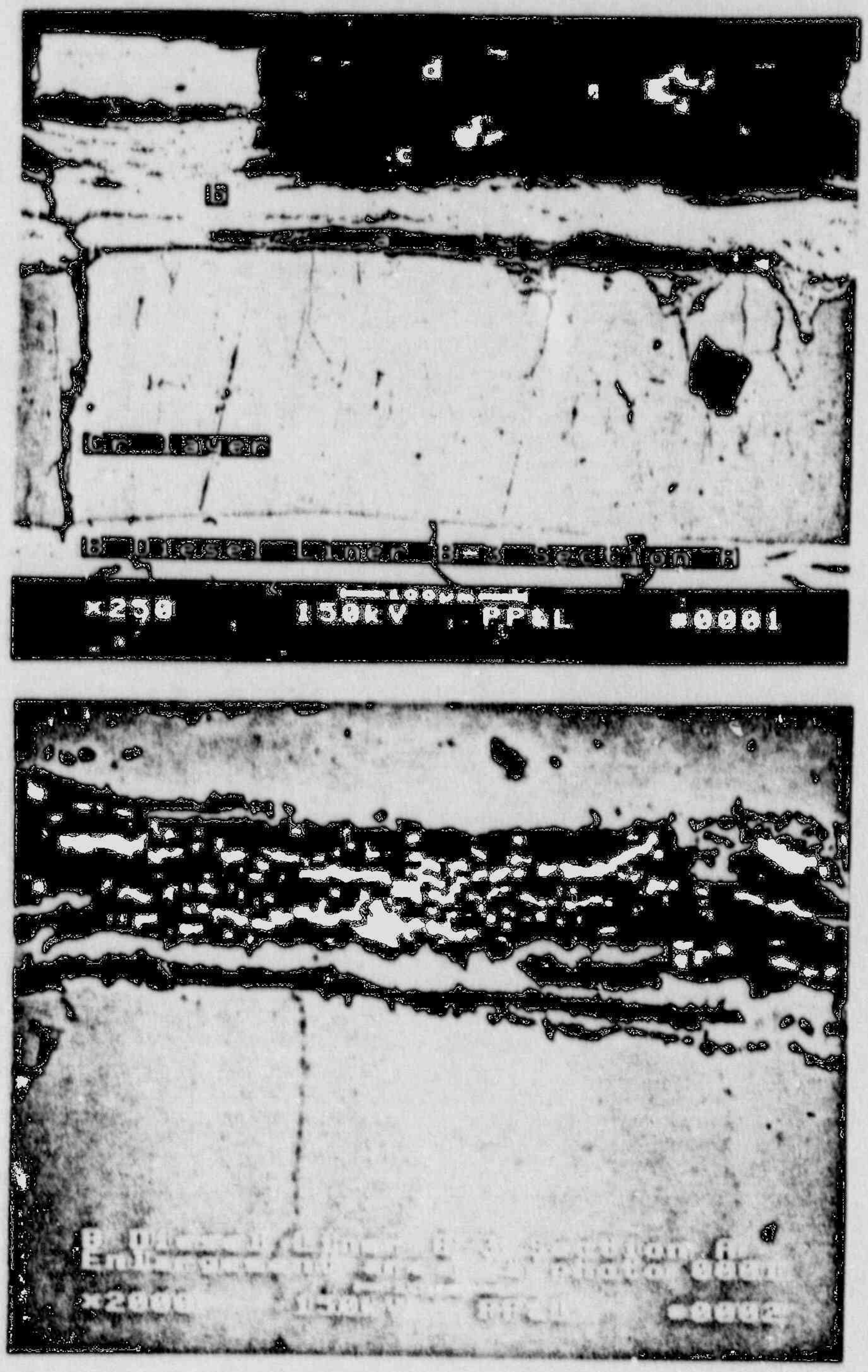
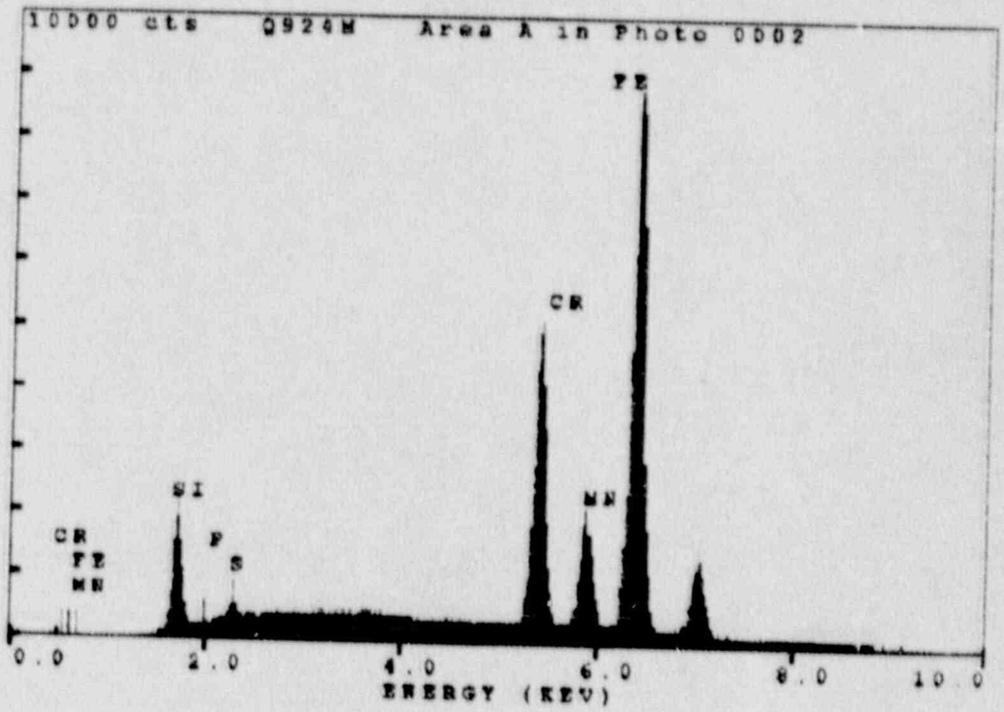


Figure B-46. Longitudinal section of the B diesel 7 L liner taken from an area 7 5/8 to 8 3/8 inches from the bottom. EDS analysis results are shown in figures B-47 and B-48.



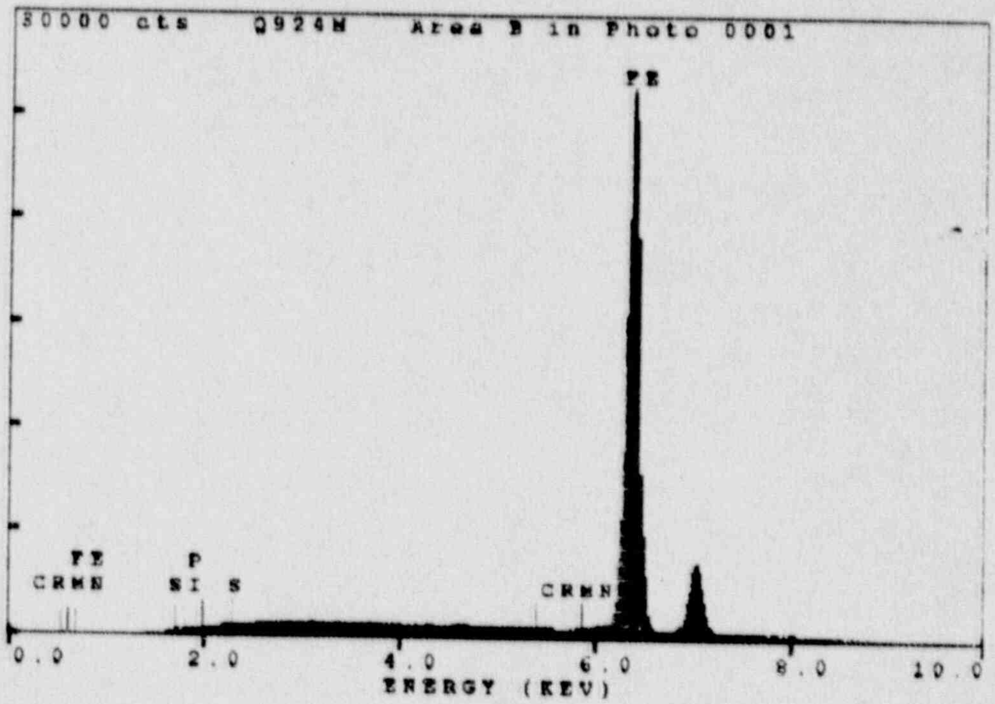
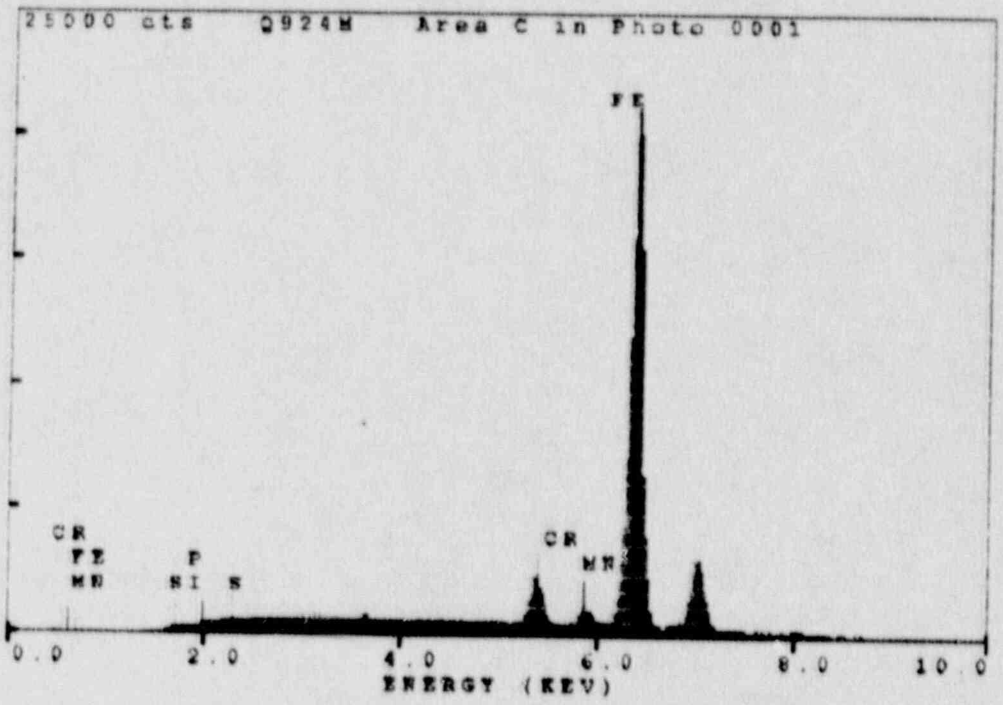


Figure B-47. EDS analysis of areas a and b in figure B-46.



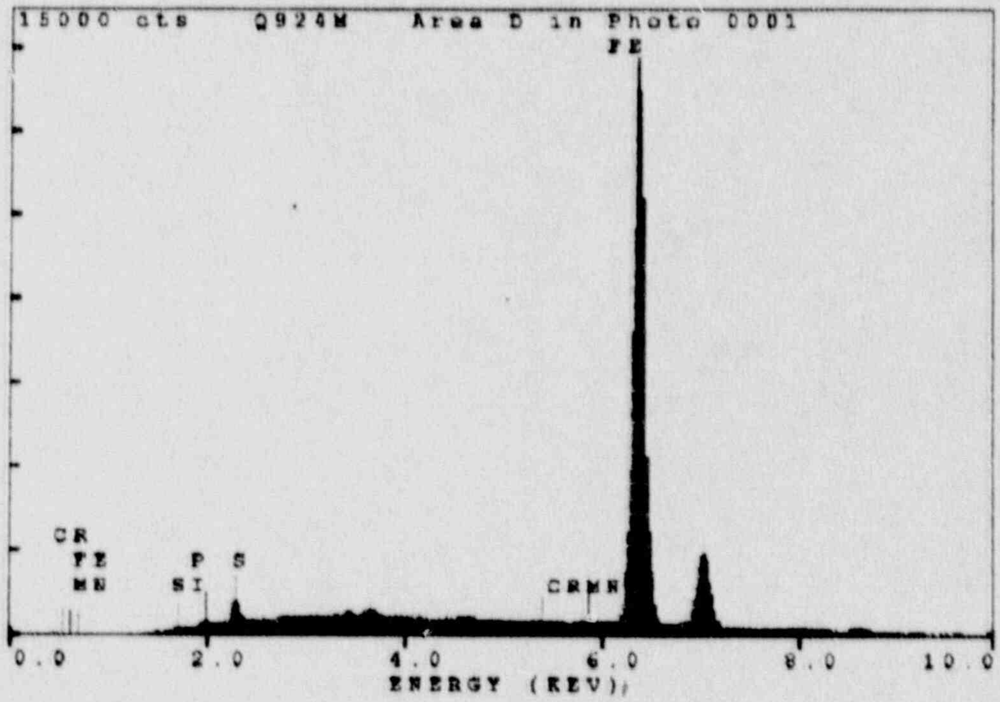


Figure B-48. EDS analysis of areas c and d in figure B-46.

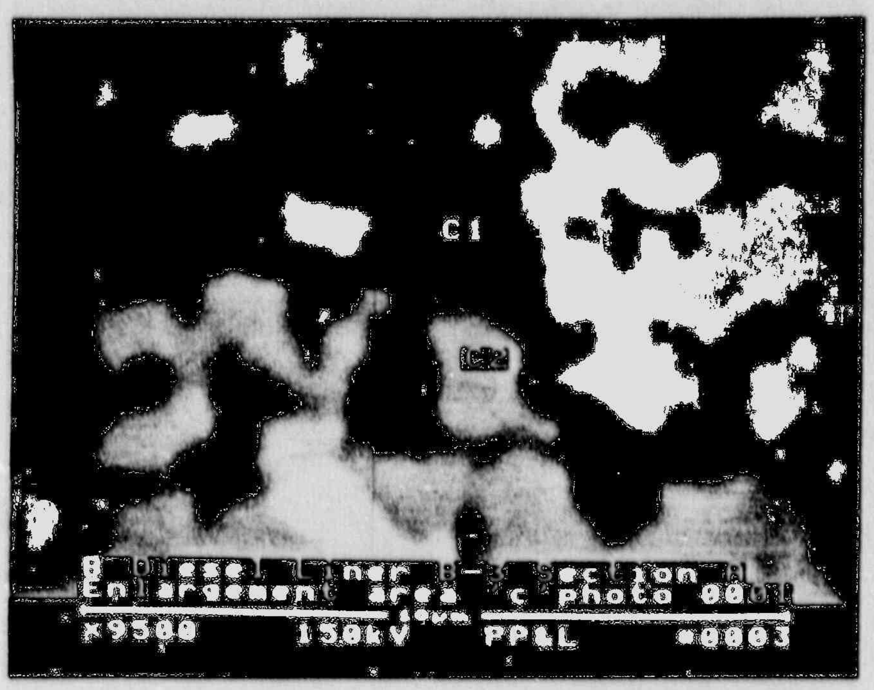


Figure B-49. Enlargement of area c in figure B-46 showing light and dark phases present. EDS analysis of these areas are shown in figure B-50.

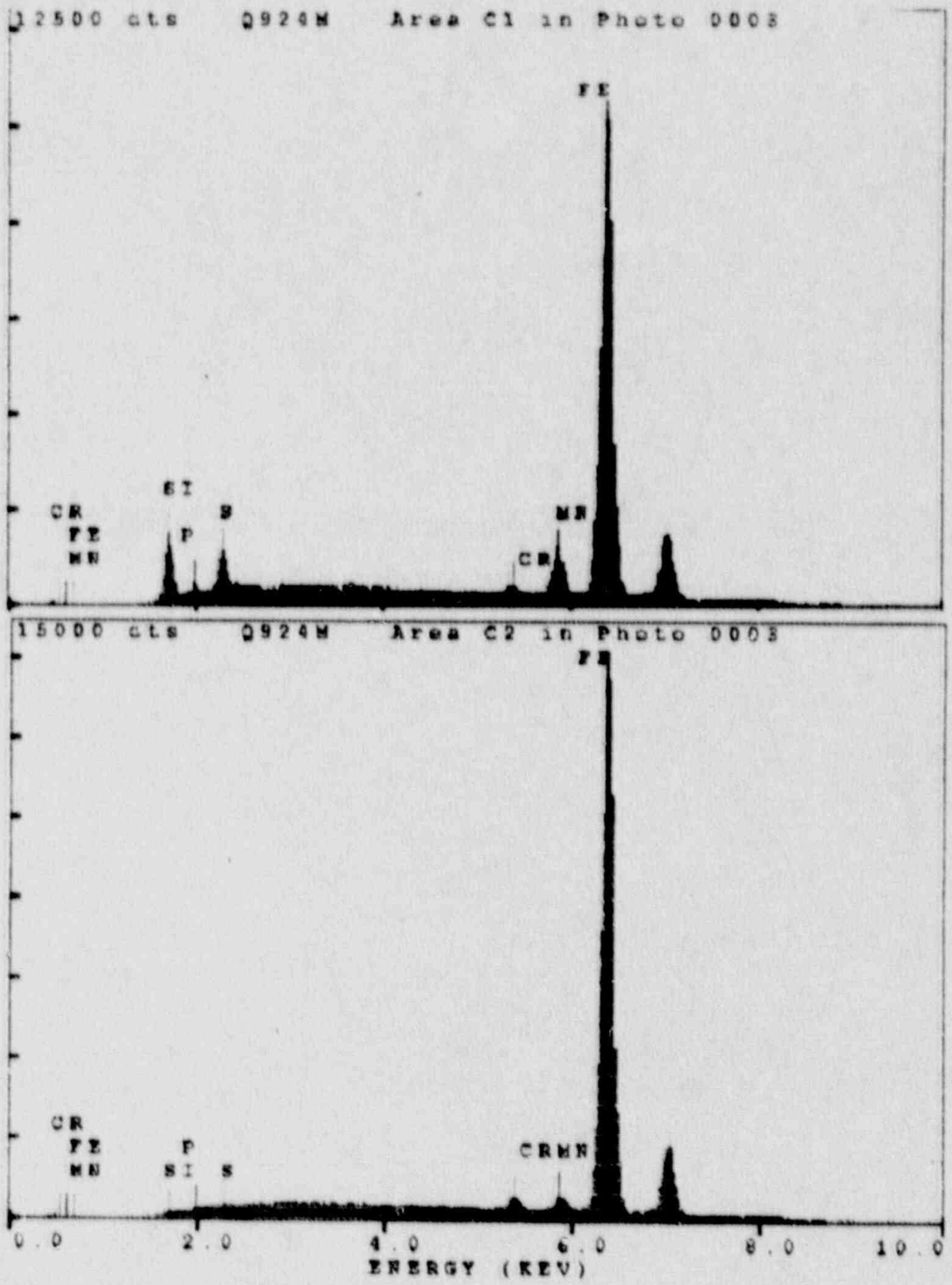


Figure B-50. Analysis of the pore from figure B-46 center right, showing that it contains mainly iron with only a trace of the elements Cr, S and Mn.

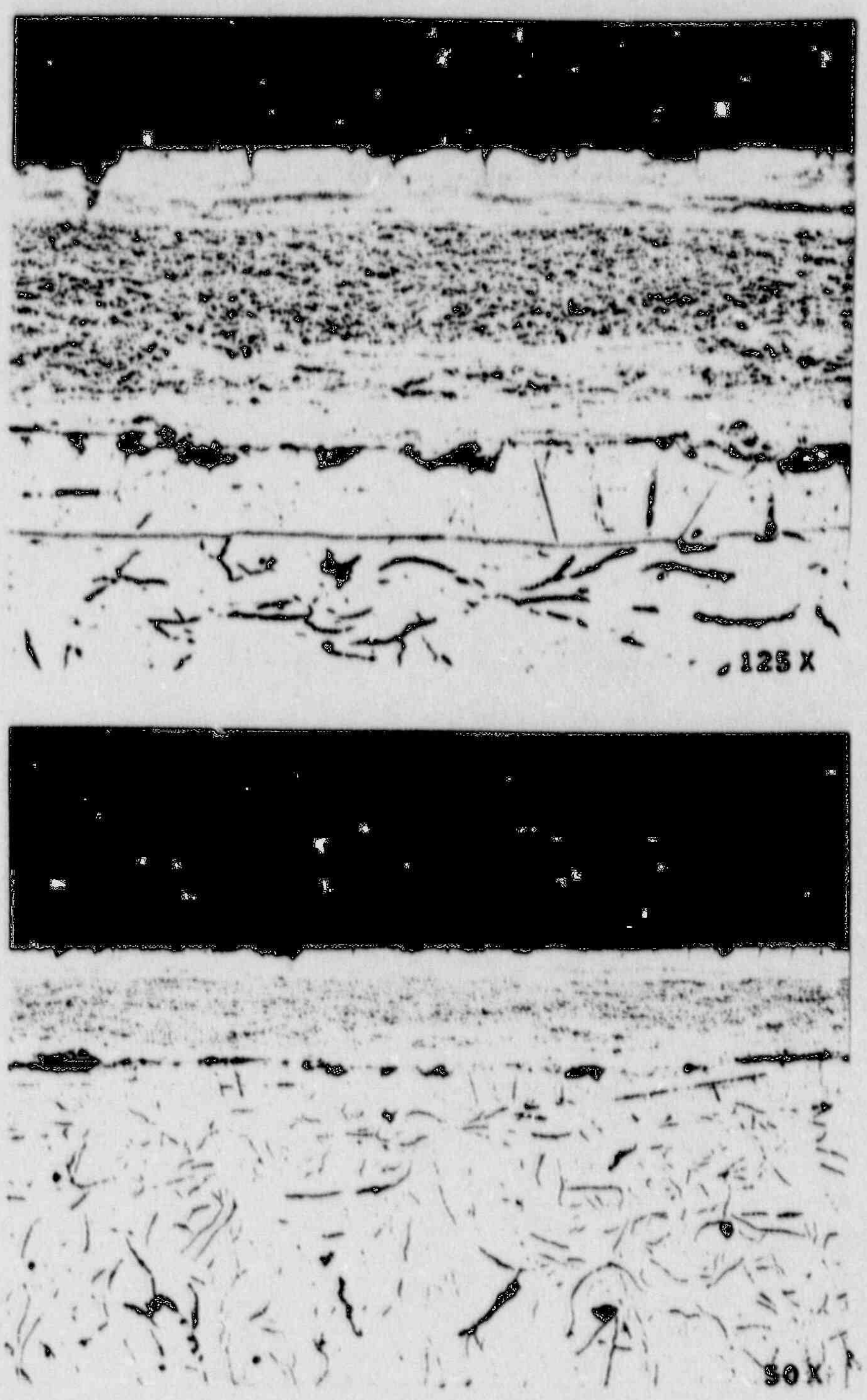


Figure B-51. Longitudinal cross-section of the B diesel 7L liner taken 7 5/8 inches from the bottom showing a very thick layer of metallic deposits covering the chromium plating.

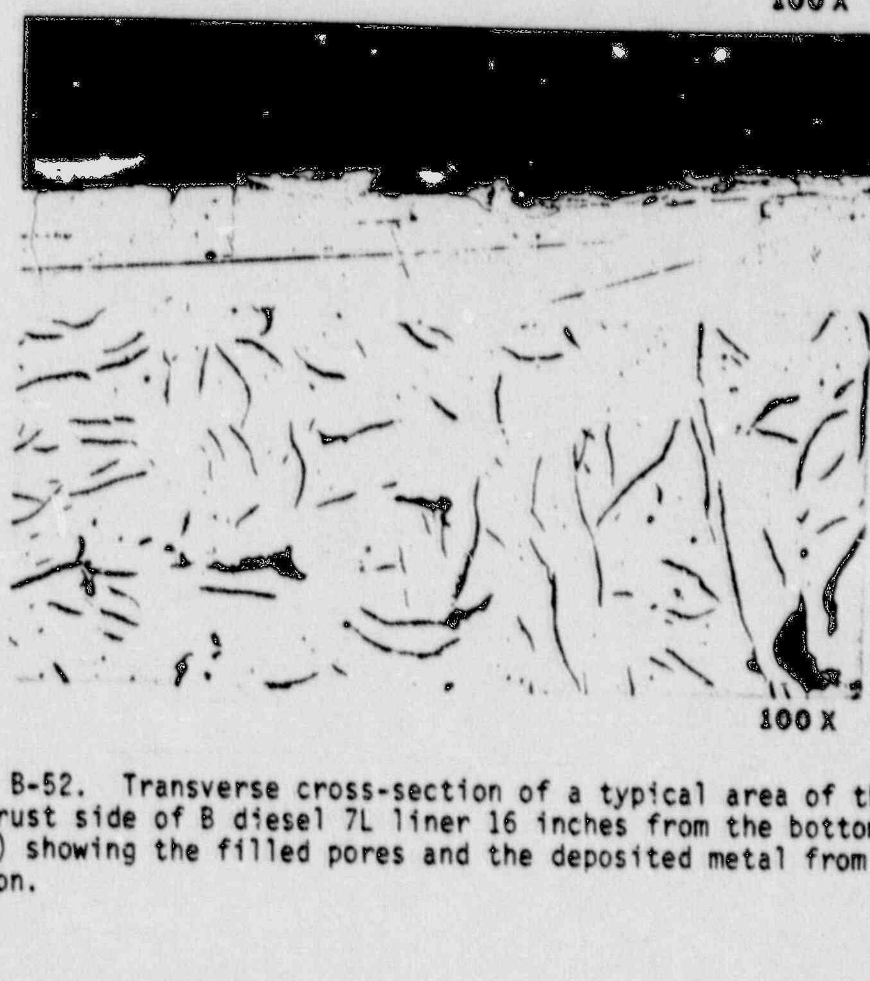
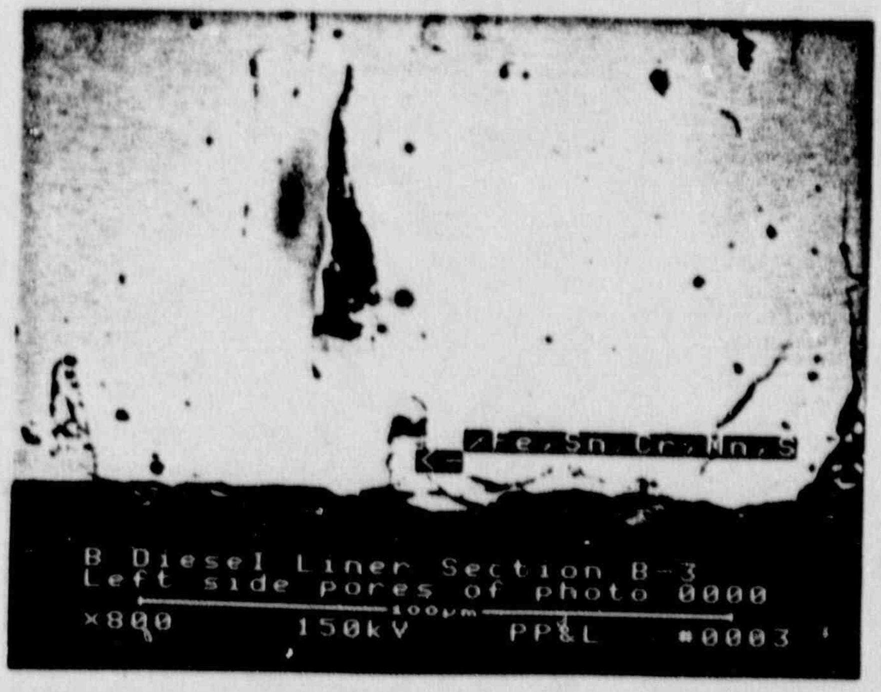


Figure B-52. Transverse cross-section of a typical area of the non-thrust side of B diesel 7L liner 16 inches from the bottom (B-3 area B) showing the filled pores and the deposited metal from this location.

Figure B-53. Longitudinal cross-section of the non-thrust side of the B diesel 7L liner taken 16.5 inches from the bottom of the liner. Typical examples of the metal deposited from the piston and rings on the Cr plating.

Figure B-54. SEM micrographs of longitudinal cross-section 'C' part B-3 of the B diesel liner at a level 16.5 inches from the bottom. Elements shown were identified as being present in the pores indicated.



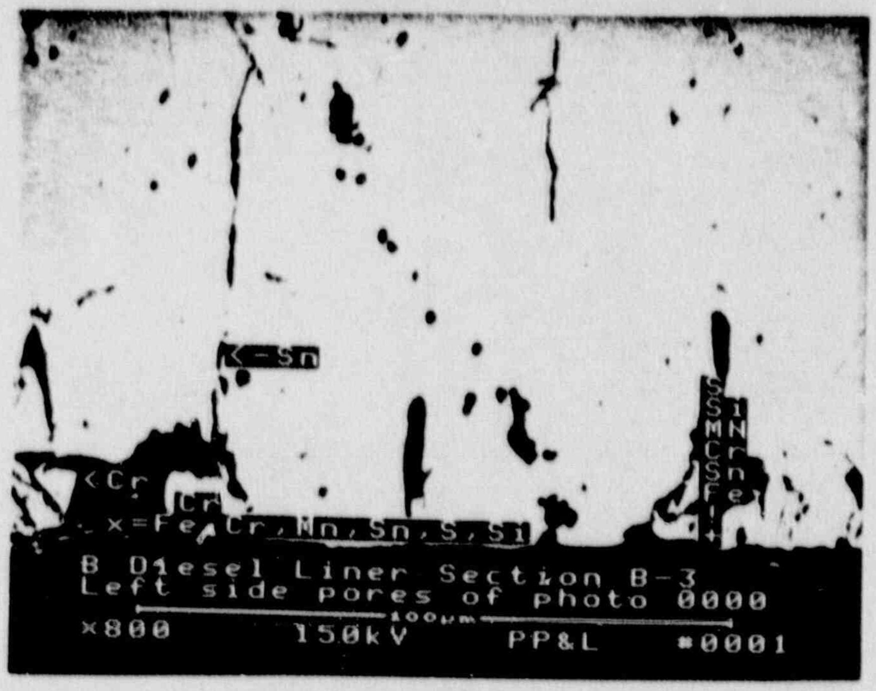
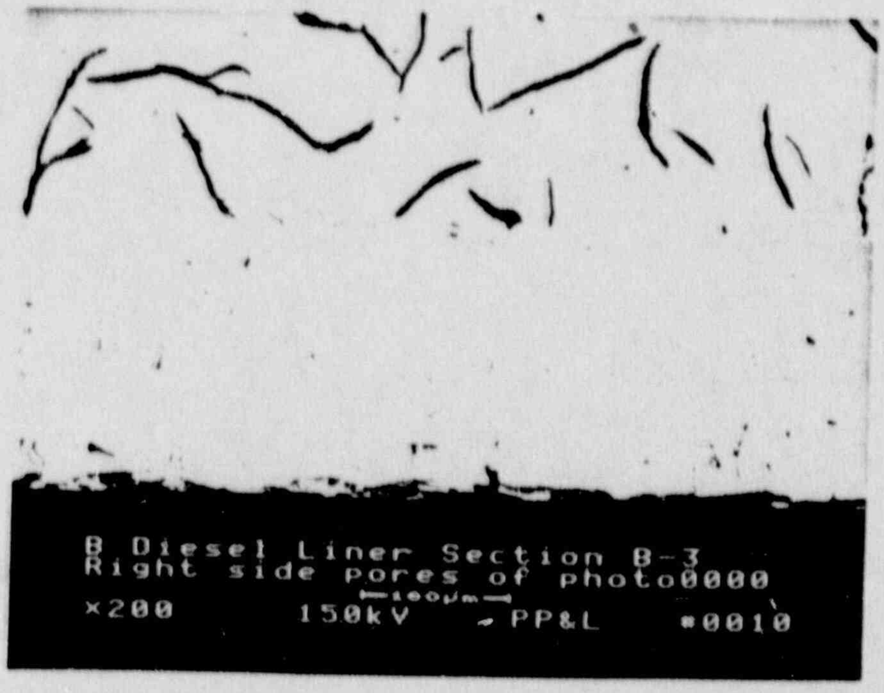


Figure B-55. Another area of pore deposits and elements found present in longitudinal cross-section 'C' part B-3 of the B diesel liner 16.5 inches from the bottom.



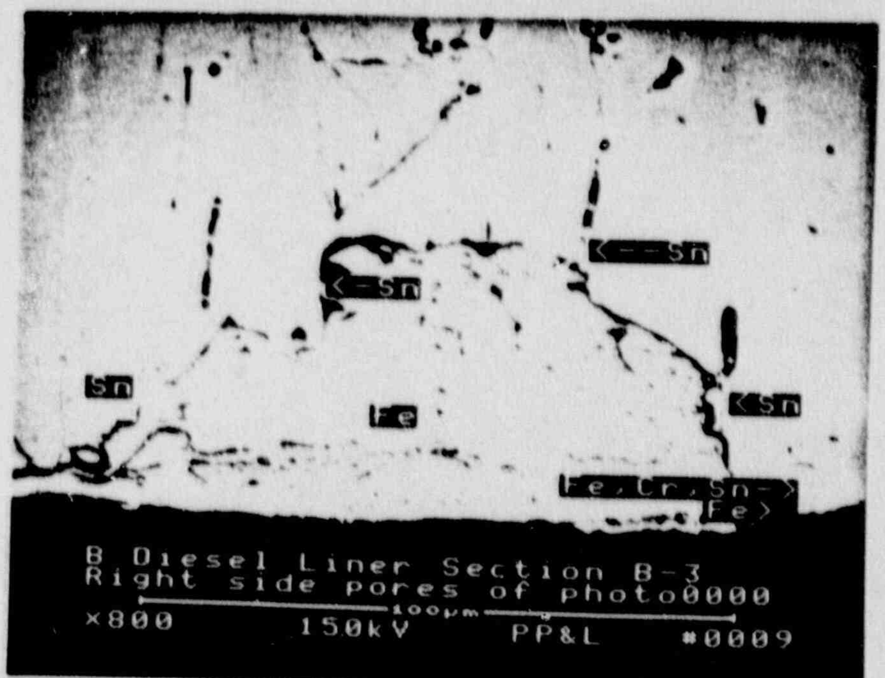


Figure B-56. Various elements and their location found in a pore from the B diesel liner 7L, part B-3, area 'C'.

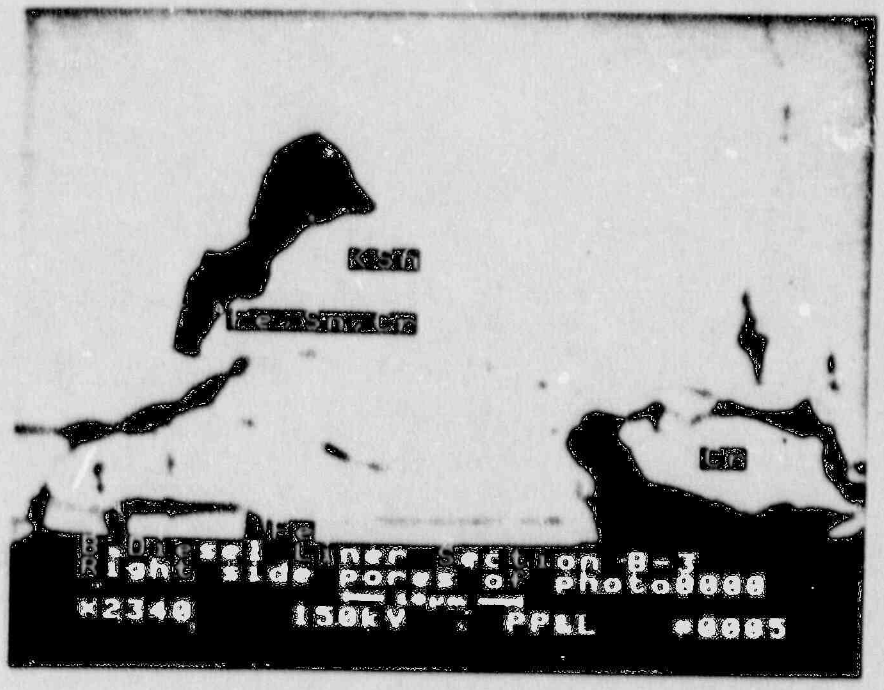


Figure B-57. Elemental contents of two pores filled with metal in the B diesel 7L liner 16.5 inches from the bottom of part B-3, area 'C'.

Figure B-58. Additional areas of filled pores found in area 'C' of part B-3 of the B diesel 7L cylinder.

V. METALLURGICAL ANALYSIS OF "C" DIESEL 5R CYLINDER FAILURE

1.0 Observations of the Failed Cylinder

1.1 The "C" diesel generator (D/G) had a crankcase explosion on October 7, 1989 during a 24-hour surveillance test. The 5R cylinder suffered major damage which ultimately caused the explosion in the crankcase. Also, the 6L cylinder had an area of "tinning" on the liner. The "C" D/G had accumulated approximately 950 hours of operation prior to the failure.

1.2 Visual Assessment After Failure

1.2.1 The 5R piston characteristics were similar to those of the 7L piston from the "B" D/G failure on September 16, 1989 - the blackened areas with scratched and smeared metal, worn rings, rings with caked debris, piston skirt cracking. The piston damage caused by the lack of lubrication and overheating is shown in photographs in Figures C-6, C-7 and C-8. The major differences were (1) the end caps were only slightly scratched, (2) the piston skirt contained two deep holes, and (3) the piston wrist pin and its bushing suffered major overheating (bluing). The bushing had broken free from its bore in the piston due to the contact between the pin and bushing caused by the pin expansion and loss of lubrication from the heating. From the heat, the bushing flowed into the shape of the oil grooves at the pin ends. There was so much contact between the pin and bushing that they had to be cut apart. The pin exhibited normal wear patterns at the interface between the pin and connecting rod, indicating no loose connections (Figure C-5a). Photos in Figures C-3, C-4 and C-5 show the damage to the pin and bushing.

1.2.2 The 6L cylinder liner contained an area of "tinning." Reference the photos of the 6L piston showing the area of detinning on the non-thrust side in Figure C-1. One possible cause of the detinning is the alignment of the compression ring gaps in a row allowing "blow-by" to melt the tin surface of the piston causing tin to be smeared onto the chrome plating of the cylinder liner. Alternatively, the absence of lubrication in this area may allow tin transfer by cold welding of the piston tin on the liner surface.

2.0 Metallurgical Investigation Findings on Specific Parts

2.1 Piston Pin End Caps - The material of the end caps is cast iron. Analysis of the cap material is in Appendix I. The design of the end caps was an intermediate design of the lands in the back of the cap - recessed more than the original but less than the new design. The outside surfaces showed minor scoring. See photos of this in Figure C-2. Both caps exhibited much discoloration on the outer surfaces associated with the overheating of the piston.

2.2 Pin Bushing - The material of the bushing is a copper alloy, C93700. There was a large through-wall crack at one end of the bushing (Figure C-3a) caused by the expansion of the pin due to the overheating. There was scoring on the outside surface of the bushing from the set screws because the bushing was forced to rotate in the piston by the seizing forces between the pin and bushing (Figure C-3b).

2.3 Piston Pin - The material is an SAE 5046 low alloy steel. The pins can be manufactured from one of the following SAE specifications: 1050, 5046, 4145 or 8620. The pin is case hardened to a minimum depth of 0.060". Hardness readings in the "blued" areas were 20 Rc as opposed to the normal hardness of 50 Rc. There was a concave bend in the pin of 0.002" between the bolt holes for the connecting rod. This could have resulted from compressive stress overload after the pin reached temperatures in excess of 1200°F during the failure process. Above 1200°F, the yield stress for this material decreases dramatically.

2.4 Piston - The material of the piston is cast iron (reference the analysis taken in Appendix I). The piston skirt, non-thrust side toward the end cap facing the auxiliary skid end of the engine, had two holes and various cracks in it (Figure C-8). The skirt was cut from the piston and a cross-section was taken from the area surrounding one hole, "A," to be metallurgically analyzed (Figure C-9 and C-10). The results indicated that the holes were inclusions from the casting process. The grain structure, shown in Figures C-11a, b, c, d, e and f, from the hole and surrounding area contained the typical mix of phases for a cast iron material heated to temperatures above 1350°F (course martensite, fine perlite and a mix of both). The analysis of the area in the Scanning Electron Microscope (SEM), using the Energy Dispersive Spectra (EDS), showed that (a) an inclusion in a lap "W" just below the hole contained iron, manganese, silicon (SiO_2), potassium, aluminum, calcium compounds (Figure C-12). The hole surface contained a lead-tin eutectic (Figure C-13). The SEM/EDS analysis performed on the surface area around the hole showed that tin, chrome and iron elements were on the surface in the form of smeared metal (Figures C-16, C-17).

2.5 Compression/Oil Rings - The material of the rings is cast iron (reference Appendix I for detailed analysis). All rings from the 5R piston were sent to Cooper Bessemer except the lower oil ring remaining in the piston skirt. A section was taken from this ring and mounted. Figure C-21 is a microphotograph of the lower oil ring section showing the wear and deformation on the ring. There was at least 0.010" removed - the dimension to the outer surface of the ring is specified as between 0.065 and 0.075". The dimensions of the compression rings were taken by PP&L and are in Appendix III. All the rings were within the dimensions expected.

2.6 Cylinder Liner - The material of the liner is a cast iron with a 0.006" chrome plating on the ID, which has a porous surface placed by a reverse plating process. The liner displayed a tartan appearance on the surface due to overheating/scoring. The liner was cut, for sectioning, longitudinally along the thrust, non-thrust and one end cap sides. Sections were then taken from these lengths (refer to Figure C-15). Only Section C-3B, taken from the top of the non-thrust side 3 1/2" down, has been analyzed. The surface of the liner displayed the chrome pores as containing iron, tin and some chrome particles (refer to Figures C-16 and 17). Figure 18 is a microphotograph of the SEM/EDS analysis of one pore in chrome liner Section C-3B. This shows the presence of iron and tin particles in the pore. Additional analysis is to be performed at a later date to determine how the various materials were dispersed and show the damage generated by the failure process.

2.7 Deposits/Debris - Deposits from the top of the piston were analyzed on the SEM and EDS. The results showed that the non-magnetic particles were elements from the fuel oil - calcium, sulfur, phosphorous and zinc (refer to Figure C-19a, b, c and d) and the magnetic particles were mostly iron with small amount of manganese and calcium and sulfur again from fuel oil (refer to Figure C-20a and b).

2.8 Lubricating Oil - The analysis of the lubricating oil for the months of July, August and September 1989 for viscosity, flash point, acidity, moisture, carbon residue, ash, ferrography and metals. All aspects of the analyses were deemed to be acceptable. Refer to Appendix II for the reports.

3.0 Discussion of C-Diesel, 5R Failure

3.1 This cylinder showed much physical damage, overheating and generation of debris. The lower oil ring showed heavy wear and deformation on the sliding surfaces.

3.2 The piston skirt contained two holes in the non-thrust side. These holes were created as part of the casting process for the piston. The presence of a lead/tin eutectic indicated that this area was repaired with solder during the manufacturing process. Cooper's QC records indicated that solder was used to fill the voids in the piston skirt. Also this portion of the liner extends beyond the bottom of the skirt at the end of the downstroke and any debris from this area would eject into the sump. Thus, these holes in the piston skirt had no impact on the failure.

3.3 The piston skirt suffered heavy scuffing all the way around indicating overheating of the piston until the the normal clearance between the piston and the liner had been filled. With a clearance of about 0.014" between the OD of the piston and the ID of the liner, a thermal expansion coefficient of 7.6E-06°F and a diameter

of 13.5"; a temperature difference of only 136°F is needed to close the gap. Since tin, which melts at 350°F, had melted off the piston surface during the failure, it is reasonable to expect a temperature difference of this magnitude had existed at the time of failure.

3.4 Pin/Bushing Lubrication and Overheating

3.4.1 The pin and bushing of the 5R piston suffered overheating and damage due to contact with each other. The lubricating oil is applied to the interface between the pin and bushing by gravity and inertial forces. The oil is pressure fed through the connecting rod and pin and sprayed onto the upper cavity of the piston above the pin. Oil that sprays up to the head falls and forms a pool of oil that covers the exposed portion of the pin and bushing. A groove in the top of the pin allows oil to flow along the length of the pin and through circumferentially oriented grooves near the ends of the pin. Rotation of the pin in the bushing is a back and forth motion which distributes the oil from the grooves over the mating surfaces. The items that could affect the lubrication of the mating surface includes:

3.4.1.1 Oil Viscosity - Oil properties have met the specifications to which they were tested.

3.4.1.2 Oil Foaming Characteristics - This quality has not been a normal test for our engine oil. However, recent tests showed no abnormal oil foaming characteristics.

3.4.1.3 Clearances Between the Pin and Bushing - Fit-up or "blue checking" of these parts has not been performed to manufacturer's specifications until recently.

3.4.1.4 Distortion of the Pin or Bushing - Several piston pins have shown bending up to 0.005" after years of operation. All of these pins except one have displayed overheating indications. The exception is one pin from the "D" diesel. This pin shows no evidence of overheating. This is being investigated further.

3.4.1.5 Debris Causing Friction Between the Pin and Bushing Surfaces - Scratching has been found on occasion from debris, but no relationship between the debris and the overheating could be found.

3.4.1.6 Debris Filling the Grooves or Oil Hole in the Pin - We have never found debris or deposits in the grooves or oil hole of the pin.

3.4.1.7 Air Pockets Preventing Oil to Flow Through the Grooves in the Pin - Since the pin/bushing lubrication is only by gravity feed and not force fed by a pump, air bubble(s) could block oil from flowing along the pin grooves. Once starved of oil, high friction areas at raised parts of the pin/bushing could cause local overheating. At the boiling point of oil, gas volume would increase dramatically and prevent any further ingress of oil into the area.

3.5 The cylinder liner is key to the lubrication of the piston rings. The liner has a porous chrome plating that retains an oil film on its surface in the pores. Oil is smeared over the cylinder wall via a splash mechanism which deposits oil on the lower portion of the liner. As the piston goes up and down, the lower oil ring spreads the oil film uniformly over the surface and removes the excess. As the piston completes many cycles, the oil film is gradually spread over the entire length of the cylinder wall to the uppermost compression ring. The oil i[illegible]etained in the pores of the liner, which is their purpose. I[illegible]se pores no longer retain oil, then the upper wall of the lir[illegible]l gradually be starved of oil, friction will increase an[illegible]t will be generated. The liner pores can be filled by debris [illegible]he combustion and oil decomposition products, ring materia[illegible] transferred from the piston surface. Evidence of this hap[illegible]ring operation was found during this investigation and wi[illegible]art of our inspection program in the future.

4.0 Conclusions

4.1 Heavy damage was found on the cylinder liner surface, piston, compression rings, oil rings, pin and pin bushing of the "C" diesel 5R cylinder. Since the piston end caps suffered only slight scoring, it can be concluded that the damage was due to the failure, not a cause like the end caps in the "B" diesel.

4.2 Tin and iron were the elements found in the liner pores analyzed to date. They indicate wear products were generated from the rings and piston surface and were deposited there early in the failure process.

4.3 The piston pin and the bushing sustained major damage from overheating due to the pin expansion causing contact with the bushing and ultimately freezing of the pin in the bushing. This type of failure of the pin/bushing would create a malfunctioning of the motion of the piston initiating a failure process.

4.4 The following sequence of events appears to provide the most logical failure scenario:

A. Friction between the pin and bushing caused them to overheat and bind together.

B. The piston motion became unstable causing contact against the liner wall particularly on the non-thrust side of the liner. Oil would be scraped off during the upward motion of the piston.

C. This contact caused particles from the piston surface and rings to fill the liner pores and increase the friction and wear in the cylinder.

D. Wear products from all parts mix and pass between the wearing surfaces as the failure progresses.

E. Eventually localized heating becomes so great in some areas that the oil cannot cool fast enough to prevent oil decomposition and oxidation of the heated parts.

F. The heat continues to build until the piston expands to fill the clearance between the itself and the liner.

G. Once the clearance closes, frictional heat raises the temperature of the parts to "red heat" raising the oil vapor and air in the region to their ignition and a crankcase overpressurization occurs.

CHAPTER V

5.0 Figures

5.1 Figure Captions

5.2 Figures C-1 to C-21

Figure C-1. Photograph of the 6L piston from the "C" diesel showing the area of detinning and the alignment of the compression rings.

Figure C-2. Photographs of the 5R end caps, "C" diesel, showing the discoloration and minor scoring.

Figure C-3. (a) Photograph of the 5R piston, "C" diesel, pin bushing through wall crack; (b) OD damage caused by the bushing rotating.

Figure C-4. Photograph discoloration (bluing) of the 5R, "C" diesel, pin bushing.

Figure C-5. (a) through (d) Photographs of the discoloration (bluing at various positions around the pin from the 5R piston, "C" diesel.

Figure C-6. Photographs of the damage on the thrust side (center of engine) of the 5R piston, "C" diesel.

Figure C-7. (a) Photographs of the 5R piston end cap facing the generator end of the "C" diesel; (b) the opposite end cap facing the aux. skid end of diesel.

Figure C-8. Photographs of the damage on the non-thrust side of the 5R piston, "C" diesel; note the two holes in the skirt at the bottom center of the photo on the right.

Figure C-9. The two holes and crack in the 5R piston skirt, non-thrust die. Hole "A" was sectioned and metallurgically examined.

Figure C-10. A microphotograph of the cross-section of the 5R piston skirt Hole "A" in Figure C-9. The letters designate the areas evaluated metallurgically.

Figure C-11. A microphotograph of typical areas around Hole "A" in the 5R piston skirt. See Figure C-10 for locations, (a) area "K-M" at 800C showing a course martensitic structure; (b) area "K-J" at 50X showing course martensitic and smear metal layer of iron chrome and tin on surface of Hole "A" (c) area "H-G" at 50X showing transition from a martensitic to a perlite structure; (d) area "C-B" at 800X showing course martensitic structure; (e) area "B-Q" at 800X showing perlite structure around graphite stringers; (f) area "A-B" at 50X showing a mixed structure of fine perlite and martensite.

Figure C-12. SEM/EDS analysis of inclusion "W" below Hole "A" in 5R piston skirt (a) SEM photo at 6.5X showing inclusion "W" (b) spectrum of the elements in the inclusion "W."

Figure C-13. SEM/EDS analysis of the Lead-Tin Eutectic on the surface of Hole "A" in 5R piston skirts (a) SEM photo at 1300X showing the Pb-Sn Eutectic on surface of hole; (b) spectrum of the elements in the light area in (a) above.

Figure C-14. SEM/EDS analysis of smeared metal layer on surface of 5R piston skirt at area "J" (see Figure C-10) photo at 4660X of smeared metal layer; (b) spectrum of LIGHTER AREA in photo in C-14a above; (c) spectrum of darker area in photo in C-14a.

Figure C-15. Photos of 5R liner Sections C-1, C-2 and C-3; (a) liner Section C-1 longitudinal section taken from thrust side; (b) Liner Section C-2 longitudinal section taken from end cap side; (c) Liner Section C-3, longitudinal section taken from the non-thrust side.

Figure C-16. Sketch of sections taken from liner pieces.

Figure C-17. SEM/EDS analysis of the debris in the pores in liner Section C-3B (see Figure C-16); (a) SEM photo at 100X of Section C-3 denotes area "A," "B," and "C" for analysis on EDS; (b) spectrum of area "A" showing the presence of mostly tin and iron; (c) spectrum of area "B" showing mostly tin and iron with a small peak of chrome; (d) spectrum of area "C" showing mostly iron and tin with a small peak of chrome.

Figure C-18. Microphoto from SEM 1010X showing the iron and tin particles in a pore in the chrome layer on liner Section C-3B.

Figure C-19. SEM/EDS analysis of non-magnetic particles taken from the top of the 5R piston, "C" diesel; (a) SEM photo of 43.4X of non-magnetic particles; (b) Spectrum of area in the SEM photo.

Figure C-20. SEM/EDS analysis of magnetic particles taken from the top of the 5R piston, "C" diesel, (a) SEM photo at 366X of magnetic particles; (b) spectrum of area in SEM photo.

Figure C-21. Microphotograph at 50X of cross-section of the lower oil ring from the 5R piston, "C" diesel. The deformed structure where the oil ring was worn can be seen at area "A."

Figure C-1. Photograph of the 6L piston from the C diesel showing the area of detining and the alignment of the compression rings.

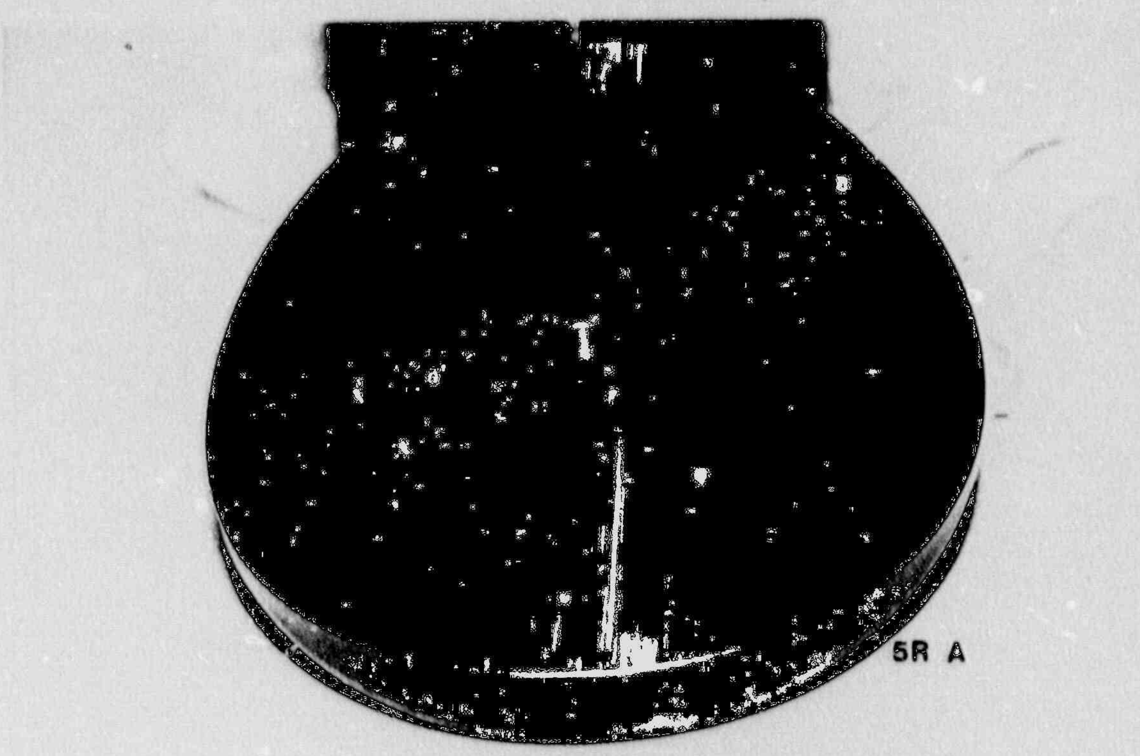


Figure C-2. Photographs of the 5R end caps, C diesel, showing the discoloration and minor scoring.

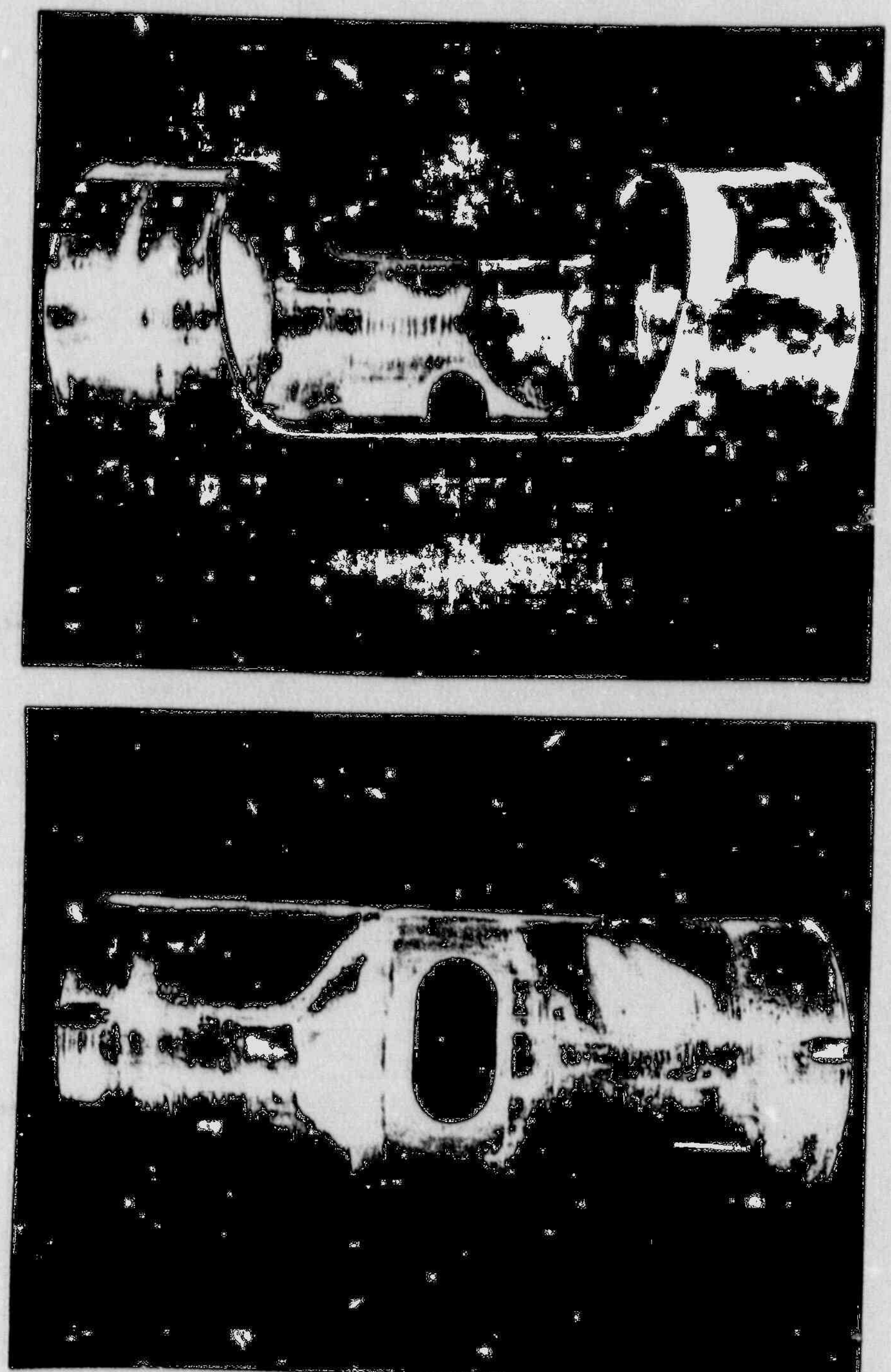


Figure C-3. a) Photograph of the 5R piston, C diesel, pin bushing through wall crack; b) OD damage caused by the bushing rotating.

Figure C-4. Photograph discoloration (bluing) of the 5R, C diesel, pin bushing.

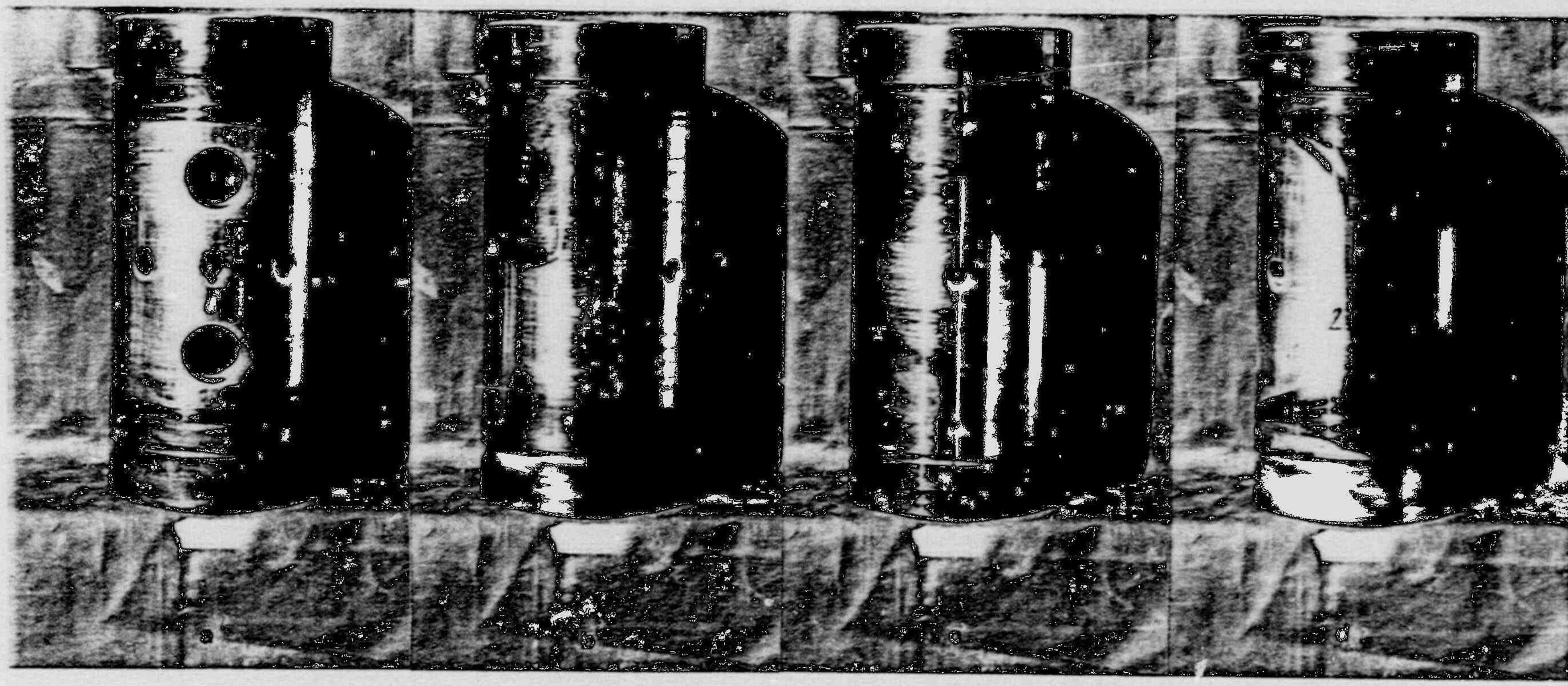


Figure C-5. a) thru d) Photographs of the discoloration (bluing at various positions around the pin from the 5R piston, C diesel.

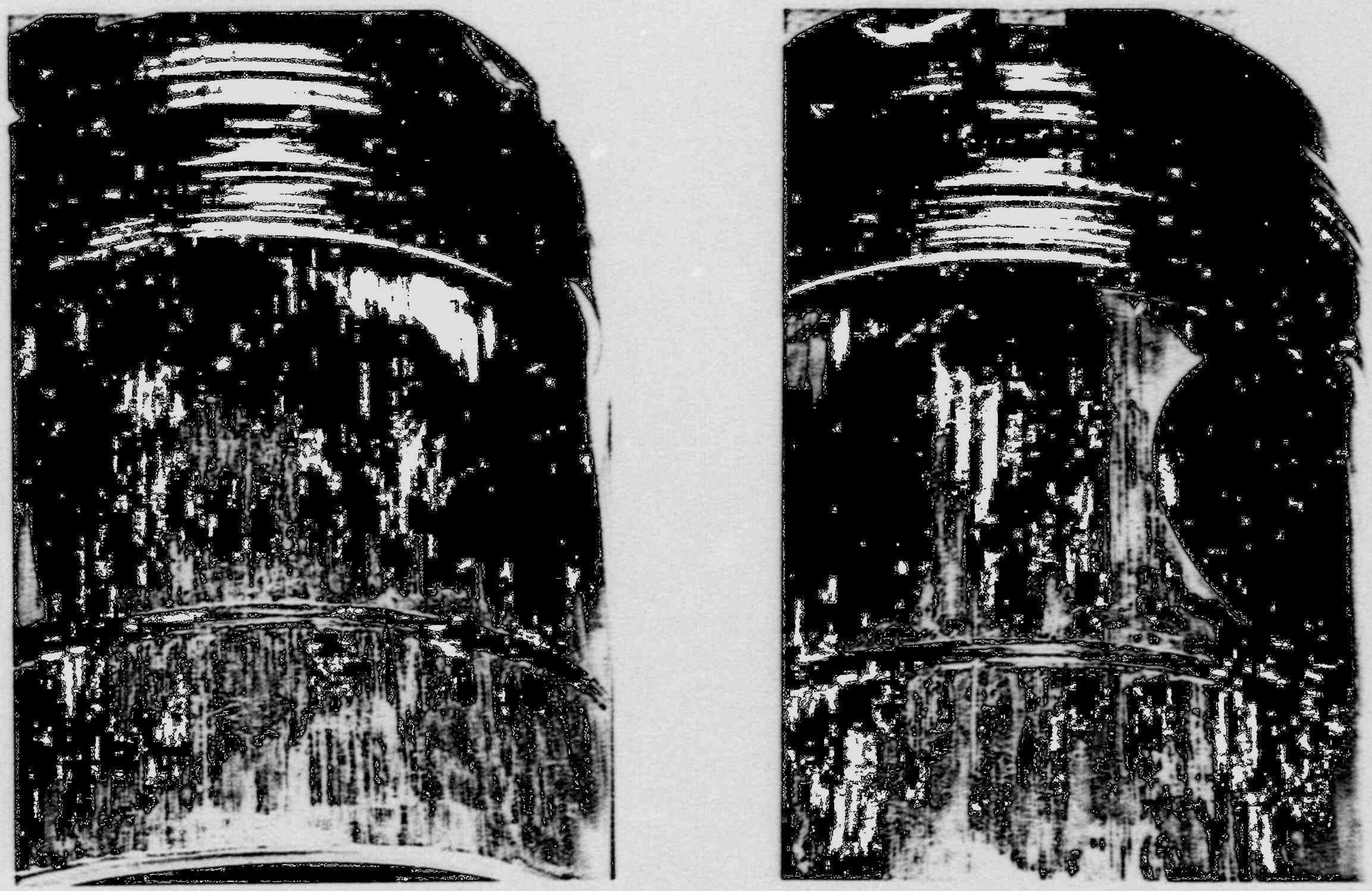
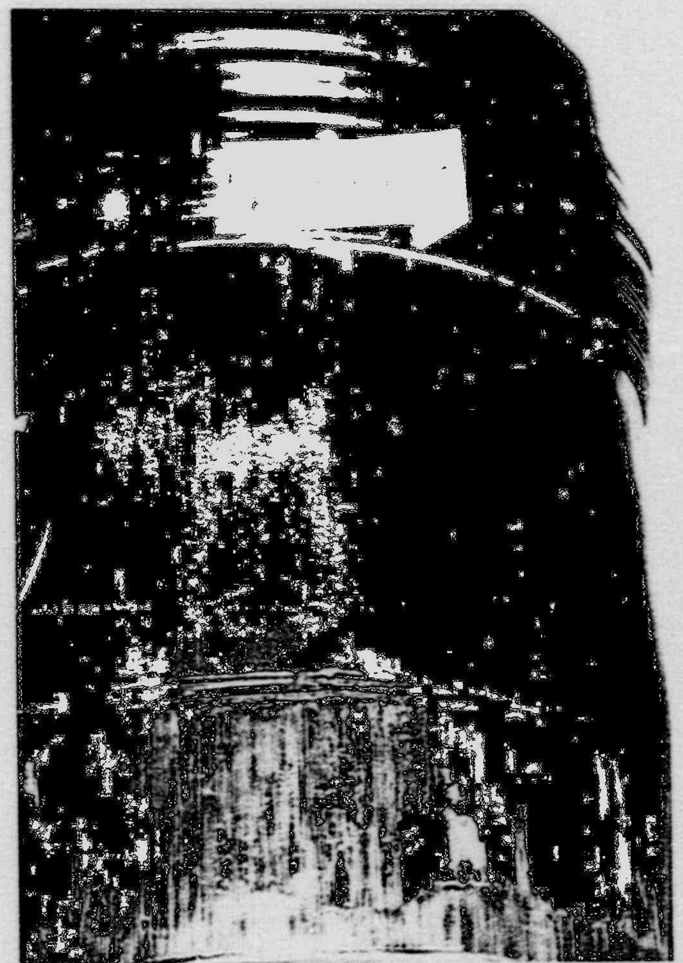


Figure C-6. Photographs of the damage on the thrust side (center of engine) of the 5R piston, C diesel.

a

b

Figure C-7. a) Photographs of the 5R piston end cap facing the generator end of the C diesel; b) The opposite end cap facing the aux. skid end of diesel.



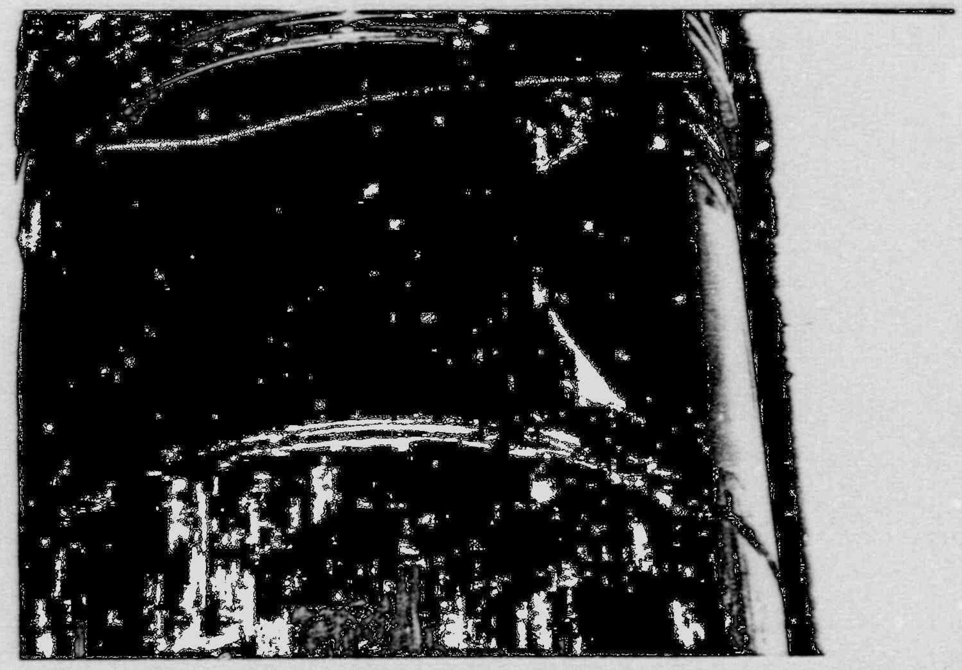


Figure C-8. Photographs of the damage on the non-thrust side of the 5R piston, C diesel; Note the two holes in the skirt at the bottom center of the photo on the right.

Figure C-9. The two holes and crack in the 5R piston skirt, non-thrust die. Hole "A" was sectioned and metallurgically examined.

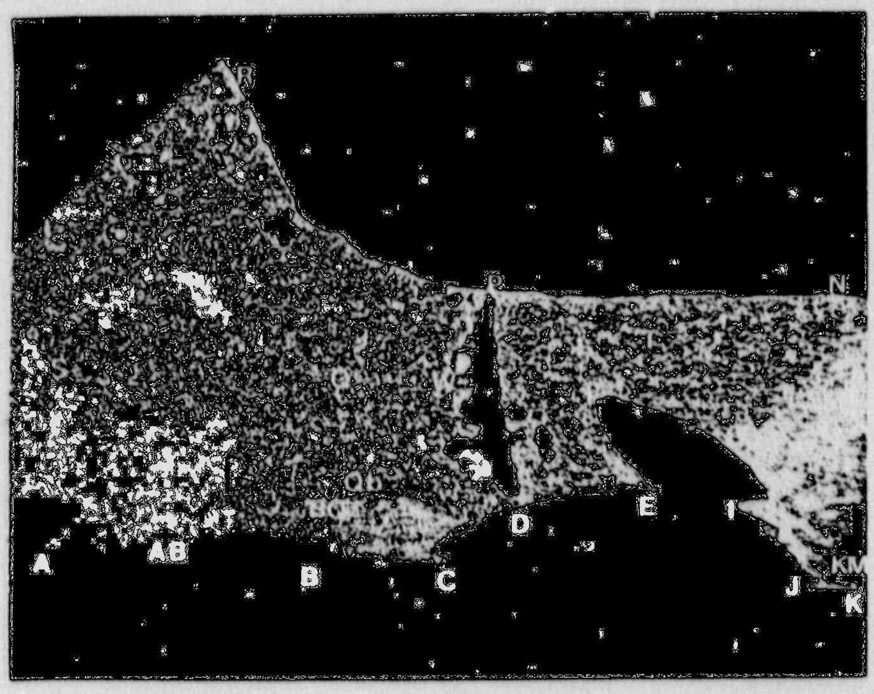


Figure C-10. A microphotograph of the cross section of the 5R piston skirt Hole 'A; in Figure C-9. The letters designate the areas evaluated metallurgically.

11a

11b

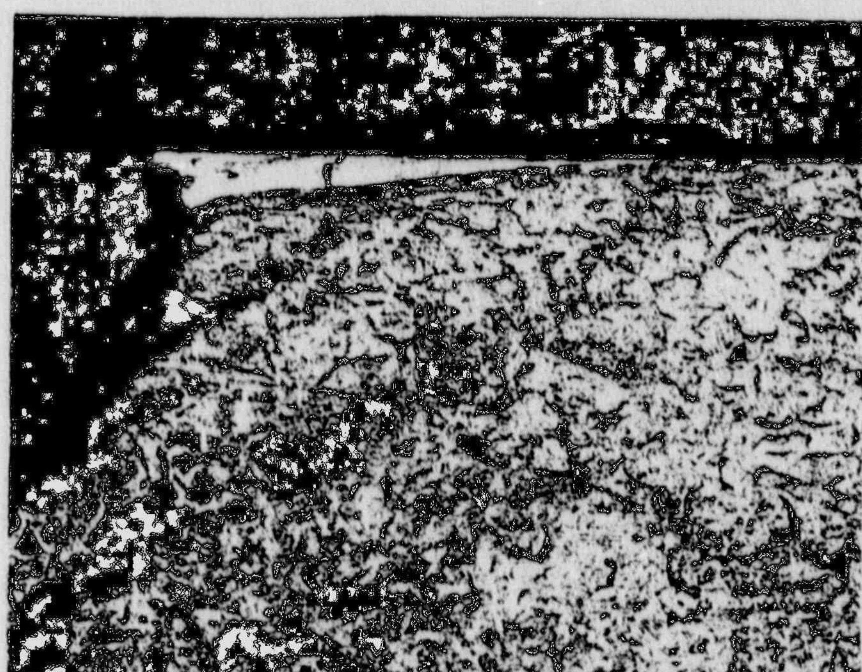
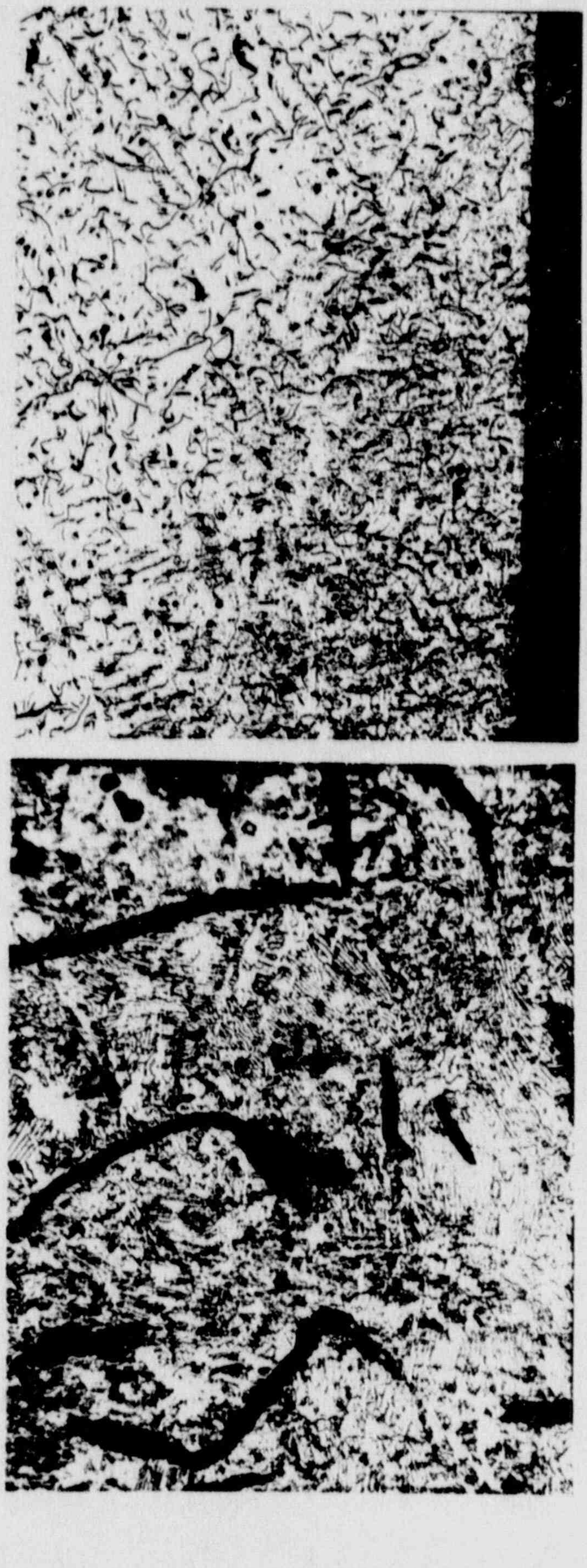


Figure C-11. Microphotographs of typical areas around hole 'A' in the 5R piston skirt. See Figure C-10 for locations, a) Area KM at 800C showing a course martensitic structure; b) Area K-J at 50X showing course martensitic and smear metal layer of iron chrome and tin on surface of hole 'A' c) Area H-G at 50X showing transition from a martensitic to a pearlite structure; d) Area C-B at 800X showing course martensitic structure;' e) Area B-Q at 800X showing pearlite structure around graphite stringers; f) Area A-B at 50X showing a mixed structure of fine pearlite and martensite.

11c

11d

Figure 11. Continued.

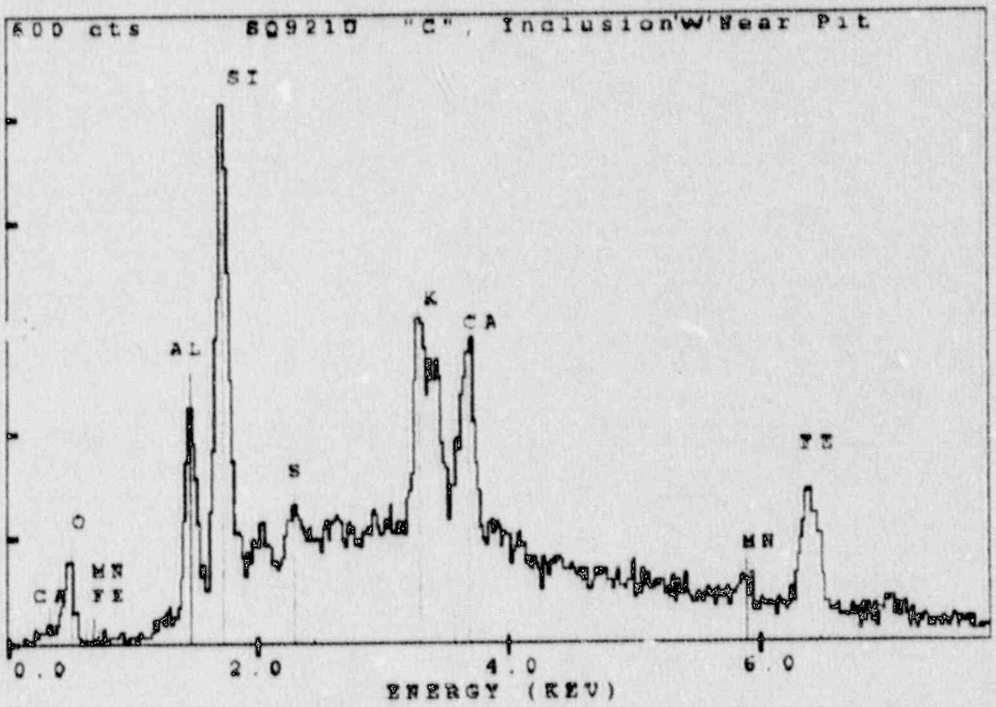


11e

11f

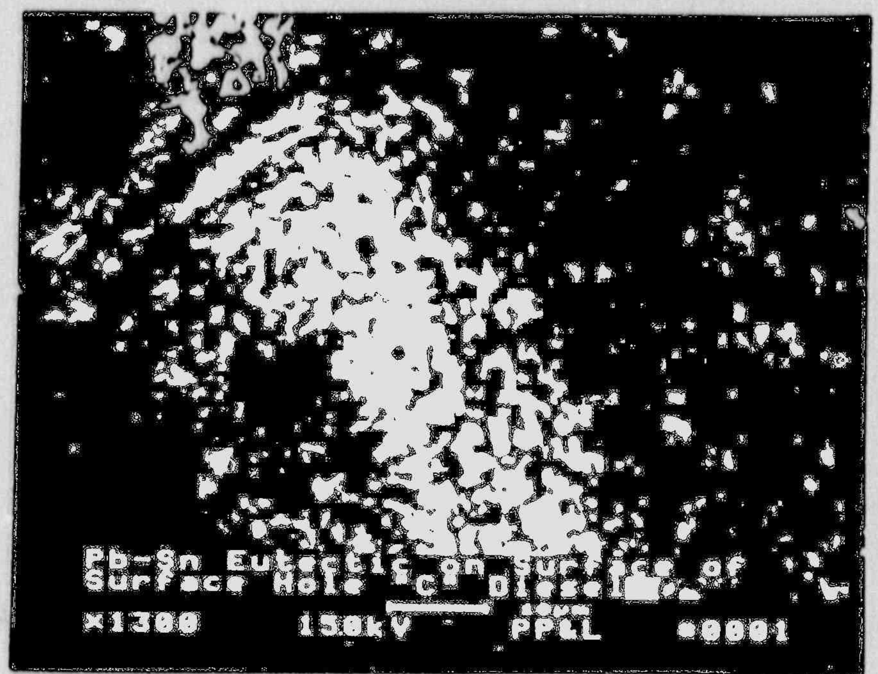
Figure 11. Continued.

a

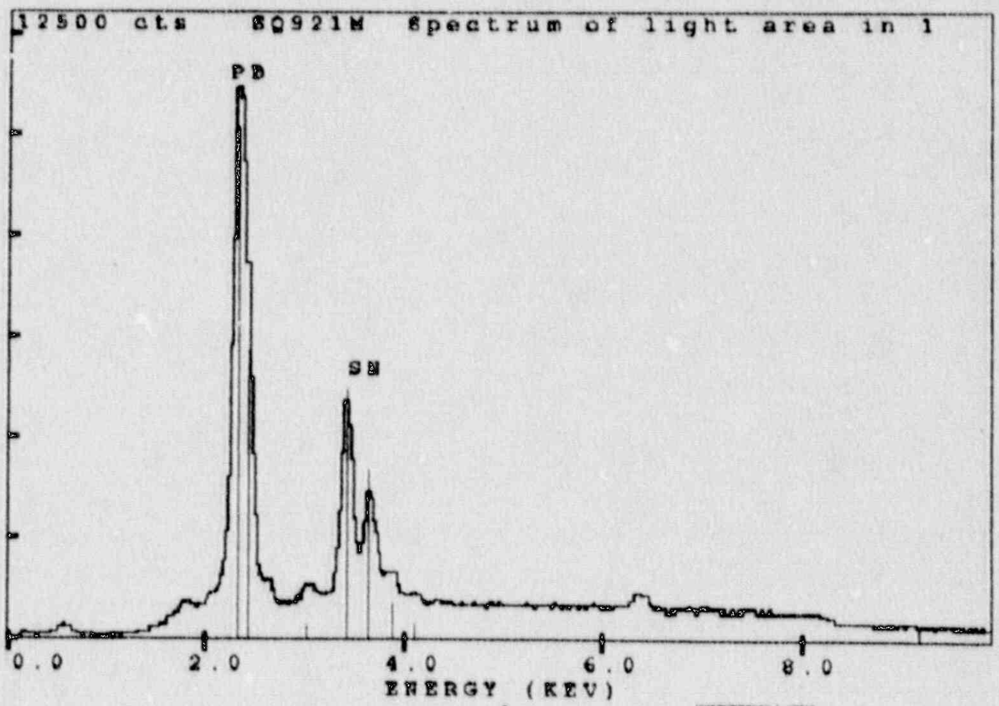


b

Figure C-12. S.E.M./E.D.S. analysis of inclusion 'W' below hole 'A' in 5R piston skirt a) SEM photo at 6.5X showing inclusion 'W' b) Spectrum of the elements in the inclusion 'W'.



a

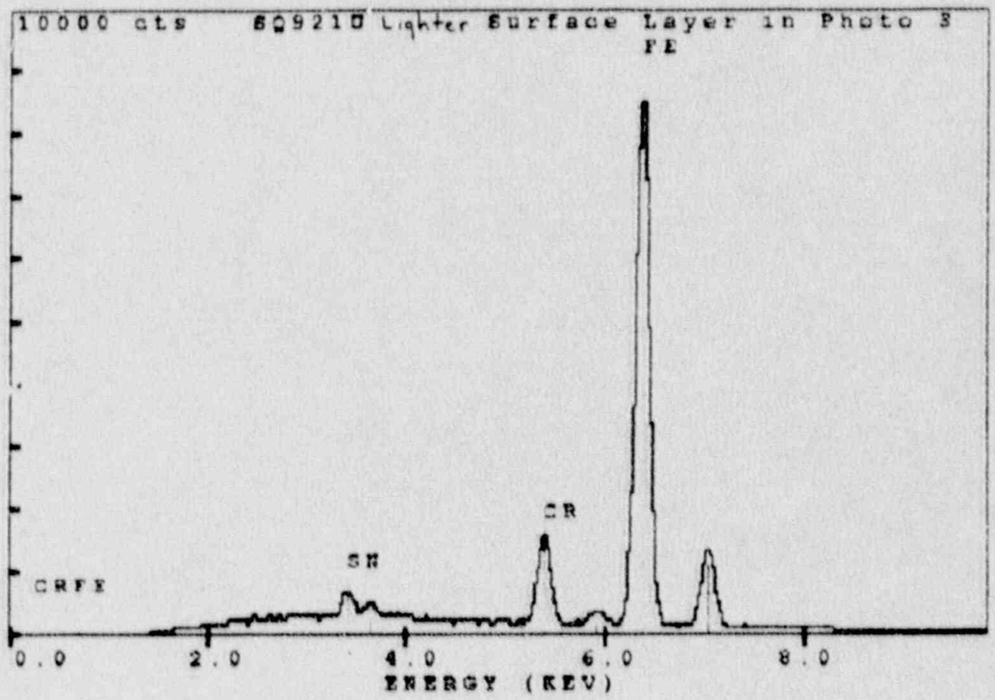


b

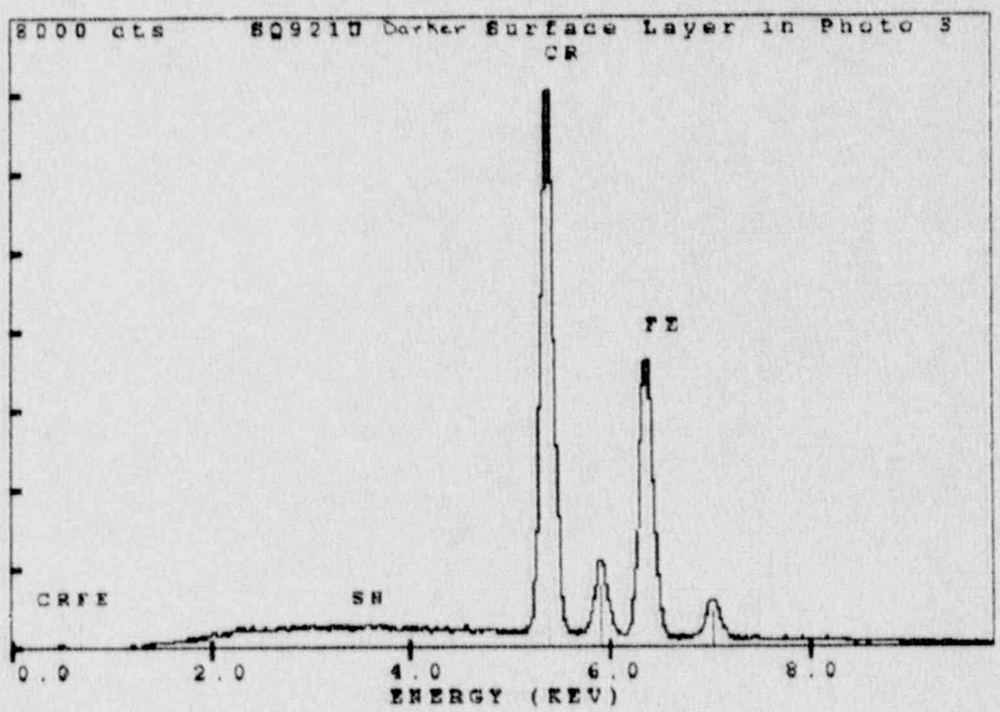
Figure C-13. S.E.M./E.D.S. analysis of the Lead-Tin Eutectic on the surface of hole 'A' in 5R piston skirts a) SEM photo at 1300X showing the Pb-Sn Eutectic on surface of hole; b) Spectrum of the elements in the light area in a above.

a

Figure C-14. S.E.M./E.D.S. analysis of smeared metal layer on surface of 5R piston skirt at Area J (see Figure C-10) photo at 4660X of smeared metal layer; b) Spectrum of LIGHTER AREA in photo in C-14a above; c) Spectrum of darker area in photo in C-14a.



b



c

Figure 14. Continued.

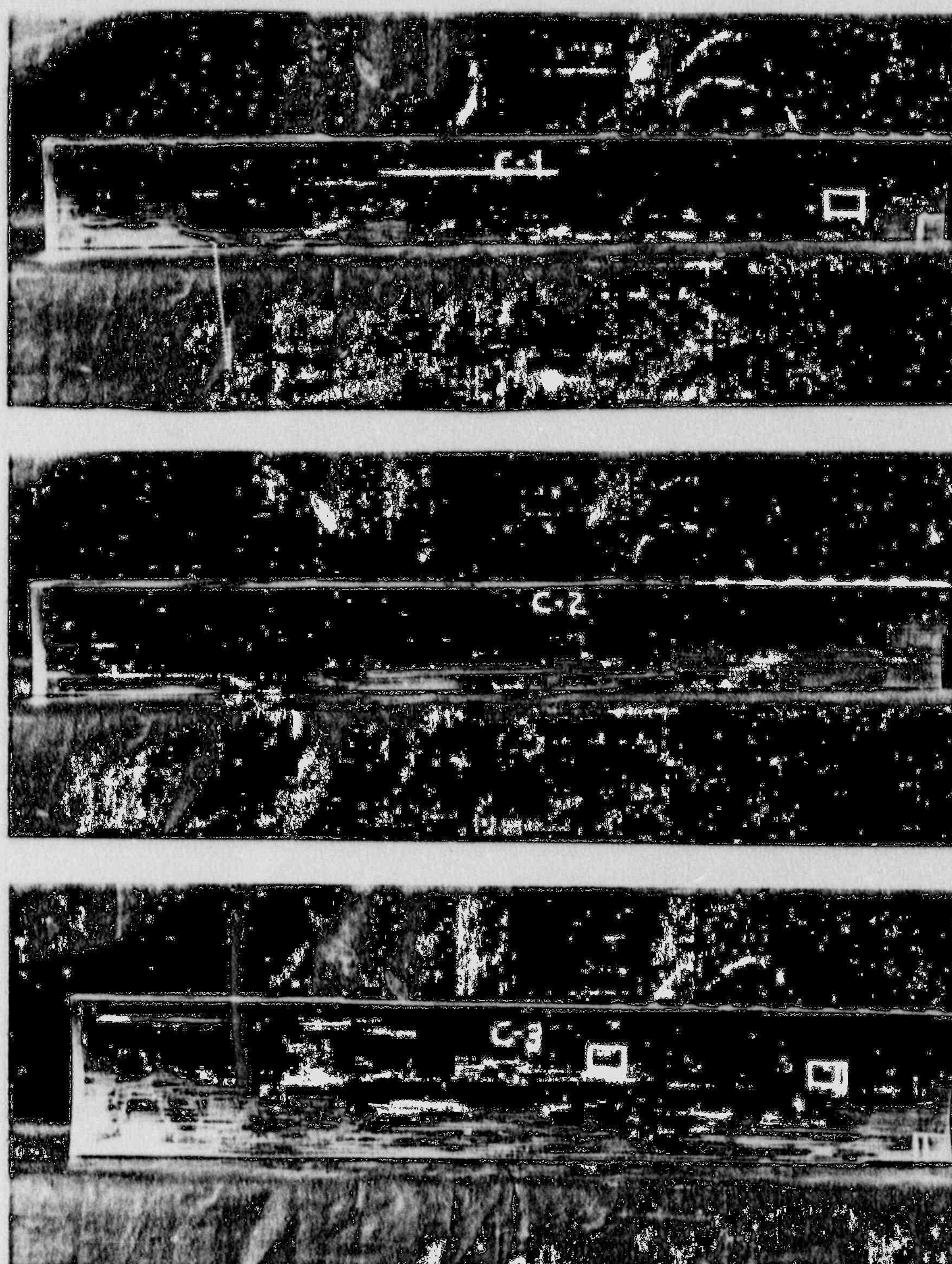


Figure C-15. Photos of 5R liner sections C-1, C-2 and C-3; a) liner section C-1 longitudinal section taken from thrust side; b) Liner section C-2 longitudinal section taken from end cap side; c) Liner section C-3, longitudinal section taken from the non-thrust side.

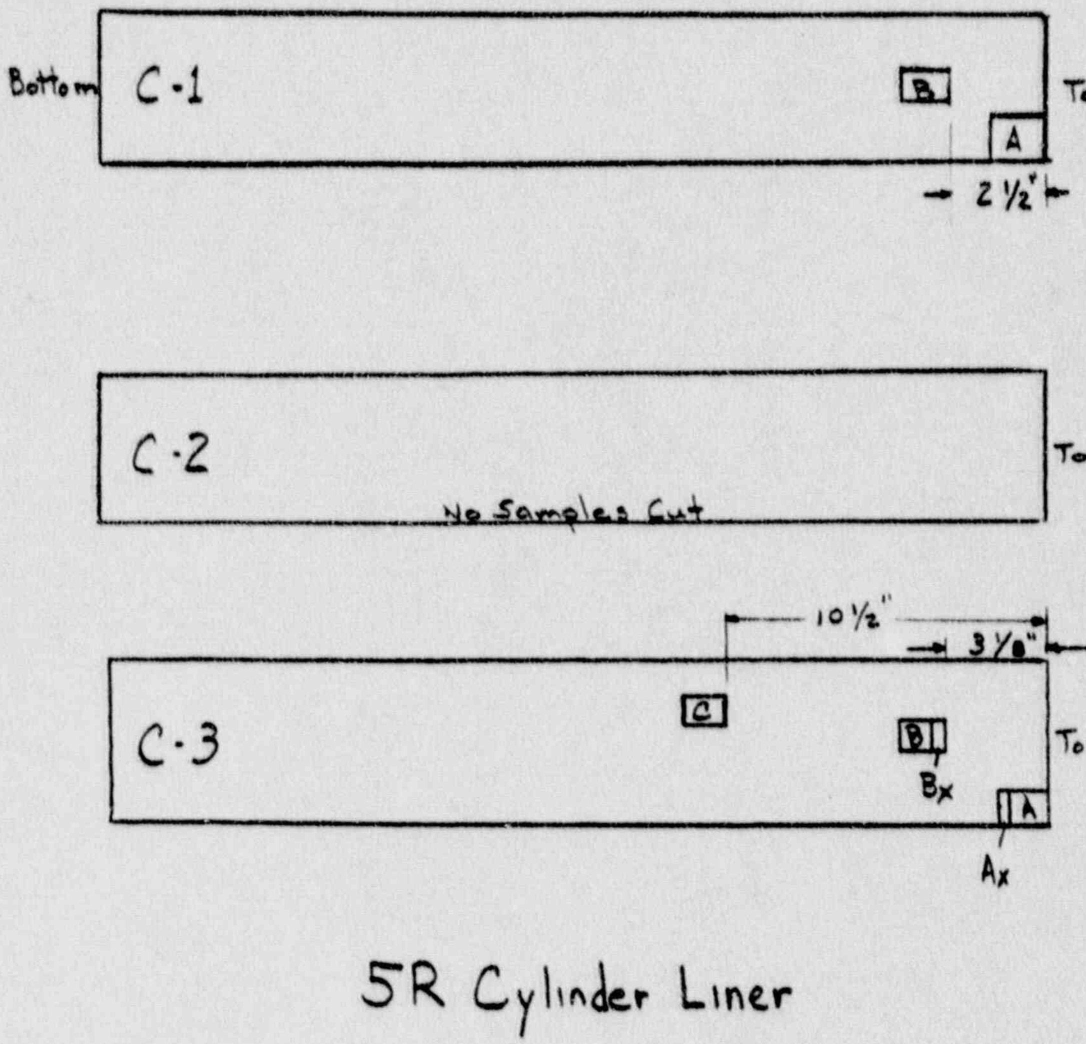
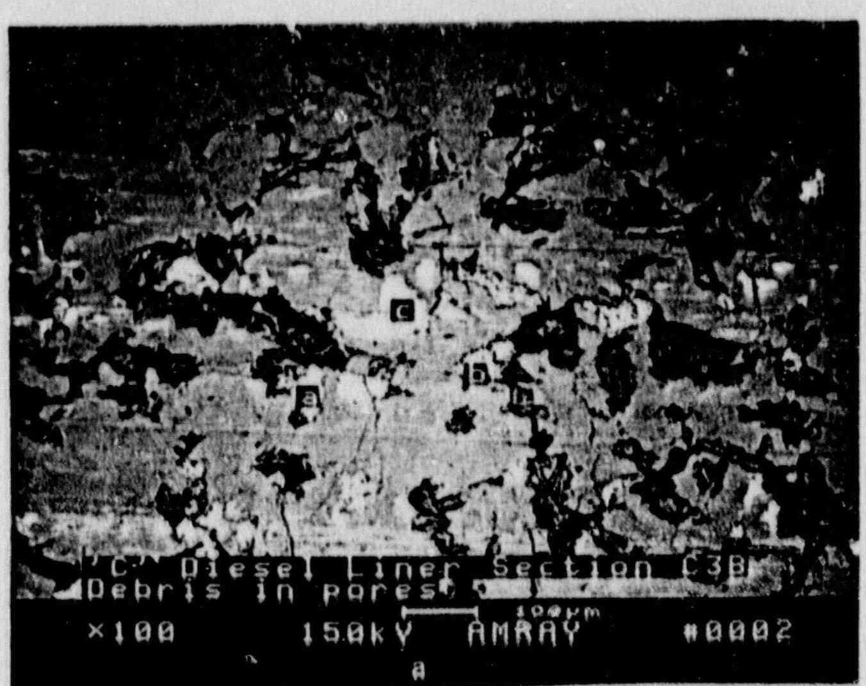
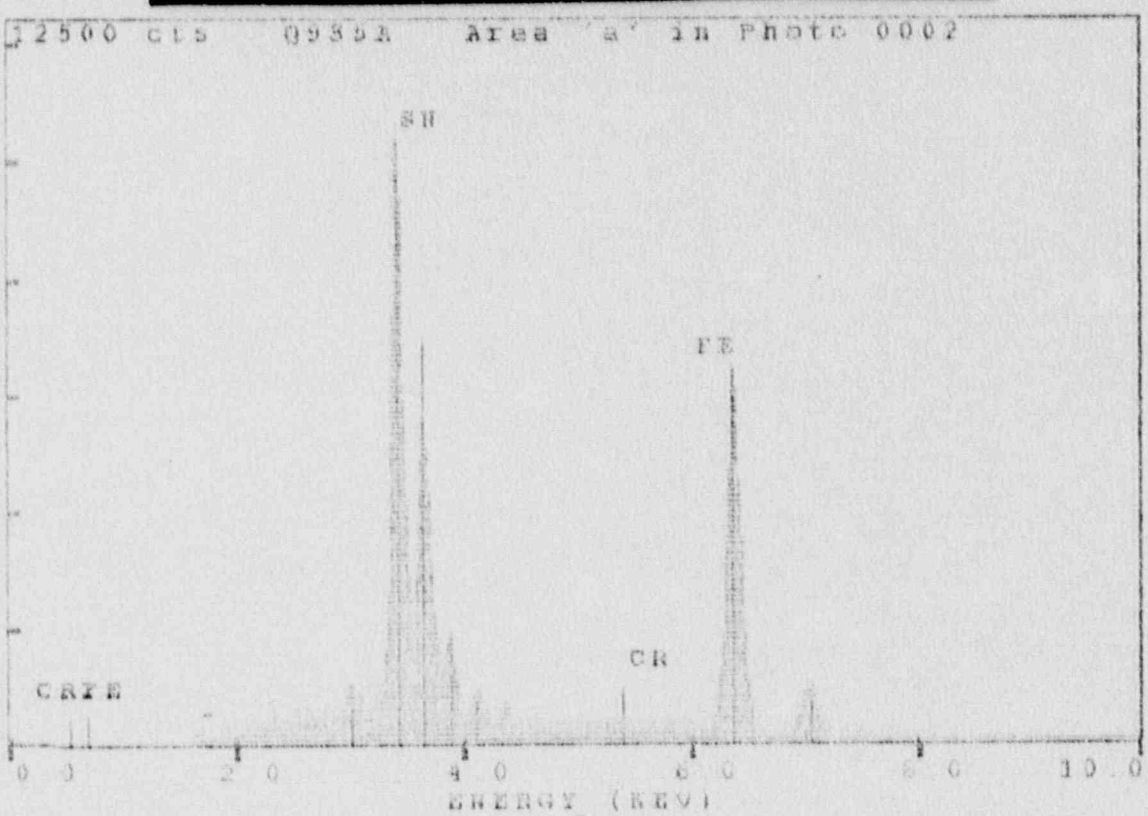


Figure C-16. Sketch of sections taken from liner pieces.

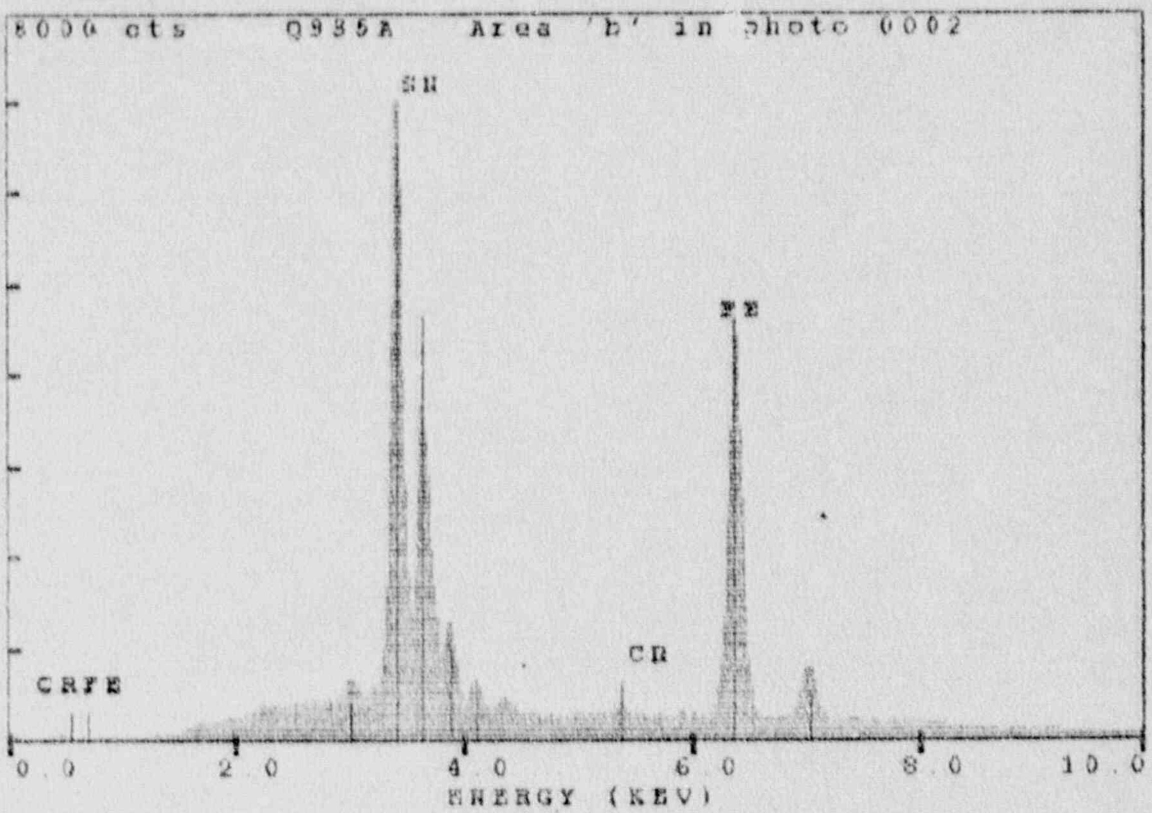


a

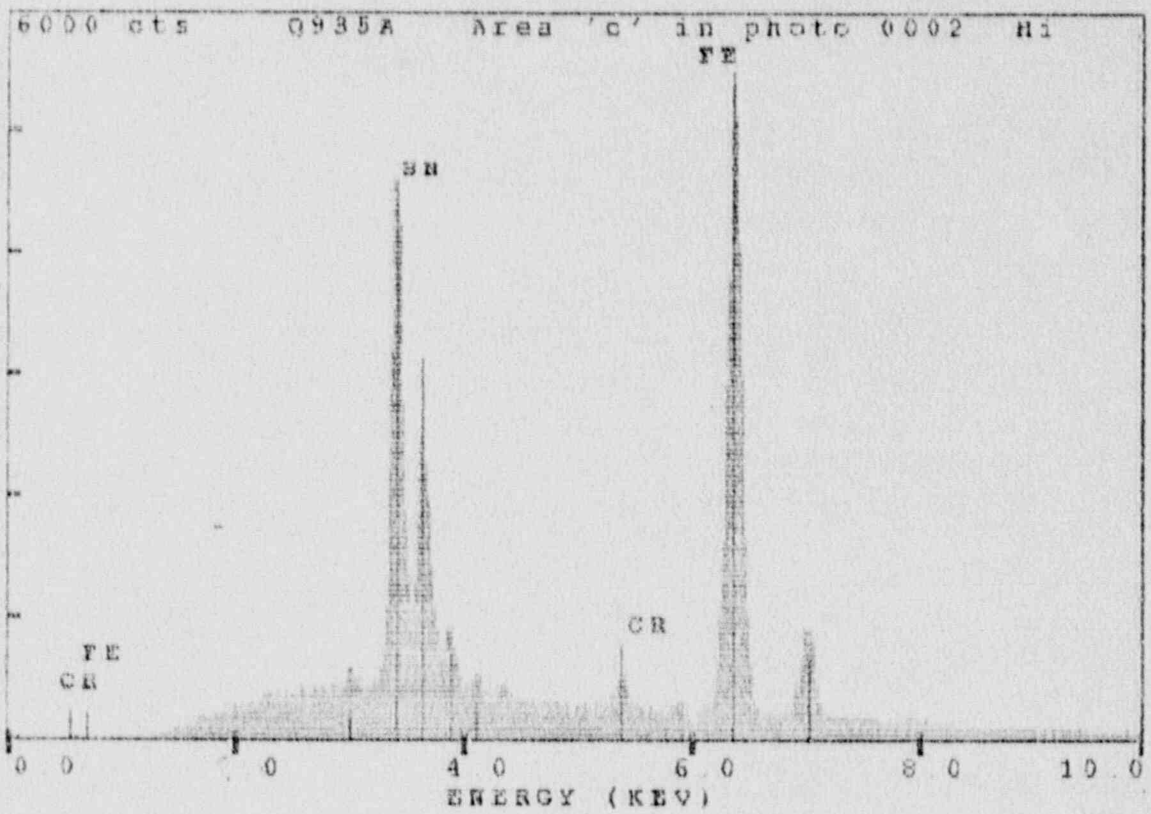


b

Figure C-17. S.E.M./E.D.S. analysis of the debris in the pores in liner section C-3B (see Fig. C-16); a) S.E.M. photo at 100X of section C-3 denotes area a,b, & c for analysis on E.D.S.; b) Spectrum of area 'a' showing the presence of mostly tin and iron; c) Spectrum of area 'b' showing mostly tin and iron with a small peak of chrome; d) Spectrum of area 'c' showing mostly iron and tin with a small peak of chrome.



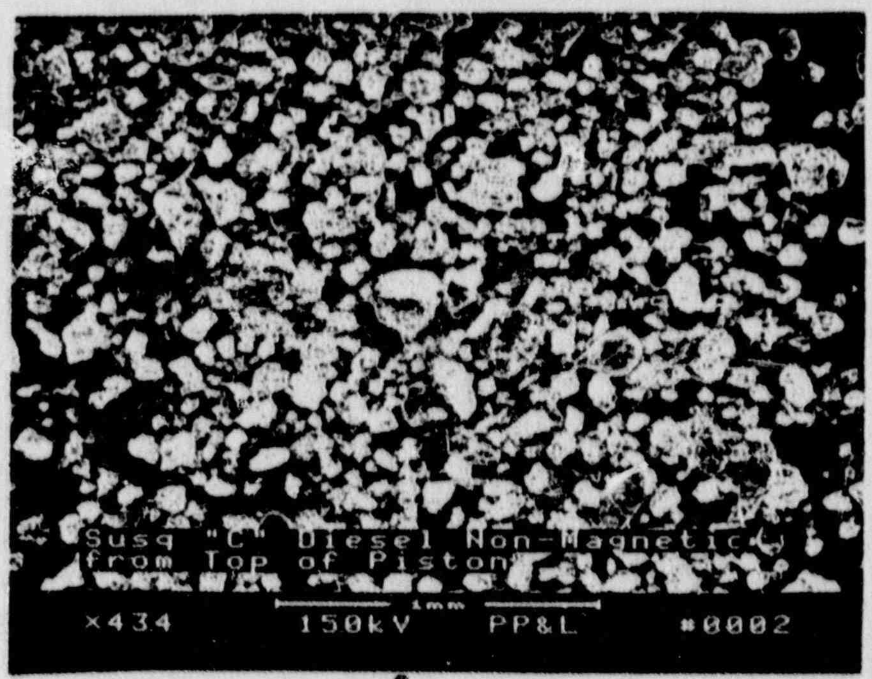
c



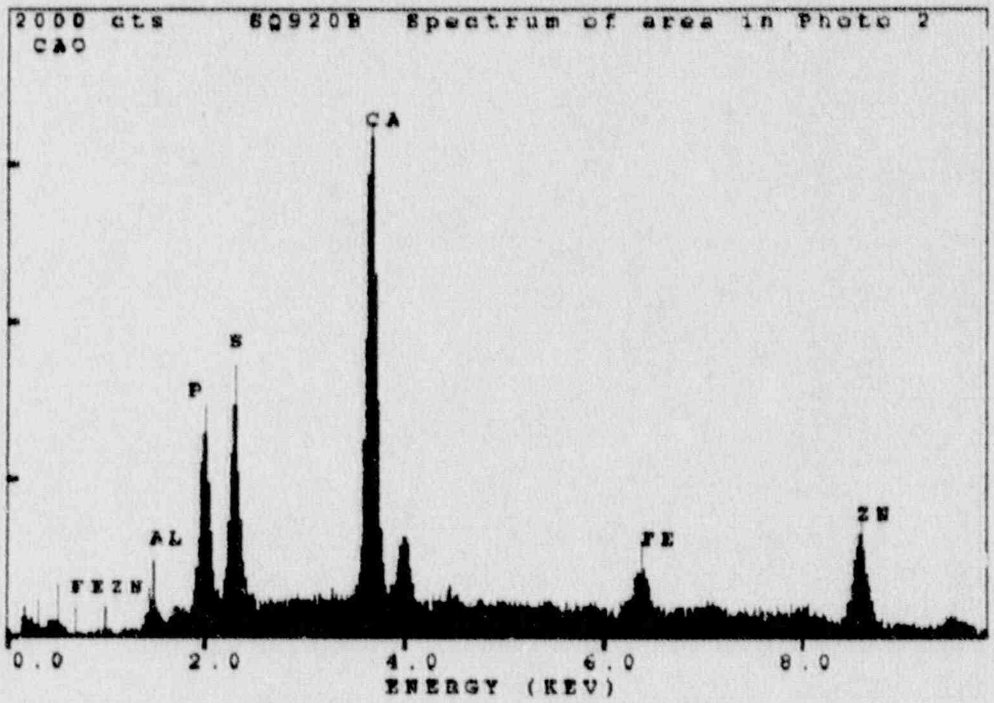
d

Figure 17. Continued.

Figure C-18. Microphoto from S.E.M. 1010X showing the iron and tin particles in a pore in the chrome layer on liner section C-3B.

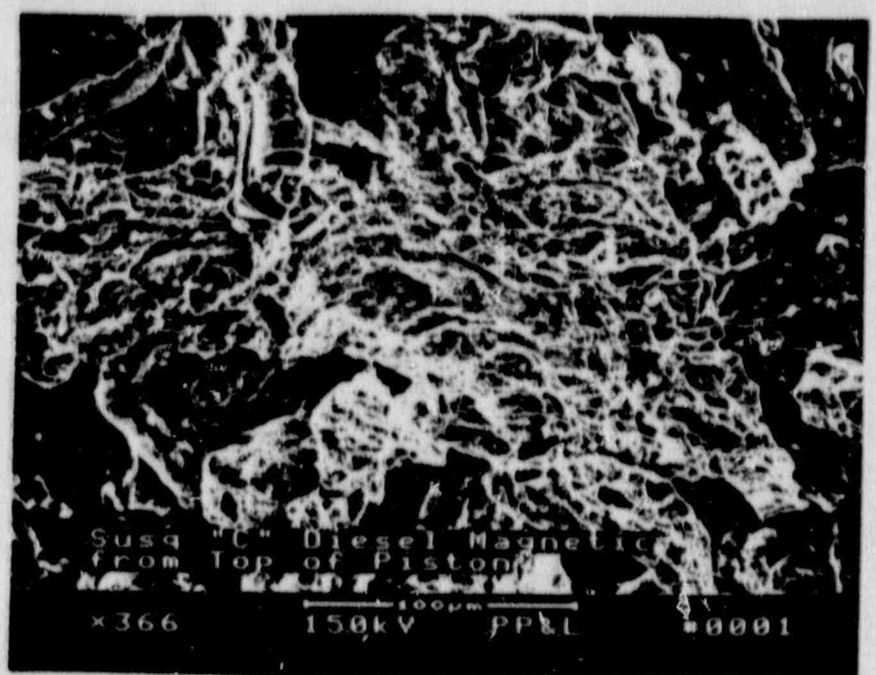


a

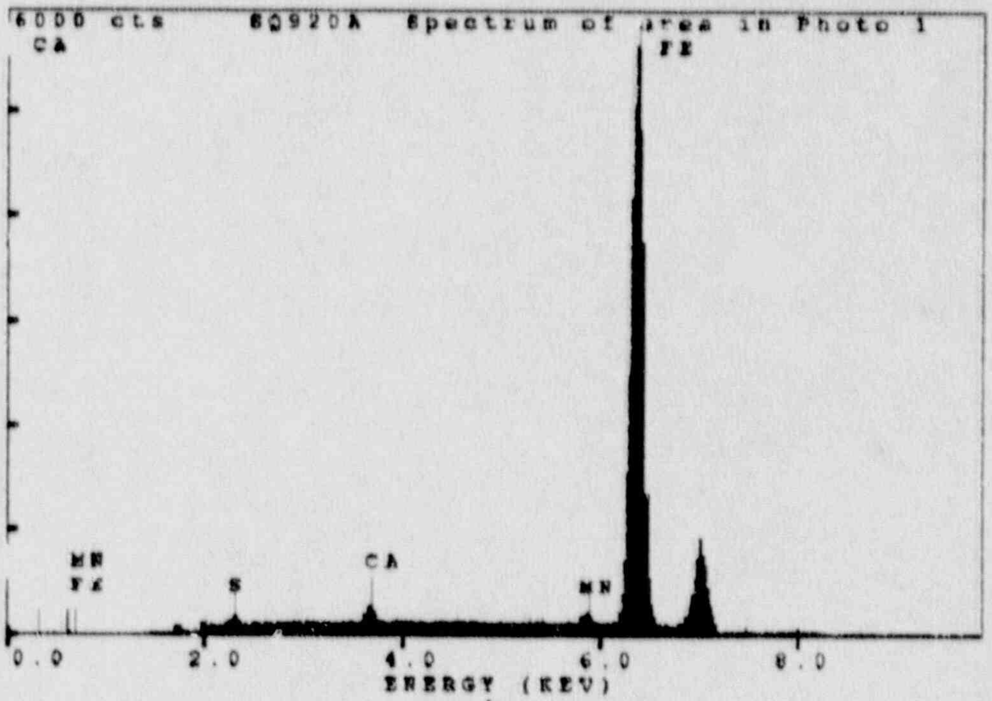


b

Figure C-19. S.E.M./E.D.S. analysis of non-magnetic particles taken from the top of the 5R piston, C diesel; a) S.E.M. photo of 43.4X of non-magnetic particles; b) Spectrum of area in the S.E.M. photo.



a



b

Figure C-20. S.E.M./E.D.S. analysis of magnetic particles taken from the cor of the 5R piston, C diesel, a) S.E.M. photo at 366X of magnetic particles; b) Spectrum of area in S.E.M. photo.

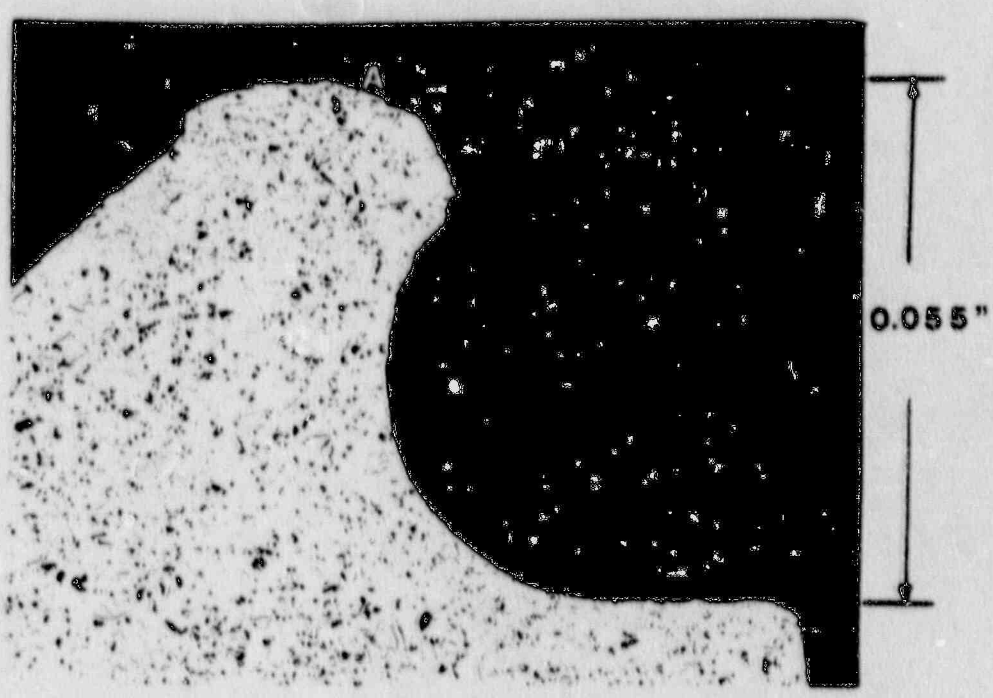


Figure C-21. Microphotograph at 50X of cross section of the lower oil ring from the 5k piston, C diesel. The deformed structure where the oil ring was worn can be seen at area 'A'.

VI. GENERAL DISCUSSION

1.0 Conditions Which Affect Engine Wear

1.1 Peak Firing Pressures

Reviews of the peak firing pressures from 1983 to the present for engines "A" - "D" have not shown any correlation between the peak pressures and piston failure. The peak firing pressure data is found in Attachment 1, which lists the pressure by cylinder number. Attachment 2 sorts the pressure data and associated cylinder and lists the pressures in descending order. A review of this sort reveals that the failed piston "C"-5R was always in the top three highest firing pressures. Yet, the "B"-7L was always, with two exceptions, in the lowest six firing pressures. Once the "B"-7L was the highest pressure cylinder and another time it was eighth highest.

A review of the sorted data for the "A" engine shows a sector of relative positions for 1R, 2R and 8R cylinders. (These cylinders were the ones which experienced tin loss.) It can be seen that these cylinders are in the middle of the pressure range or at the bottom.

The sorted data for the "D" engine shows that the 2L cylinder (which caused the crankcase explosion) was the sixth lowest pressure cylinder on the analyzer run before the explosion. Therefore, we have concluded that there is no correlation between the peak firing pressures and the crankcase explosions since they have been both high and low.

1.2 Limiting Factor for the 4700 KW Rating

The limiting factor for PP&L's 4700 KW overload rating is the size of our heat rejection system and muffler size. This information has been obtained from Cooper-Bessemer (C-B) (see Attachment 3 and paragraph 4.8 of Attachment 4).

Other utilities have 16-cylinder engines with overload rating higher than ours. Nine Mile Point and Waterford 3 both have overload ratings of 4840. Cooper Nuclear Station has a one-hour overload rating of 5000 KW.

Waterford 3 has a continuous rating of 4400 KW compared to PP&L's of 4000 KW.

C-B has established a limit of 250 BMEP (brake mean effective pressure) for the KSV engine as structural limit. According to C-B, the head studs prove to be the limiting factor. At 4700 KW, we are at 91 percent of the 250 BMEP limit.

When C-B tested a KSV in their shop, they obtained 250 BMEP at a peak firing pressure of 1690 psi.

1.3 Compression Ring Size and Metallurgy

1.3.1 Ring Dimensions: During the investigation of the diesel failures, someone found that some of the rings had wrong dimensions and questioned whether or not the supplier of the rings supplied the wrong type. This initiated an investigation that identified and characterized all the rings that had been recently removed from the SSES 1-3 engines. Attachment 5 at the end of this chapter shows the ring dimensional data that was obtained. The radial width (or thickness) is the width of the compression ring starting at the surface in contact with the cylinder liner and proceeding towards the ID of the ring. Attachment 6 shows the measured radial thickness for the first and second compression rings, specified to be 0.430 to 0.450", and the third and fourth rings, 0.445 to 0.465". The measurement data shows that of eleven measured ring sets, five piston sets had one or two wrong rings.

1.3.2 Metallurgical Condition of the Rings: Attachment 7 shows hardness and microstructural data obtained from four sets of rings sent to the PP&L Hazleton Chemistry laboratory for study. Three of the sets of rings came from actual pistons removed for other purposes and one ring was a new set supplied from the SSES warehouse where we would normally get replacement sets. Attachment 8 contains materials information for the materials of construction of the four compression rings. The first two rings, starting at the top of the piston are made of K28 iron which is a hard, martensitic based, nodular iron with a hardness of 40-46 Rc. The third and fourth rings are made of a perlitic based graphitic iron that is much softer than the nodular iron and whose hardness is 72-88 Rc. Hardness and microstructures obtained on the rings were consistent with their identified positions in all cases.

1.3.3 Discussion

1.3.3.1 The ring dimension in the radial direction is important to the operation of the engines because extra large rings would not completely recess into the grooves in the piston as the piston moves back and forth in the cylinder liner. If this happens, the rings bear all the inertial and reaction forces on the piston and would probably wear excessively fast. Even though we found five sets of rings that were oversized in the radial direction, we could not determine conclusively that they were so large that

they would not have fit completely into the ring groove when new. There is a minimum of 0.020" available by design behind the ring in the upper grooves before a properly fit ring would "bottom-out" and it is unlikely that the wrong rings would have been that much oversize to use up all the gap available. At worst, the wrong rings may have been machined to the dimensions of the third and fourth rings (per C-B specified dimensions) in which case only 0.015" of the designed gap would have been used up and this would have been acceptable from a clearance argument.

1.3.3.2 Metallurgically, the materials of construction of the rings we examined were found to be as per the design criteria. Therefore, we conclude that the extra wear found on these rings could not be attributed to the use of the wrong material.

1.4 Engine Oil Condition

1.4.1 Lube Oil Foaming

There is a concern that the lube oil is foaming and a foam oil mixture or at least entrained air is being pumped to the engine bearing. This type of defect in the oil would be expected to affect the more heavily loaded bearings, i.e., the piston pin and crankshaft bearings.

An in situ oil foaming test was conducted on the "A" engine on December 1, 1989. A clear plastic door was put on the engine in place of one of the lower doors and affixed with long studs and nuts. The regular door was placed over it and nuts placed on the studs. The metal door remained in place during the 12-hour post-maintenance run for personnel protection. It was removed after the engine was unloaded but before shutdown.

While the engine was running, about 2-3" of foam was present on the side of the plastic door. Normal oil level (engine at rest) would be at about the same height.

After the engine was shut off, a deluge of oil was draining down from the top side of the engine. When the flow of the draining oil approached that of the prelube pump flow, a continuous mat of foam, approximately 1/2" thick, was observed. Small bubbles were also observed rising in the lube oil. The mat of foam started to break up in 15 minutes so that small islands of oil could be seen. The presence of the small bubbles migrating to the surface was still observed at 15 minutes after shutdown; however, the amount of bubbles was reduced from that of initial engine shutdown.

PP&L, after consultation with Gulf, the oil supplier, Chevron, Ricardo and PP&L's in-house oil lab, could not come to a conclusive good/bad decision. We decided to take oil samples from engines "A"-"D" and new oil and send them to an independent laboratory for an oil foaming test.

Oil foaming test results from SGS Control Services, Inc., are shown in Attachment 9. All of the oil tests shown were performed per ASTM D-892 standards and reveal that there is no foaming problem with the oil used in any of these engines. The two deviations from a "Nil" result in the "A" and "B" engine results are not significant since 10 minutes results show them to become "Nil." Section 8 of ASTM D-892 is shown in Attachment 10 which explains how the test sequences are performed.

1.4.2 North Anna (VEPCO) Foaming Experience

Colt Industries supplied the emergency diesel generators to the North Anna Station of Virginia Electric & Power Company. Colt has reported to PP&L that they were experiencing piston to liner seizures at North Anna. The cause was oil foaming. This oil foam mixture did not provide the lubricity to the piston pin/bushing that was required. This caused overheating of the pin and bushing, the bushing expanded and distorted the piston carrier, and the carrier distorted the piston which contacted the liner and produced excessive heat. This led to seizing and failure of the piston/liner.

North Anna was using Gulf Super Duty 40 oil when they were experiencing the diesel problems, and once the root cause was determined they switched to Chevron.

Colt remarked that they do not feel Gulf is consistent with their additive packages and one batch of oil can perform fine and another batch may cause problems.

1.5 Piston Pin Condition

Our initial investigation into the cause of the crankcase explosions focused on the piston and liner; hence, the condition of the pins were not thought to be an important factor, i.e., the pin condition was secondary to the piston damage. Our opinion on the pin condition changed after three pistons which experienced tin loss were pulled from the "A" engine. These pistons with various degrees of tin loss all had pins with some degree of heat bluing. It was also noticed that the end caps had started to migrate out of the pistons. A matrix of observations was created to make correlations among all the observations. This matrix is Attachment 11.

The two failed pistons from the "B" and "C" engines (B-7L & C-5R) virtually gave no clue as to the cause of the failure. The only thing to prove to be significant was the blued condition of the piston pins which was not recognized until later in the investigation; see Attachment 11. Bluing of steel is a high temperature oxidation of the steel and occurs at about 900°F. The pin bluing was considered by C-B to be a secondary effect of the piston seizure; see Attachment 12. It was not until three pistons that had lost tin were pulled from the "A" engine and blued pins were discovered in these pistons that the blued pins and piston tin loss seemed to correlate. An effort was made to closely examine all the pins that had been removed for signs of overheating. As it can be seen from Attachment 11, blued pins or heat distress was found on six of ten pins. Three of the other pins were worn or scratched. The fourth needed to have its bushing scraped in order to obtain an acceptable fit. It can be concluded that these four pins were experiencing higher than normal friction but not enough to cause the pins to turn blue. At least one piston pin has been annealed (approximately 1200°F) in situ since its hardness went from a Rc of 50 to 20. The metallurgical sections of this report contain more detail about the specific pins and their property changes. Note: Insufficient bearing contact area on the A-3R bushing would also cause higher than normal friction.

The high friction in the piston pin joint will inhibit the toggling of the piston relative to its connecting rod. This condition will cause the top edge of the non-thrust side of the piston to try to scrape the oil from the liner on the upward strokes. The torque developed by the piston pin will be reacted by a couple.

A couple is two equal forces producing rotation by acting in parallel but opposite directions. One side of the couple will be made by the liner pushing on the top of the piston on the non-thrust side. Note: This will occur just under the bottom of the middle oil ring since this is where the piston is at its largest diameter; see Attachment 13. The other half of the couple will be made up of possibly two forces. One is the normal thrust load on the piston acting at the piston pin. The second force would be produced on the lower skirt on the thrust side. This latter force has not produced any signs of piston wear for a couple of reasons:

1. This force is only going to reach a magnitude to satisfy the unbalanced torque. Note: The length of the moment arm here is long compared to the first part of the couple's moment arm.

2. The angle of the piston would mate with the liner due to the piston pin torque trapping the oil on the liner and the piston would "hydroplane." This is not the case for the top of the piston on the non-thrust side where the sharp corner of the piston would remove the oil film.

With the protective oil film removed from the non-thrust side of the liner surface, it is possible for cold welding to take place between the liner surface and the piston surface. Relative motion then can transfer tin and iron to the liner and conversely chrome can be transferred to the piston. This makes the mating surfaces rough and produces added friction, increasing the wear rate and eventually leading to an overheated piston. This failure scenario is supported by the fact that an inspection in July 1987 of the "C" diesel revealed loss of tin on the 5R piston; see Attachment 14. Since adequate inspection criteria was not in place at that time, the piston was returned to service. Cylinder 5R is the cylinder which caused the crankcase explosion on the "C" engine.

The previous inspection report for the "B" engine was reviewed and no tin loss was noted for the 7L cylinder. It should be noted that the type of inspection performed was an underside inspection. The inspection doors are removed and the pistons brought to top dead center; hence, only the lower half of the cylinder liner can be viewed in this type of inspection.

1.6 Correlation of Running Time and Starts to Explosion

The data from Attachment 15 has revealed that the "B" engine suffered a crankcase explosion after 547 starts but if this number is corrected for the 300 start tests on "B" (actually about 320 starts), it would approach 870 starts. Our start log does not include start-up testing starts. The "B" engine ran 1,033 hours before the explosion, and this represents the total engine run time including shop tests.

The similar data for the "A," "B," "C," and "D" engines are as follows:

	A	B	C	D
Hours Run	982	1033	979	966
Starts	511	1870	555	549

One can see that "A" engine has more run time than "C" and "D" engines but has not had a crankcase explosion at SSES. The start data (date and start number) is not currently available prior to December 1986. However, a straight line estimate using available start data of number of starts that "D" accumulated before its crankcase explosion of January 14, 1984 is 381 starts. This would have crankcase explosions at:

Engine	Starts
B	870
C	555
D	381

"B" engine has had approximately twice the starts as "D" before it experienced an explosion and about 50 percent more starts than "C" would if it had a crankcase explosion. Therefore, we have concluded from this scatter there is no correlation of the number of starts to the crankcase explosions.

1.7 End Cap Migration

The piston pin end caps are installed in the piston to prevent ring blow-by gases from entering the pin/bushing interface and interfering with proper lubrication of the pin. The end caps are held in place by an interference fit of 1/2 to 2 1/2 mils. These caps are driven into place by tapping them in with a hammer until they seat against the piston bushing hole.

The outer portion of the end caps are radiused so they match the OD contour of the piston. On the interior, they have two intersecting ribs. These ribs have been a subject of concern since some are machined flush with the plane of the seating surface and others are cast so that the top of the rib is below the plane of the seating surface. C-B has the same part number for each style and contends that their difference is only in foundry technique. This difference is not a factor in the function of this part. It was initially thought that the relief built into the end cap was for piston pin thermal growth. However, measurements of the pin length and bushing length showed that there is a 1/4" difference; the pin being shorter. It would be virtually impossible for the pin to move that distance by thermal growth alone to push the end caps out forcibly. Combinations of thermal growth of the bushing and mechanical movement may be possible to put a force on the end caps, but this has not been pursued adequately at this point.

There is one benefit of having the relief on the inside of the end cap and that is in the assembly of the piston and rod into the engine. There is enough room between the bushing cut out in the interior of the piston to allow the pin and connecting rod to slide in the piston pin bushing more than 1/8" in either direction. Hence, the relief would allow more movement than without the relief in the end cap. From a practical view point, this stated benefit is of little use since the pistons and rods are handled slowly and carefully during their insertion into the engine.

The end cap migration or unseating of the end caps from the bushing has caused some problems. The liners have been burnished by these caps and one was replaced due to burnishing and metal loss caused by end cap migration.

What are some of the mechanisms of end cap movement outwards to the liner? The following list sums up our thinking on this subject and a brief explanation as to why the mechanism is acting.

1. Too much or too little force fit clearance between the OD of the cap and the hole into which it fits. The cap is manufactured to specifications that require it to be 0.0005 to 0.0025" bigger than the hole. The cap has to be tapped into place with a soft hammer. If the clearance is not tight enough, the cap would be loose and vibration may cause it to "walk" out of the hole as the engine is running.

   A cap too much oversized would be difficult to install and would be easily discovered by the mechanic performing the installation. If he could force fit a large cap in the hole, it may be possible that it would not go all the way in to bottom out as it should. This would be easily noticed. We have not made a systematic investigation of cap sizes and hole sizes to determine if a QC problem exists here. It has been noticed, however, that all caps removed manually required a significant amount of force to remove and in some cases, a hydraulic press had to be used to remove them. Hence, we do not believe that size is a problem.

2. Thermal and or mechanical ratcheting. The idea here is that the easy direction of movement of the cap is outwards. Any force acting outwards on the cap will more likely move it out, whereas inward forces would be less likely to move it in. This is because the outer surface of the piston and cap is more elastic (i.e., can expand more easily) than material below the surface. That is, the piston is stiffer in areas where there is more material surrounding it. The outer surface of the cap is stiffer because it is solid metal. Whereas the inner surface is ribbed and hence more elastic. Thus, after the cap is inserted into the perfectly cylindrical hole, the cap and the hole will become slightly conical. If the cap is moved outwards for any reason, the hole and cap will be even more conical. Therefore, if the cap is in an intermediate position more force would be required to move the cap inwards than outwards. The conclusion one can come to, after realizing the above, is that any thermal or mechanical vibrational forces acting on the cap will gradually move the cap outwards over a period of many cycles, if the forces are sufficiently large.

3. Thermal expansion of the bushing. The bushing is shrink fit into the machined hole in the piston and is held there by an interference fit. The bushing material, being bronze, has a higher thermal expansion coefficient than the cast iron material of the piston. We know from the bluing by oxidation on the piston pin (which is in intimate contact with the bushing) that temperatures of 1200°F or more are present. This would certainly raise the temperature of the bushing up to its initial melting point of 360°F. At this temperature, the bushing would have expanded some 0.022" over its entire length. Since the end cap is in contact with the end of the bushing, one of the caps

could be rationalized to be moved outwards this distance at one time. This is not enough distance to cause contact between the end cap and the liner since the caps are recessed from the liner some 0.063" or more. Furthermore, we have not noticed the ends of the bushings sticking out beyond the edge of the bottom of the cap recess. More inspections are needed to make a definite statement that this is not actually happening. We do know that severe blackening (oxidation, oil decomposition) of the bushing material takes place and therefore high temperatures have been experienced by the bushings.

4. Physical movement of the engine crank shaft. By design, the connections of the crank to the rod to the piston pins should hold the pin roughly centered in the pin bushing length leaving a gap of about 1/8" on either end of the pin before the end cap is reached. On a crank that is some 20' long, it is possible that machining tolerances, thermal expansion and material instability could make this 1/8" tolerance vary considerably from one end of the engine to the other. It is therefore possible that one or more piston pins may contact and push against an end cap forcing it against a liner. C-B has told us that this is physically not possible by design, but we feel that this is an open item to be proven one way or another.

1.8 Tin/Metal Transfer to Linings

In the chapters of this report discussing the B-7L and C-5R cylinder failures, there was considerable discussion of metal transfer between the piston surface and liner surface. During the final stages of degradation leading up to the crankcase explosions, a lot of tin, iron, and chrome is transferred back and forth between the piston and the liner. The beginning of metal transfer, however, appears to occur far ahead of the final failure in time. We have noticed in our recent inspections that tin transfer is found in all of our engines, but not necessarily in all cylinders of a given engine. Attachment 11 shows a table of observations made on various cylinders of engines "A," "B," "C," and "D" and reveals that tin smear is common.

The importance of tin or metal transfer to the cylinder liners is that the normal lubrication process is degraded by this process. As tin or iron from the piston collects in the pores of the chrome plating on the liner, it displaces the oil that should be retained there for lubricating the interface between the liner and compression ring mating surfaces. If no oil is present in some areas, the rings will run dry and friction will increase. This raises the temperature of the rings and causes additional wear. In addition, the tin can pick up wear particles produced by the friction and retain them. This makes the tin surface even more abrasive, accelerating the wear of the rings.

Discussions with C-B representatives informed us that the transfer of tin to the liner is a normal, expected process during the break-in cycle of the engine early in life. Any tin transferred to the liner usually sloughs off during running and eventually disappears entirely. The reason the tin is put on the piston OD surface in the first place is to provide a soft bearing type surface which will slip easily over the surface of the liner and pick up any abrasive particles trapped between the liner and the piston. This essentially is supposed to prevent these particles from further abrading either surface. This, in theory, is what happens in any sliding bearing, such as a Babbitt or bronze bearing running against a steel surface.

We have observed (after many inspections of our engines) that this tin layer transferred to the linings can be found in various stages of appearance from some very localized thinly smeared layer, to heavier deposits covering larger areas, but which are still very thin with no evident roughness. Heavier areas have been observed on other linings which are thick enough to catch a fingernail rubbed over it and which may contain scoring marks of some depth. In all cases, the tin has been transferred only from the non-thrust side of the piston (the side not in contact with the liner during the power stroke of the cylinder).

In cases where tin wear and transfer has been high, it was found that the tin had been completely removed from the surface and the underlying iron had been exposed and worn down. At this point in the life cycle, iron is being transferred to the liner and is probably causing more than acceptable wear on all the parts. If inspections can determine when this stage is occurring, the pistons should be removed and refurbished or replaced along with the liners to prevent subsequent damage.

1.9 Failure of Head Gaskets

During the removal of the heads from the engines after the B-7L cylinder failure, it was noticed that the head gaskets had disintegrated. This has never been observed in the past so an investigation followed which quickly determined the reason. The original engine gaskets supplied by C-B were manufactured with asbestos filler material and it was discovered that recent shipments of replacement gaskets were actually composed of a graphite based filler material. It was the graphite material that was found to have disintegrated. It is not known if the disintegration particles actually made it down to the piston head or if they were in any way involved in the failure of the "B" engine. Contact was made with C-B after this was discovered and it was agreed that all future purchases of gaskets will be of asbestos composition until this issue can be resolved.

2.0 Consultants

PP&L hired two consultants, Ricardo (U.K.) and Southwest Research Institute, to assess the damaged parts, operation and maintenance of the diesels and to make a recommendation on what needs to be pursued to determine the root cause of the failures. Both consultants' reports are attached as Attachments 16 and 17.

The salient points from each report are as follows:

2.1 Southwest Research Institute (SRI)

- o SRI believed the problem to be the way in which we operated the engines, full load in 90 seconds after starting. They feel that the rapid load application causes the top of the piston to expand at a rate higher than that of the liner. This expansion causes an interference between the piston and liner; hence, the scoring and crankcase explosions.

- o SRI did not have the benefit of seeing a piston which had lost its tin plating on the non-thrust side. This information was communicated to them by telephone, but it did not change their initial impression.

- o SRI submitted a quotation in their letter report to install RTDs to the inside of a piston. Their signal would be transmitted by a device attached to the interior of the piston. With this arrangement, SRI would be able to measure the temperature of piston. This data would then be inputted to a computer model which would predict the size and shape of the piston during transient.

It should be emphasized that these extreme temperature excursions will only occur in the top section of the piston, the area of the crown and down to the compression rings. The pistons that have been pulled due to tin loss have not shown any initial signs of the top contacting the liner. It has been observed in some pistons that the deposit layer on the vertical area above the top compression ring scraped the liner, but we believe this is secondary to the tin removal and wear of the piston.

PP&L has discounted SRI's theory of piston swell causing the crankcase explosions for the reason cited above because this scenario should affect all pistons equally and all KSV engines in other utilities equally. This has not been the case.

2.2 <u>Ricardo</u>

Ricardo Consulting Engineers, Ltd., from England were also hired to assist in the investigation. They reviewed the failed parts from

the "B" and "C" crankcase explosion and also saw a piston which had lost its tin on the non-thrust side.

Ricardo's initial impression of the blued piston pin from the "C" engine failure was that it was secondary to the piston skirt seizure. After the discovery of the blued pins and piston tin loss on the "A" engine, they agreed with PP&L that higher than normal friction in the piston pin/bushing area could cause the tin loss and ultimately the crankcase explosions.

Ricardo proposed to construct a model of the piston and rods to determine the sensitivity of the piston side loads from high pin friction. PP&L plans to investigate the need for this model.

Ricardo noted that the relief on the piston top (smaller diameter than the skirt) seems generous and they would not expect the rapid loading of the engine to cause the crankcase explosions.

Ricardo suggested that more investigation should be conducted into the piston pin bushing area, i.e., manufacturing lot, dimensions and oil quality. These areas are being pursued by PP&L.

Chapter VI
Attachment 1

A D/G Peak Firing Pressures

Hours	?	573.00	671.30	?	734.80	?
Load (Kw)	4,000.00	4,250.00	4,000.00	4,100.00	4,100.00	4,100.00
Cylinder				PEAK FIRING PRESSURES (PSI)		
1r	1,630.00	1,684.00	1,598.00	1,891.00	1,370.00	1,457.00
2r	1,538.00	1,668.00	1,570.00	1,824.00	1,342.00	1,436.00
3r	1,606.00	1,783.00	1,715.00	1,966.00	1,459.00	1,560.00
4r	1,580.00	1,731.00	1,637.00	1,916.00	1,419.00	1,521.00
5r	1,616.00	1,731.00	1,586.00	1,912.00	1,380.00	1,513.00
6r	1,612.00	1,796.00	1,638.00	1,866.00	1,390.00	1,506.00
7r	1,548.00	1,739.00	1,571.00	1,880.00	1,357.00	1,475.00
8r	1,584.00	1,836.00	1,586.00	1,909.00	1,436.00	1,536.00
1l	1,594.00	1,663.00	1,575.00	1,681.00	1,473.00	1,504.00
2l	1,572.00	1,680.00	1,743.00	1,814.00	1,535.00	1,542.00
3l	1,622.00	1,676.00	1,777.00	1,795.00	1,545.00	1,521.00
4l	1,648.00	1,691.00	1,730.00	1,736.00	1,518.00	1,486.00
5l	1,592.00	1,678.00	1,625.00	1,769.00	1,462.00	1,461.00
6l	1,688.00	1,723.00	1,721.00	1,828.00	1,540.00	1,548.00
7l	1,662.00	1,815.00	1,712.00	1,853.00	1,521.00	1,549.00
8l	1,624.00	1,650.00	1,665.00	1,729.00	1,488.00	1,526.00
Average	1,607.25	1,721.50	1,653.06	1,835.56	1,452.19	1,508.81

B D/G Peak Firing Pressures

B" Diesel Generator Peak Firing Presures (#7158)

Date	5/17/83	12/20/84	10/3/85	12/11/85	5/3/86	11/16/87
Hours	460.00	604.00	706.90	731.00	751.20	?
Load (Kw)	4,000.00	3,800.00	3,800.00	4,000.00	4,000.00	?
Cylinder		Peak Firing Presures (psi)				
1r	1,546.00	1,621.00	1,654.00	1,667.00	1,505.00	1,547.00
2r	1,578.00	1,725.00	1,718.00	1,645.00	1,624.00	1,606.00
3r	1,580.00	1,672.00	1,700.00	1,641.00	1,558.00	1,552.00
4r	1,604.00	1,682.00	1,768.00	1,692.00	1,652.00	1,656.00
5r	1,638.00	1,704.00	1,761.00	1,669.00	1,635.00	1,654.00
6r	1,648.00	1,725.00	1,757.00	1,714.00	1,683.00	1,651.00
7r	1,690.00	1,757.00	1,766.00	1,717.00	1,671.00	1,771.00
8r	1,466.00	1,473.00	1,481.00	1,494.00	1,487.00	1,450.00
1l	1,540.00	1,754.00	1,688.00	1,635.00	1,519.00	1,547.00
2l	1,542.00	1,805.00	1,712.00	1,642.00	1,554.00	1,552.00
3l	1,520.00	1,914.00	1,772.00	1,672.00	1,618.00	1,709.00
4l	1,604.00	1,711.00	1,797.00	1,578.00	1,630.00	1,615.00
5l	1,626.00	1,727.00	1,743.00	1,694.00	1,628.00	1,658.00
6l	1,586.00	1,706.00	1,774.00	1,713.00	1,631.00	1,616.00
7l	1,458.00	1,617.00	1,712.00	1,457.00	1,725.00	1,580.00
8l	1,652.00	1,769.00	1,774.00	1,707.00	1,640.00	1,673.00
Average	1,579.88	1,710.13	1,723.56	1,646.06	1,610.00	1,614.81

B D/G Peak Firing Pressures

B" Diesel

Date	1/1/88		1/23/89
Hours	?		?
Load (Kw)	?		?
Cylinder			
1r	1,441.00	1r	1,508.00
2r	1,612.00	2r	1,612.00
3r	1,522.00	3r	1,516.00
4r	1,665.00	4r	1,652.00
5r	1,635.00	5r	1,606.00
6r	1,647.00	6r	1,589.00
7r	1,663.00	7r	1,684.00
8r	1,464.00	8r	1,439.00
1l	1,557.00	1l	1,603.00
2l	1,406.00	2l	1,470.00
3l	1,671.00	3l	1,688.00
4l	1,655.00	4l	1,727.00
5l	1,682.00	5l	1,774.00
6l	1,651.00	6l	1,676.00
7l	1,577.00	7l	1,623.00
8l	1,674.00	8l	1,643.00
Average	1,595.13		1,613.13

F. J. Czysz

C Diesel Peak Firing Pressures

C" Diesel Generator Peak Firing Presures (#7159)

Date	5/19/83	12/18/84	10/2/85	12/12/85	5/10/86	7/8/87
Hours	312.00	482.00	566.40	586.00	613.70	705.00
Load (Kw)	4,000.00	3,900.00	2,900.00	4,085.00	4,150.00	4,200.00
Cylinder		PEAK FIRING PRESSURES (PSI)				
1r	1,738.00	1,723.00	1,829.00	1,760.00	1,678.00	1,661.00
2r	1,770.00	1,837.00	1,860.00	1,839.00	1,747.00	1,629.00
3r	1,714.00	1,785.00	1,765.00	1,696.00	1,636.00	1,654.00
4r	1,736.00	1,822.00	1,834.00	1,762.00	1,694.00	1,685.00
5r	1,778.00	1,860.00	1,856.00	1,776.00	1,734.00	1,673.00
6r	1,750.00	1,848.00	1,857.00	1,805.00	1,591.00	1,675.00
7r	1,670.00	1,848.00	1,756.00	1,696.00	1,610.00	1,632.00
8r	1,630.00	1,754.00	1,749.00	1,709.00	1,652.00	1,657.00
1l	1,478.00	1,566.00	1,506.00	1,492.00	1,443.00	1,360.00
2l	1,620.00	1,718.00	1,627.00	1,637.00	1,592.00	1,526.00
3l	1,596.00	1,591.00	1,721.00	1,606.00	1,604.00	1,513.00
4l	1,658.00	1,757.00	1,785.00	1,738.00	1,667.00	1,628.00
5l	1,684.00	1,826.00	1,822.00	1,755.00	1,702.00	1,643.00
6l	1,624.00	1,724.00	1,788.00	1,749.00	1,681.00	1,590.00
7l	1,602.00	1,724.00	1,660.00	1,559.00	1,502.00	1,501.00
8l	1,648.00	1,728.00	1,757.00	1,658.00	1,597.00	1,536.00
Average	1,668.50	1,756.94	1,760.75	1,702.31	1,633.13	1,597.69

F. J. Czysz

D D/G Sorted By Peak Firing Pressures

D" Diesel Generator Peak Firing Presures (#7160)

Date		5/18/83		2/6/84		12/21/84		10/4/85
Hours		413.00		536.00		627.00		712.30
Load (Kw)		4,000.00		4,000.00		4,100.00		4,000.00
Cylinder				Peak Firing Pressures (psi)				
	5r	1,656.00	5r	1,810.00	2r	1,855.00	5r	1,784.00
	2r	1,636.00	3r	1,784.00	5r	1,766.00	2l	1,772.00
	3r	1,622.00	6r	1,778.00	1r	1,762.00	1l	1,739.00
	7r	1,616.00	2r	1,772.00	6r	1,746.00	3r	1,738.00
	6r	1,600.00	7r	1,760.00	7r	1,731.00	6r	1,738.00
	4r	1,598.00	4r	1,728.00	#8r	1,730.00	8l	1,733.00
	7l	1,596.00	7l	1,726.00	8l	1,701.00	2r	1,726.00
	#8r	1,590.00	#8r	1,724.00	3r	1,696.00	7r	1,723.00
	8l	1,566.00	1r	1,708.00	4r	1,686.00	5l	1,709.00
	5l	1,558.00	1l	1,704.00	7l	1,680.00	1r	1,707.00
	*2l	1,556.00	5l	1,692.00	6l	1,653.00	7l	1,707.00
	1r	1,546.00	6l	1,692.00	5l	1,640.00	6l	1,706.00
	1l	1,542.00	8l	1,680.00	2l	1,635.00	#8r	1,695.00
	6l	1,540.00	4l	1,650.00	1l	1,632.00	3l	1,687.00
	4l	1,530.00	3l	1,632.00	4l	1,579.00	4r	1,678.00
	3l	1,518.00	2l	1,618.00	3l	1,560.00	4l	1,678.00
==========		==========		==========		==========		==========
Average		1,579.38		1,716.13		1,690.75		1,720.00

Piston with tin loss
* Piston involved crankcase explosion 1/14/84

D D/G Sorted By Peak Firing Pressures

D" Diesel

Date		12/12/85		5/17/86		8/3/87		1/23/89
Hours		737.80		761.40		847.00		?
Load (Kw)		4,085.00		4,000.00		4,000.00		?
Cylinder								
	5r	1,740.00	5r	1,614.00	3r	1,631.00	7l	1,776.00
	7l	1,717.00	6r	1,575.00	6r	1,568.00	3r	1,728.00
	6r	1,707.00	8l	1,573.00	5r	1,565.00	2r	1,726.00
	2r	1,706.00	7r	1,552.00	2r	1,564.00	6r	1,714.00
	#8r	1,701.00	3r	1,549.00	4l	1,529.00	8l	1,706.00
	3r	1,699.00	2r	1,545.00	#8r	1,525.00	5r	1,694.00
	7r	1,694.00	6l	1,537.00	7l	1,523.00	5l	1,684.00
	8l	1,691.00	5l	1,534.00	7r	1,521.00	4r	1,675.00
	2l	1,671.00	7l	1,530.00	1l	1,520.00	6l	1,673.00
	1l	1,665.00	1r	1,523.00	8l	1,517.00	#8r	1,669.00
	4r	1,661.00	2l	1,517.00	6l	1,510.00	7r	1,655.00
	1r	1,652.00	1l	1,516.00	1r	1,509.00	3l	1,625.00
	6l	1,652.00	4r	1,513.00	5l	1,507.00	4l	1,609.00
	3l	1,647.00	3l	1,506.00	4r	1,505.00	1l	1,587.00
	5l	1,647.00	#8r	?	3l	1,482.00	1r	1,469.00
	4l	1,607.00	4l	?	2l	1,477.00	2l	1,432.00
==========		==========		==========		==========		==========
Average		1,678.56		1,541.71		1,528.31		1,651.38

F. J. Czysz

" Diesel Generator Peak Firing Presures (#7157)

Date	5/20/83	12/17/84	10/1/85	12/10/85
Hours	?	573.00	671.30	?
Load (Kw)	4,000.00	4,250.00	4,000.00	4,100.00
Cylinder	PEAK FIRING PRESSURES (PSI)			
	61 1,688.00	8r 1,836.00	31 1,777.00	3r 1,966.00
	71 1,662.00	71 1,815.00	21 1,743.00	4r 1,916.00
	41 1,648.00	6r 1,796.00	41 1,730.00	5r 1,912.00
	#1r 1,630.00	3r 1,783.00	61 1,721.00	8r 1,909.00
	81 1,624.00	#7r 1,739.00	3r 1,715.00	#1r 1,891.00
	31 1,622.00	4r 1,731.00	71 1,712.00	#7r 1,880.00
	5r 1,616.00	5r 1,731.00	81 1,665.00	6r 1,866.00
	6r 1,612.00	61 1,723.00	6r 1,638.00	71 1,853.00
	3r 1,606.00	41 1,691.00	4r 1,637.00	61 1,828.00
	11 1,594.00	#1r 1,684.00	51 1,625.00	#2r 1,824.00
	51 1,592.00	21 1,680.00	#1r 1,598.00	21 1,814.00
	8r 1,584.00	51 1,678.00	5r 1,586.00	31 1,795.00
	4r 1,580.00	31 1,676.00	8r 1,586.00	51 1,769.00
	21 1,572.00	#2r 1,668.00	11 1,575.00	41 1,736.00
	#7r 1,548.00	11 1,663.00	#7r 1,571.00	81 1,729.00
	#2r 1,538.00	81 1,650.00	#2r 1,570.00	11 1,681.00
==========	==========	==========	==========	==========
Average	1,607.25	1,721.50	1,653.06	1,835.56

Piston with tin loss
* Piston involved in crankcase explosion

A D/G Sorted By Firing Pressures

A" Diesel

Date	6/26/86	7/31/86
Hours	734.80	?
Load (Kw)	4,100.00	4,100.00
Cylinder		
	31 1,545.00	3r 1,560.00
	61 1,540.00	71 1,549.00
	21 1,535.00	61 1,548.00
	71 1,521.00	21 1,542.00
	41 1,518.00	8r 1,536.00
	81 1,488.00	81 1,526.00
	11 1,473.00	4r 1,521.00
	51 1,462.00	31 1,521.00
	3r 1,459.00	5r 1,513.00
	8r 1,436.00	6r 1,506.00
	4r 1,419.00	11 1,504.00
	6r 1,390.00	41 1,486.00
	5r 1,380.00	#7r 1,475.00
	#1r 1,370.00	51 1,461.00
	#7r 1,357.00	#1r 1,457.00
	#2r 1,342.00	#2r 1,436.00
==========	==========	==========
verage	1,452.19	1,508.81

B D/G Sorted By Firing Pressure

B" Diesel Generator Peak Firing Presures (#SORTED BY FIRING PRESSURE

Date	5/17/83	12/20/84	10/3/85	12/11/85
Hours	460.00	604.00	706.90	731.00
Load (Kw)	4,000.00	3,800.00	3,800.00	4,000.00
Cylinder				
	7r 1,690.00	3l 1,914.00	4l 1,797.00	7r 1,717.00
	8l 1,652.00	2l 1,805.00	6l 1,774.00	6r 1,714.00
	6r 1,648.00	8l 1,769.00	8l 1,774.00	6l 1,713.00
	5r 1,638.00	7r 1,757.00	3l 1,772.00	8l 1,707.00
	5l 1,626.00	1l 1,754.00	4r 1,768.00	5l 1,694.00
	4r 1,604.00	5l 1,727.00	7r 1,766.00	4r 1,692.00
	4l 1,604.00	2r 1,725.00	5r 1,761.00	3l 1,672.00
	6l 1,586.00	6r 1,725.00	6r 1,757.00	5r 1,669.00
	3r 1,580.00	4l 1,711.00	5l 1,743.00	#1r 1,667.00
	2r 1,578.00	6l 1,706.00	2r 1,718.00	2r 1,645.00
	#1r 1,546.00	5r 1,704.00	2l 1,712.00	2l 1,642.00
	2l 1,542.00	4r 1,682.00	*7l 1,712.00	3r 1,641.00
	1l 1,540.00	3r 1,672.00	3r 1,700.00	1l 1,635.00
	3l 1,520.00	#1r 1,621.00	1l 1,688.00	4l 1,578.00
	8r 1,466.00	*7l 1,617.00	#1r 1,654.00	8r 1,494.00
	*7l 1,458.00	8r 1,473.00	8r 1,481.00	*7l 1,457.00
==========	==========	==========	==========	==========
Average	1,579.88	1,710.13	1,723.56	1,646.06

Piston with tin loss
* Piston involved in crankcase explosion

B" Diesel

Date	5/3/86	11/16/87	1/1/88	1/23/89
Hours	751.20	?	?	?
Load (Kw)	4,000.00	?	?	?
Cylinder				
	*7l 1,725.00	7r 1,771.00	5l 1,682.00	5l 1,774.00
	6r 1,683.00	3l 1,709.00	8l 1,674.00	4l 1,727.00
	7r 1,671.00	8l 1,673.00	3l 1,671.00	3l 1,688.00
	4r 1,652.00	5l 1,658.00	4r 1,665.00	7r 1,684.00
	8l 1,640.00	4r 1,656.00	7r 1,663.00	6l 1,676.00
	5r 1,635.00	5r 1,654.00	4l 1,655.00	4r 1,652.00
	6l 1,631.00	6r 1,651.00	6l 1,651.00	8l 1,643.00
	4l 1,630.00	6l 1,616.00	6r 1,647.00	*7l 1,623.00
	5l 1,628.00	4l 1,615.00	5r 1,635.00	2r 1,612.00
	2r 1,624.00	2r 1,606.00	2r 1,612.00	5r 1,606.00
	3l 1,618.00	*7l 1,580.00	*7l 1,577.00	1l 1,603.00
	3r 1,558.00	3r 1,552.00	1l 1,557.00	6r 1,589.00
	2l 1,554.00	2l 1,552.00	3r 1,522.00	3r 1,516.00
	1l 1,519.00	#1r 1,547.00	8r 1,464.00	#1r 1,508.00
	#1r 1,505.00	1l 1,547.00	#1r 1,441.00	2l 1,470.00
	8r 1,487.00	8r 1,450.00	2l 1,406.00	8r 1,439.00
==========	==========	==========	==========	==========
Average	1,610.00	1,614.81	1,595.13	1,613.13

F. J. Czysz

C" Diesel Generator-Sorted By Peak Firing Presures (#7159)

Date	5/19/83	12/18/84	10/2/85	12/12/85
Hours	312.00	482.00	566.40	586.00
Load (Kw)	4,000.00	3,900.00	2,900.00	4,085.00
Cylinder				
	*5r 1,778.00	*5r 1,860.00	2r 1,860.00	2r 1,839.00
	2r 1,770.00	6r 1,848.00	6r 1,857.00	6r 1,805.00
	6r 1,750.00	7r 1,848.00	*5r 1,856.00	*5r 1,776.00
	1r 1,738.00	2r 1,837.00	4r 1,834.00	4r 1,762.00
	4r 1,736.00	5l 1,826.00	1r 1,829.00	1r 1,760.00
	3r 1,714.00	4r 1,822.00	5l 1,822.00	5l 1,755.00
	5l 1,684.00	3r 1,785.00	#6l 1,788.00	#6l 1,749.00
	7r 1,670.00	4l 1,757.00	4l 1,785.00	4l 1,738.00
	4l 1,658.00	8r 1,754.00	3r 1,765.00	8r 1,709.00
	8l 1,648.00	8l 1,728.00	8l 1,757.00	3r 1,696.00
	8r 1,630.00	#6l 1,724.00	7r 1,756.00	7r 1,696.00
	#6l 1,624.00	7l 1,724.00	8r 1,749.00	8l 1,658.00
	2l 1,620.00	1r 1,723.00	3l 1,721.00	2l 1,637.00
	7l 1,602.00	2l 1,718.00	7l 1,660.00	3l 1,606.00
	3l 1,596.00	3l 1,591.00	2l 1,627.00	7l 1,559.00
	1l 1,478.00	1l 1,566.00	1l 1,506.00	1l 1,492.00
==========	==========	==========	==========	==========
Average	1,668.50	1,756.94	1,760.75	1,702.31

Piston with tin loa

* Piston involved in crankcase explosion

C" Diesel

Date	5/10/86	7/8/87
Hours	613.70	705.00
Load (Kw)	4,150.00	4,200.00
Cylinder		
	2r 1,747.00	4r 1,685.00
	*5r 1,734.00	6r 1,675.00
	5l 1,702.00	*5r 1,673.00
	4r 1,694.00	1r 1,661.00
	#6l 1,681.00	8r 1,657.00
	1r 1,678.00	3r 1,654.00
	4l 1,667.00	5l 1,643.00
	8r 1,652.00	7r 1,632.00
	3r 1,636.00	2r 1,629.00
	7r 1,610.00	4l 1,628.00
	3l 1,604.00	#6l 1,590.00
	8l 1,597.00	8l 1,536.00
	2l 1,592.00	2l 1,526.00
	6r 1,591.00	3l 1,513.00
	7l 1,502.00	7l 1,501.00
	1l 1,443.00	1l 1,360.00
==========	==========	==========
Average	1,633.13	1,597.69

F. J. Czysz

D D/G Sorted By Peak Firing Pressures

D" Diesel Generator Peak Firing Presures (#7160)

Date	5/18/83	2/6/84	12/21/84	10/4/85
Hours	413.00	536.00	627.00	712.30
Load (Kw)	4,000.00	4,000.00	4,100.00	4,000.00
Cylinder		Peak Firing Pressures (psi)		
	5r 1,656.00	5r 1,810.00	2r 1,855.00	5r 1,784.00
	2r 1,636.00	3r 1,784.00	5r 1,766.00	2l 1,772.00
	3r 1,622.00	6r 1,778.00	1r 1,762.00	1l 1,739.00
	7r 1,616.00	2r 1,772.00	6r 1,746.00	3r 1,738.00
	6r 1,600.00	7r 1,760.00	7r 1,731.00	6r 1,738.00
	4r 1,598.00	4r 1,728.00	#8r 1,730.00	8l 1,733.00
	7l 1,596.00	7l 1,726.00	8l 1,701.00	2r 1,726.00
	#8r 1,590.00	#8r 1,724.00	3r 1,696.00	7r 1,723.00
	8l 1,566.00	1r 1,708.00	4r 1,686.00	5l 1,709.00
	5l 1,558.00	1l 1,704.00	7l 1,680.00	1r 1,707.00
	*2l 1,556.00	5l 1,692.00	6l 1,653.00	7l 1,707.00
	1r 1,546.00	6l 1,692.00	5l 1,640.00	6l 1,706.00
	1l 1,542.00	8l 1,680.00	2l 1,635.00	#8r 1,695.00
	6l 1,540.00	4l 1,650.00	1l 1,632.00	3l 1,687.00
	4l 1,530.00	3l 1,632.00	4l 1,579.00	4r 1,678.00
	3l 1,518.00	2l 1,618.00	3l 1,560.00	4l 1,678.00
==========	==========	==========	==========	==========
Average	1,579.38	1,716.13	1,690.75	1,720.00

Piston with tin loss
* Piston involved crankcase explosion 1/14/84

D D/G Sorted By Peak Firing Pressures

D" Diesel

Date	12/12/85	5/17/86	8/3/87	1/23/89
Hours	737.80	761.40	847.00	?
Load (Kw)	4,085.00	4,000.00	4,000.00	?
Cylinder				
	5r 1,740.00	5r 1,614.00	3r 1,631.00	7l 1,776.00
	7l 1,717.00	6r 1,575.00	6r 1,568.00	3r 1,728.00
	6r 1,707.00	8l 1,573.00	5r 1,565.00	2r 1,726.00
	2r 1,706.00	7r 1,552.00	2r 1,564.00	6r 1,714.00
	#8r 1,701.00	3r 1,549.00	4l 1,529.00	8l 1,706.00
	3r 1,699.00	2r 1,545.00	#8r 1,525.00	5r 1,694.00
	7r 1,694.00	6l 1,537.00	7l 1,523.00	5l 1,684.00
	8l 1,691.00	5l 1,534.00	7r 1,521.00	4r 1,675.00
	2l 1,671.00	7l 1,530.00	1l 1,520.00	6l 1,673.00
	1l 1,665.00	1r 1,523.00	8l 1,517.00	#8r 1,669.00
	4r 1,661.00	2l 1,517.00	6l 1,510.00	7r 1,655.00
	1r 1,652.00	1l 1,516.00	1r 1,509.00	3l 1,625.00
	6l 1,652.00	4r 1,513.00	5l 1,507.00	4l 1,609.00
	3l 1,647.00	3l 1,506.00	4r 1,505.00	1l 1,587.00
	5l 1,647.00	#8r ?	3l 1,482.00	1r 1,469.00
	4l 1,607.00	4l ?	2l 1,477.00	2l 1,432.00
==========	==========	==========	==========	==========
Average	1,678.56	1,541.71	1,528.31	1,651.38

F. J. Czysz

COOPER-BESSEMER RECIPROCATING

October 9, 1989

Our Ref: QCG-6392

Mr. Frank Czysz
Pennsylvania Power and Light
2 North Ninth Street
Mailstop Bldg. A6-3
Allentown, PA 18101

Dear Mr. Czysz:

This is further to our telephone discussion regarding a possible relationship between the recent problems and the engine rating. Although many aspects of the engine are load sensitive, it is our considered opinion, based upon experience with diesel engines, that there is no direct relationship between load and piston problems as it relates to the root cause. The following is provided for your information:

1. For continuous operation (24 hours per day) the engine would be rated at 200 BMEP or 4130 KW.

2. The basic engine rating as sold to P. P. & L. is 210 BMEP at 4300 KW, however:

3. The actual rating as sold allows the engine to run two hours every twenty-four at 227 BMEP or 4700 KW, and we consider this to be well within the design limits of the engines. Please note that we have successfully run a similar engine, but of older design, continuously for 500 hours at 250 BMEP or 5176 KW.

Very truly yours,

W. H. A. Lambert
Manager, Q.C.

cc: M. J. Helmich
T. W. Kearns
R. A. Miklos
M. A. Schleigh
File: SO-0188/P2

WHAL/kll

Lincoln Avenue
Grove City, Pennsylvania 16127
(412) 458-8000 Telex: 499-7257 A/B CBCORPGRCT

FORM 133

COOPER-BESSEMER RECIPROCATING

INTER - OFFICE MEMO

DATE 11/7/89

TO A. E. Bice
D. T. Blizzard
W. O. Ferguson
B. C. Guntrum - C-E
B. K. Hall
M. J. Helmich
J. M. Horne
T. W. Kearns
T. Leishman - MV
R. A. Miklos
M. A. Schleigh
B. R. Sedelmyer
F. B. Stolba

FROM W. H. A. Lambert

SUBJECT Pennsylvania Power and Light Company
Standby Diesel Generators
Comments on Recent Crankcase Explosions

Our Ref: QCG-6487

Interim "comments" regarding events at the Susequehana Steam Electric Station (SSES) have been requested and they are being transmitted herewith in report form (ref. QCG-6420). It should be clearly understood these comments reflect observations pertinent at the time of writing and do not represent conclusions with respect to the root causes of recent crankcase explosions. Two examples will serve to illustrate the point:

- Event No.1 (see para. 3.1 of QCG-6420) - Probable cause was dirt in the piston pin. Looseness of the piston pin bolts most likely occurred when the piston assembly seized and the bolts subsequently stretched.

- Event No. 3 (see para. 3.1 of QCG-6420) - At the time of writing a casting defect was considered to be the initiation of the piston/liner seizure. Subsequent observation revealed excessive wear of the compressor rings. This unusual wear is now the focus of the root cause investigation.

Never-the-less, the dicussion regarding engine loading is appropriate and requires serious consideration by all parties involved.

A meeting was held in Grove City on November 3, 1989 with P. P. & L. representatives to discuss the evaluation of problems at SSES and their resolution.

W. H. A. Lambert
Manager, Quality Assurance
and Nuclear Operations

WHAL/bjb

Enclosure

cc: F. J. Czyzs - PP&L
W. Haass - NRC
E. Tomlinson - NRC
File: 5-1
SO-0188/P2
K5fa17

REPORT

Pennsylvania Power and Light Company
Standby Diesel/Generator
Comments on Recent Crankcase Explosions

1.0 Introduction

1.1 Two recent crankcase explosions, one September 16, 1989 and the other on October 7, 1989, have been regarded by the Operator as a common mode failure. Consequently, serious questions have arisen concerning the engines load carrying capability with an adequate margin of safety.

1.2 These recent, and earlier events as well as various aspects of KSV engine operation and maintenance were discussed both with P. P. & L. and NRC personnel on October 12 and 13, 1989 in particular, and on previous occasions. Therefore, to the extent that is appropriate these "comments" may be regarded as a trip report. The "comments" contained herein represent an overview of specific happenings at P. P.& L. and the operation of the KSV engine in general.

1.3 The compliment of emergency standby diesel/generators installed at the Susquehanna Steam Electric Station (SSES) are as follows:

Engine	Unit	S/N	Initial Start	Up to October 17, 1989 Total Hours	No. of Starts
KSV-16-T	A	7158	1981	981.6	542
KSV-16-T	B	7159	Jan/82	1033.1	537
KSV-16-T	C	7160	1981	796.9	526
KSV-16-T	D	7161	July/81	965.5	531
KSV-20-T	E	7218	May/87	86.7	76

1.4 Attached as Appendix "A" is a copy of a handwritten commentary regarding the October 7, 1989 event and left at SSES on October 13, 1989.

1.5 It should be clearly understood that this document is by no means a final summary or conclusion as to the cause or causes of recent events at SSES. The reviewing and analysis of various facets is an on-going process. A final report will be prepared at the appropriate time.

2.0 Philosophical Comments

2.1 Any engine applied to the emergency standby requirements of a nuclear power plant is subject to unique operation parameters. Industry experience is based upon base load operational (approx. 8000 hours operation per year) or peak lopping (daily or frequent starting and stopping during which an engine will be required to run for several hours).

2.2 Whereas the KSV engine has been appropriately applied to the nuclear industry, the long term effects on major components resulting from frequent starts, fast loading and short runs are not known. Consequently, there has to be a very close relationship between operator and manufacturer (industry wide) in order to keep abreast of lessons learned.

2.3 It is well understood that in order to meet the specified requirements of LOCA or LOOP events, an emergency diesel/generator must be capable of a rapid start and rapid acceptance of load. The KSV has demonstrated such capability. There are, however, technical specification requirements (see para. 9.0) which, in the interest of demonstrated reliability, take an unknown element of longevity out of the engine.

2.4 Of concern to Cooper-Bessemer (C-B) is not the fast start, but the rapid loading, again with respect to longevity. Of paramount consideration are the thermal stresses induced in the combustion zone components during a transient engine load condition. Consequently, C-B recommends an evaluation of surveillance procedures with respect to rapid loading, sequence of loading, duration of tests, and in particular, the "Break-In" of an engine following replacement of running gear components.

2.5 With respect to operational experience in general and recent events at P. P. and L., it is clearly understood that positive measures must be taken to increase the reliability of the diesel/generator system - a continuous process.

3.0 Crankcase Explosions at P.P & L.

3.1 The following five events have occurred:

Event No.	Unit	Event	Date
1	B	Loose piston pin bolt 5L. Due to looseness of pin on rod oil "spilled" through pin to rod clearance thereby precluding adequate piston cooling.	Jan. 18, 1986
2	B	Piston skirt distress 7L. Unknown source of material "rolled" between piston and liner.	Sept. 16, 1989
3	C	Piston skirt distress 5R. Defect in piston skirt - unique.	Oct. 7, 1989
4	D	Lube oil pump bearing failure on initial start-up.	Nov. 29, 1981
5	D	Fuel oil dilution of lube oil in cylinder 2L caused high friction between piston and liner.	Jan. 14, 1984

3.2 The only similarity between events 2, 3, and 5 (piston/liner failures) is in fact that the piston in each case ultimately "seized" in its liner. The primary, or root cause, in each event was different.

3.3 Lubrication, particularly during a load transient, is very important to successful engine operation. If lubrication, for any reason is "removed", then heat will be generated, and if the condition causing loss of lubrication is not corrected (self corrected), then heat generated will lead to a point of incandescence. This, of course, becomes the source of ignition for a crankcase explosion.

3.4 As a consequence of the observations made we currently see no common thread between the root causes of these events. Commonality is only present in the manifestation of the effects of these events.

3.5 It should be clearly understood that for each of the circumstances which led to the five crankcase explosions, engine load per se and, therefore, the engine load bearing capability was not a factor. In the case of the three piston/liner failures, therefore, motion aggravated by load was the significant element.

4.0 Engine Rating - Nominal

4.1 For the purpose of discussing nominal engine rating BMEP (brake mean effective pressure) as a measure of engine power output only will be considered. The actual rating will be discussed in paragraph 5.0.

4.2 As sold to P. P. and L. the continuous rating is 194.9 BMEP
10% Overload 214.4 BMEP
18% Overload (4700KW) 227.0 BMEP
Maximum safe load 250.0 BMEP

Continuous rating is 77.96% of max. safe load.
10% Overload is 85.76% of max. safe load.
118% Overload is 90.80% of max. safe load.

4.3 The maximum safe load is based upon not exceeding a peak firing pressure of 1600 PSI.

4.4 The maximum safe load has been confirmed by actual engine testing up to the limit of 250 BMEP. Approximately 500 hours of mostly continuous running was performed on one of the Commonwealth Edison, Zion engines. Subsequent to this testing and actual field experience it was determined that the original "gas/diesel" cylinder head was not suitable for a straight diesel application (heads cracked on the fire dome side). Consequently, a new "diesel" head was designed.

4.5 Five engines with the gas/diesel head were supplied to CECo, Zion and two to Nebraska Public Power District, Cooper Nuclear Station. These may be regarded as the "first" generation nuclear service diesel engines.

4.6 A "second" generation was developed. Whereas the first generation were built prior to the imposition of 10 CFR 50 App. "B", the second generation complied fully with it. In addition to the new design of head, the camshaft was redesigned increasing its diameter to 3-1/2" from 3". Other than changes made to accommodate the new camshaft and changes internal to the cylinder head, no other structural changes were made between the first and second generation engines. No changes were made in the running gear and no changes made in significant clearances such as piston to liner, piston and articulated pin to bushing and crankshaft bearings.

4.7 The P. P. and L. "A" engine was the first of the second generation engines to be built and was extensively tested prior to shipment. This engine was tested up to 250 BMEP in order to confirm structural integrity, to obtain heat rejection data, air flows, and to determine fuel system characteristics to establish compatibility between fuel consumption and peak firing pressures. In addition, compatibility of data obtained was cross-checked with a test engine (KSV) in the C-B R and D Laboratory.

4.8 It should be noted that the limiting factor to load carrying ability of the engines installed at P. P. and L. is heat rejection of the jacket water cooling system. As designed, this system has more than adequate capacity for the P. P. and L. rated condition.

5.0 Engine Rating - P.P. & L. KSV-16-T Engines

5.1 In relation to other installations, the 4700KW load, which is 18% over the continuous rating, appears to be an anomaly. Usually the specified and designed overload is 10%. The effect of the 4700KW was evaluated and the engine and its supporting systems were designed to accommodate it.

5.2 Paragraph 5.3 lists various documents attached as Appendix "B". They are summarized as follows:

5.2.1 The original specification called for 4000KW continuous rating load and left the overload "open".

5.2.2 System characteristics would create a peak load requirement of 4700KW.

5.2.3 Initially, it appeared that an increase of WR^2 of the flywheel would prohibit start in 10 seconds - ultimately not a problem.

5.2.4 The generator design accommodates the 4700KW load.

5.2.5 Exhaust and cooling systems resized for the 4700KW load.

5.2.6 Revision 5 of specification issued to cover 4700KW requirement.

5.2.7 Diesel/generator supplied in compliance with the specification.

5.3. Appendix "B" comprises:

5.3.1 Pages 1 and 11 of Specification 8856-M-30 Rev. 0.

5.3.2 Letter dated October 15, 1973 from C-B to Bechtel.

5.3.3 Letter dated October 19, 1973 from E.P. Portec, Inc., to C-B.

5.3.4 Letter dated February 5, 1974 from C-B to Bechtel.

5.3.5 Meeting notes of 7/12/74.

5.3.6 Letter dated July 30, 1974 from Bechtel to C-B with Pages 1 and 10 of attachment. See "Comments - General" on Page 10.

5.3.7 Pages 1 and 11 of Specification 8856-M-30 Rev. 1.

5.3.8 Pages 1, 10, and 11 of specification 8856-M-30 Rev. 5.

5.4 The diesel/generator rating is, therefore, confirmed to be 4000KW continuous at 105°F ambient-temperature with a 2000-hour overload rating of 4700KW also at 105°F ambient temperature.

6.0 Engine Running - Post Maintenance

6.1 Whenever items in the running gear such as piston rings, bushings, bearing, etc. are replaced it is necessary to gradually expose them to the operational environment in order to ensure optimum conformity of mating components.

6.2 The practice at SSES, following such maintenance, has been to start the engine, and shortly thereafter load to 100% or higher. Such operation is not conducive to establishing conformity (break-in) to ensure long term reliable operation and consequently availability. This matter was the subject of a great deal of discussion which was concluded by issuance of an internal P.P. & L. memorandum dated October 13, 1989, a copy of which is attached as Appendix "C".

6.3 It should be noted that the 4-hour run at 600 RPM unloaded has been shortened due to the heat generated in the H.P. fuel pumps because of by-passing (internally) a large quantity of fuel oil at no load operation. It was agreed that the balance of the 4 hour no load run would be added to the 25% load run.

7.0 Engine Running and Inspection - Unit "B"

7.1 Following the rebuild of Unit "C" it was agreed to run the engine in accordance with the "break-in" schedule per Paragraph 5.2 and 6.3 above. This break-in would preceed a 24-hour surveillance run as required by SSES technical specification.

7.2 Upon completion of this 36-hour (plus) run, it was agreed that a visual inspection within the crankcase would be made in order to primarily assess the conditions of piston skirts, liners and pins insofar as they could be seen without disassembly.

7.3 Acceptance criteria was developed and is outlined in P.P. & L. internal memo dated October 15, 1989, a copy of which is attached as Appendix "D"". This criteria is understood to be a guideline subject to change based upon subsequent experience.

7.4 The inspection of Unit "C" indicated acceptable engine condition.

8.0 <u>Inspection - Unit "B"</u> 1

8.1 The inspection of Unit "B" commenced on October 23, 1989 and four piston/liners did not meet the Appendix "D" criteria. The piston/liner which was the furthest outside the referenced criteria was removed from the 4 right location and has been received at the Cooper-Bessemer, Grove City facility for further evaluation. In addition, it was determined that a visual observation of piston pin "color" was misleading due to the reflectivity of the polished pin surface.

8.2 During the inspection it was observed that in 12 cylinders the piston pin caps had contacted the liner surface. The same phenomena had previously been witnessed, but not to the extent seen in the "B" unit; and also has not been known to be destructive, although it may have been a contributory factor with respect to events at SSES. (See also Paragraph 11.2).

9.0 <u>Engine Running - Surveillance Testing</u>

9.1 IEEE Std. 387-1977 defines continuous and short term ratings and C-B has provided equipment compatible with these ratings. Specifically, it is required that the Diesel/Generator demonstrate full load capability for an interval of not less than 24 hours of which 22 hours should be at a load equivalent to the continuous rating and 2 hours at the specified overload.

9.2 As interpretted at SSES (per Reg. Guide 1.108) the 24-hour surveillance run requires the engine to be started, loaded immediately for the first two hours at 4700KW and the last 22 hours at 4000KW. The purpose behind this test is to demonstrate the reliability of the engine to perform in the event of an emergency. The reality is that such a test procedure reduces the diesel/generator reliability. This is because starting and loading to the overload condition without first stabilizing engine temperatures (structural and running gear members) accelerates wear rates and, therefore, reduces the engine capability to accept and recover from distress.

9.3 We presume that the number of 24-hour surveillance tests at many different utilities using different engine manufacturers equipment has demonstrated a basic capability of this equipment to respond to an emergency. Therefore, C-B recommends that the 24 hour surveillance test be run as follows:

Start engine
15 min. at no load
1 hour at 25% load
1/2 hour at 50% load
1/2 hour at 75% load
22 hour at 100% load
2 hour at overload (4700KW for SSES)

9.4 Reg. Guide 1.108 also requires a diesel/generator to demonstrate full-load carrying capability at continuous rating for an interval of not less than one hour with no more than 31 days between test periods. At SSES this test run is completed within the one hour time period which in fact is not enhancing engine reliability. For long term life/reliability, C-B recommends that a loaded engine run should not be less than four hours. Such a duration will permit thermal stability within the engine components prior to shutdown.

10.0 Lube Oil

10.1 Gulf Superduty 40 H.D. is the lube oil being used in the KSV engines at SSES. Provided that the oil supplied complies with tlhe data sheets for this CD-SF type oil it meets C-B requirements. It should be noted the B-P has acquired the Gulf Oil business and should be issuing their own data sheets equivalent to the Gulf product.

11.0 Further Evaluation

11.1 An enigma surrounds the application of the lower oil control ring. The purpose for this ring is, as its description implies, oil control. Although this ring has been applied since the original design of over thirty years ago, there is some question as to its effectiveness in an engine which does not run continously. Consequently an evaluation is being carried out to determine the advisability of the permanent removal of the lower oil control ring.

11.2 The piston pin end caps are assembled with an interference fit, and once assembled should blend with the piston contour and therefore should not exhibt noticeably unique liner contact. Yet such contact has been observed (see para. 8.2) at SSES and insofar as unit "B" is concerned, is extensive. This has given rise to speculation that "debris" from the rubbing of the piston pin cap may be trapped by the lower oil control ring and thereby adversly affect the piston to geometry during a load change. It should be noted that the thermal transients which occur during a load change, "upset" the piston shape which relaxes when load/thermal stability is achieved - a situation which is aggravated by extremely rapid load changes such as those described in paragraph's 6.0 and 9.0. Therefore, and in conjunction with evaluating the effectiveness of the lower oil control ring, the application of the piston pin end cap is being reviewed.

11.3 Transference of tin plate from the piston skirt to the liner surface, most noticeably on the non-thrust side, has been observed on both the "C" and "B" units at SSES, (see paragraph 7.0 and 8.0) The No. 1 right cylinder and liner from Unit "B", which is currently at the Grove City facility of C-B, will be thoroughly examined to determine the root cause of this phenomena, and will be considered in relation to the anomalies observed with respect to the piston pin cap and lower oil control ring. Plans for such an evaluation are currently being formulated.

11.4 In summary, it is necessary to complete the evaluation outlined above, and in conjuction with all Utilities maintain adequate communication to ensure enhancement of diesel/generator reliability.

W. H. A. Lambert
Manager, Quality Assurance
and Nuclear Operations

cc: R. J. Brager
A. E. Bice
D. T. Blizzard
F. J. Czyzs - PP&L
W. O. Ferguson
B. C. Guntrum - C-E
B. K. Hall
W. Haass - NRC
M. J. Helmich
J. M. Horne
T. W. Kearns
W. Leishman - MV
R. A. Miklos
M. A. Schleigh
B. R. Sedelmyer
F. B. Stolba
Ed Tomlinson - NRC
File: 5-1
SO-0188/P2
K5fa17

WHAL/kll

Oct. 13-89

P.P.&L. KSV-16-T
Unit "C" S.N. 7160
No. 5 Right Piston
Failure - Evaluation

The Failed Components Removed From The Engine Following A Crankcase Explosion On October 7, 1989 Have Been Inspected And The Circumstances Reviewed. As A Result It Is Concluded That The Area Of The Primary Failure Was The Piston Skirt And The Probable Cause Was Defects In The Lower Portion Of The Piston Skirt. The Reasons Are Summarized As Follows:

1. The Liner Was Heavily Damaged (Burned And Heat Checked) In The Lower Half And For Most Of Its Circumference.

2. The Piston Skirt Was Extensively "Worn" Consistent With The Pattern Of Distress Observed On The Liner.

3. Two "Blow Holes" Were Observed At The Bottom Of The Piston Skirt With Cracks Running Down From One Of Them. As Machined These Were Subsurface Defects. (Such ~~Con~~ "Blow Holes" If Found On The Surface Are Cause For Rejection). A "Gouge" Adjacent To The Larger Of The Two "Blow" Holes Was Caused By Debris Presumably From The Defective Area. Debris Trapped Between The Piston And Liner Would Have "Rubbed And Rolled" Causing A Rise In Temperature Swelling The Piston Causing It To Tend To Seize - A Self Destructing Process

4. The Failure Of The Piston Pin And Bush Is Considered To Be A Secondary Failure. As The Piston Swelled The Rod On The Upstroke Would Have Pushed The Piston Upwards Thereby Severely Overloading The Top or "Back" Portion Of The Piston Pin Bushing.

[signature]
Cooper - Bessemer
Q.A. Manager

APPENDIX B

Q

BECHTEL

MATERIAL REQUISITION

MATERIAL: DIESEL GENERATORS

SHOP INSPECTION ☒ IS ☐ IS NOT REQUIR

COST CODE: 4.095

JOB SITE DELIVERY DATE: Aug. 1, 1975
Units 1, 2

M/R STATUS

REV NO	DATE	REVISIONS	ORIGINATOR (NAME)	APPROVALS	T/NO
A	11/4/	Issued for Client Approval	E. Hanna		
0	6/28/73	Issued for Bids	E. Hanna		
1	8/24/73	Revised to add Addendums 1 and 2	E. Hanna		

Job	Requisition No.	Rev
8856-M-30		1

5.2 **Service Conditions**

5.2.1 The standby diesel generators shall be designed for the following conditions:

Number of units required: 4

Starting time to rated speed and voltage:	10 seconds max.
Rating per unit: Continuous	4000 KW min.
Overload (2000 hours)	= Mfg. to State
7050 Overload (200 hours)	= Mfg. to State
Overload (30 mins.)	= Mfg. to State
Overload (10 seconds)	= Mfg. to State
Power factor, lagging	0.8
Frequency	60.0 Hertz
Voltage	4160 V
Phase	3 phase, wye
Neutral connection	External
Overload capacity (2 hours in any 24 hours)	10.0 percent
Service	Standby Emergency
Speed, not more than	1200 rpm
Largest motor HP to be started	2000 HP
Elevation at site	676 ft

5.2.2 **Design ambient Temperature at Diesel Generator Room:**

a) Engine off (Min/Max) 72°/104°F

b) Engine Running:
Max. At Generator 120°F
At Engine 160°F

c) Outside Air Temperature
Dry-bulb air temperature, minimum/maximum -19°F/105°F

Wet-bulb air temperature, minimum/maximum -13°F/82°F

LETS TACK

RECEIVED
OCT 17 1973
MT. VERNON, OHIO

October 15, 1973

Bechtel Corporation
P. O. Box 3965
San Francisco, California 94119

Attention: Mr. R. C. Higgins, Senior Buyer

Subject: Pennsylvania Power & Light/Susquehanna
Your Bid Request No. 8856-M-30
Our VNP-4-W-MJ-73-3-024

Gentlemen:

Referring to my September 28, 1973 letter, page 2, I submitted higher ratings for the Model KSV-16 engine. It was mentioned that three (3) Coolers and the Exhaust Silencer must be re-sized and re-priced to get these ratings. Listed below are the increases in price. These prices are for the four (4) Diesel Generator Units , and to be added to the contract price given August 31, 1973.

A- 8760 Hr. 4000 KW
2000 Hr. 4700 KW
200 HR 5050 KW
120 Hr.
or less 5100 KW

PRICE: SIXTEEN THOUSAND, EIGHT DOLLARS ------------------ $16,008.00

B- 8760 Hr. 4000 KW
2000 Hr. 4700 KW
200 Hr. 5050 KW

PRICE: FOURTEEN THOUSAND, SIX HUNDRED TEN DOLLARS ---------$14,610.00

PER ENGINE GENERATOR SET

As mentioned in my letter, dated October 11, 1973, next to the last paragraph, the Lube Oil and Fuel Oil Filters are not manufactured to ASME Section III, Class 3. They are manufactured to ASME 8.

C- 8760 Hr. 4000 KW
2000 Hr. 4700 KW

PRICE: EIGHT THOUSAND, NINETY-FIVE DOLLARS ---------------($ 8,095.00)

Bechtel Corporation — October 15, 1973

Addendum 1, as requested by Bechtel Corporation;

Five (5) Relays per Diesel Generator Unit, including installation in the panel and testing of panel ----------------

PRICE, per Unit: TWENTY-THREE HUNDRED DOLLARS ------------ $2,300.00

×4

PRICE, per Four (4) Units: NINETY-TWO HUNDRED DOLLARS ----- $9,200.00

Addendum 2, please refer to my letter dated October 10, 1973, page 6, under PRICING, 2-A, 2-B. These prices were for a Woodward Synchronizing System, SM Synchronizer. Descriptive Bulletins will be attached to this letter.

Electric Products Company have given their price for a Synchronizing System. These items are listed and priced on BASLER Model 310 as follows:

A- One (1) Synchronizing System to synchronize and monitor four (4) engine units by Electric Products Company ----------------

FIVE THOUSAND DOLLARS ---------------------------------- $ 5,000.00

B- Four (4) Synchronizers, independent system, to monitor each individual engine --

ELEVEN THOUSAND, FIVE HUNDRED DOLLARS ------------------- $11,500.00

Cooper-Bessemer KSV-16 engines are presently installed in the following Atomic Plants as emergency diesel generators. Please refer to Section 4, page 1, of original proposal, dated August 31, 1973.

Five (5) KSV-16 Commonwealth Edison Company Zion Stations 1 & 2

Two (2) KSV-16 Nebraska Public Power

The generators were manufactured by Ideal Electric Company
Electric Products Company have given a listing of their Atomic Plant Installations.

Yours very truly,

Harris

H. A. Johnson
San Francisco Manager
HAJ:vj
w/attachment

bcc: R. L. Spetka
J. E. Taucher
H. A. LaBrun

cc: Mr. E. Hanna

Phone. (216) 481-1500
T. W. X. 810-421-8224

PORTEC inc.
Electric Products Division 1725 CLARKSTONE ROAD • CLEVELAND, OHIO 44112

October 19, 1973

REPLY TO: 124 PAUL DRIVE
SAN RAFAEL, CALIFORNIA 94903
Telephone 415/479-9510
T.W.X. 910-384-4259

Cooper Bessemer Company
Division of Cooper Industries Incorporated
111 Pine Street, Room 709
San Francisco, California

Attention: Mr. Harris Johnson

Subject: Pennsylvania Power & Light Company
Susquehanna Units No. 1 & 2
Emergency Generator Technical Data Supplement No. 3
Our Reference Negotiation 73-283

Gentlemen:

We are pleased to submit the following response to questions which you and Mr. Bhudeb Lodh, Bechtel Corporation, raised during our telephone conversations earlier this week, supplementing our previous submittals dated September 27 and October 2, 1973:

No. 1 - Reference specification 8856-M-30, data sheet 11-15, paragraph 3.4 (q). The air temperature into the generator should be changed from 130° F to 120° F, and the air temperature out of the generator should be changed from 160 to 150. This will correspond to paragraph 5.2.2 of the subject specifications defining design ambient temperature at the diesel generator room.

No. 2 - The computer runouts for the short circuit decrement curves are based on generator winding temperatures of 75° C as covered under item 8 of our October 2, 1973 supplement. If we assume minimum room ambient temperature, with the generator at room temperature, we would expect the sustained short circuit value to increase by approximately 8 to 10% over that shown on the computer runouts. If we assume maximum hot temperature conditions, where the room and generator are at their maximum operating temperatures, we would expect the sustained short circuit current to be approximately 88% of the values shown on those computer runouts.

No. 3 - The generator we have proposed will be suitable, without price addition, for operation at 4,000 KW continuously, 4700 KW for 2,000 hours, 5050 KW for 200 hours, and 5100 KW for 120 hours, without exceeding the temperature limitations stated in the NEMA Standards for Class F insulation systems, such as we would provide for these particular units.

No. 4 - There will be no increase in price for changing the current transformers from 800/5 ratio, to 1000/5 ratio for the increased generator output current when operating at 5100 KW for 120 hours.

No. 5 - The size of the static excitation system would be increased slightly to accommodate the excitation requirements for the 5100 KW operation, for 120 hours, and your factory has been notified of this nominal price adjustment.

No. 6 - Reference addendum No. 1, page 40, paragraph 7.14.1 (m). As explained to Mr. Lodh, during our meeting last week, it is not possible to provide a manual voltage control which will track the automatic control such that an abrupt change in excitation will not occur on transfer. The change in excitation will depend upon the manual voltage rheostat setting at the time the transfer takes place from automatic to manual operation. It is possible for example to set the manual voltage adjusting rheostat for rated voltage at rated design load, which will reduce any change in excitation to a minimum on transfer from automatic operation to manual with design load on the equipment. This we believe would be a satisfactory operation procedure.

There is another method that might be considered to protect the excitation system in the event of a potential transformer failure, and that would be to use redundant or dual voltage regulators as part of the static exciter voltage regulator system. This would require two sets of potential transformers, one for each voltage regulator, with an automatic transfer from one voltage regulator to the other taking place upon failure of the potential transformer, initiated either by the unbalance relay called for in the specifications, or using an overvoltage relay such as we have discussed with Mr. Lodh. The voltage adjusting rheostats on the two voltage regulators will be

connected together in tandem so that the standby voltage regulator will always be tracking the voltage regulator in operation, and in this manner prevent an abrupt change in excitation as stated in the specifications. This method was employed on the static excitation system for the Arkansas Power & Light nuclear power plant. The price addition for this feature is relatively nominal, and your factory has been given the proposal.

No. 7 - The relays itemized under paragraph 7.14.2.C-8, of addendum No. 1, are included in our proposal.

No. 8 - We have been asked to comment on the thermal limit of these generators when operating under short circuit conditions. Since the customer will be furnishing a neutral grounding device to limit line to ground fault conditions, we are only concerned with line to line and three phase symetrical short circuit possibilities. Assuming that all the heat in watt-seconds is absorbed in the generator windings, and assuming that there is no heat conduction in the core, or cooling of the windings, the negative sequence per unit current squared times time in seconds will have a limit of approximately 180 to 200. The corresponding value for the per unit positive current in a three phase symetrical short circuit condition could go as high as twice this value.

No. 9 - Our factory has given your people in Mount Vernon a price for a Basler model 210 synchronizer. Mr. Lodh has asked why we have not quoted the Basler model 170 instead, since that is a single phase sensing synchronizer, compared to the three phase sensing for model 210. We offered the model 210 because our control personnel have experienced problems in the past with systems employing only single phased sensing synchronizers, however we are not adamant about this recommendation should Bechtel prefer the single phase sensing model. The Basler model 170 is more expensive than the 210 since it is front panel mounted, whereas the model 210 is mounted within the cabinet and not accessible without opening the cabinet door.

The price addition for model 170 over model 210 previously quoted, [redacted] each. For either model we would include a vari-sync package, which provides a safety backup to pre vent synchronization in the event there is a circuit failure or malfunction within either of these two synchronizer models.

We trust that we have covered all of the remaining questions which you and Bechte Corporation have brought to our attention and we hope that the response is satisfactory. If you need any additional information or clarification please contact the writer.

Sincerely,

L. C. Madison
Regional Manager

LCM:dd

Bechtel Power Corporation
Engineers—Constructors

Fifty Beale Street
San Francisco, California
Mail Address: P O Box 3965, San Francisco, CA 94119

Cooper-Bessemer Company
111 Pine Street, Room 1011
San Francisco, California 94111

July 30, 1974

Attention: Mr. H.A. Johnson
San Francisco Manager

Subject: Susquehanna Steam Electric Station
Units 1 and 2
Job 8856
File M-30PO
Your SO-0188
P.O. 8856-M-30
Diesel Generator-Meeting Minutes

Gentlemen:

Attached for your information is a copy of the minutes of the meeting between Cooper-Bessemer and Bechtel held on July 9 and 10, 1974, to discuss engineering aspects of the M-30, "Diesel Generator" purchase order.

The action items are being pursued. If you require additional information or have comments on any of the indicated items, do not hesitate to contact us.

Very turly yours,

G. L. Parkinson
G. L. Parkinson
Project Engineer

CLP/JAW/sdl

RECEIVED
JUL 31 1974
Cooper-Bessemer Co.
San Francisco Office

MEETING MINUTES

8856-M-30 DIESEL GENERATORS

COOPER BESSEMER/BECHTEL

DATES OF MEETING: July 9 & 10, 1974

PLACE OF MEETING: Mt. Vernon, Ohio

LIST OF SUBJECTS DISCUSSED:

The purpose of this meeting was to resolve outstanding engineering problems in the mechanical, electrical/control and layout areas.

ATTENDEES:

Bechtel	Cooper Bessemer	Portec (Electric Products
E. Hohn	K. Beightol	R. Evans*
J. Palmer	H. Johnson	L. Madison*
C. Piette	T. Kearns	
J. Weyandt	J. Rentz	
	J. Taucher	

CB En-Tronics
J. Lahr*
H. Lenz*

*Part Time

Documents Exchanged:

From	To	Identification No.	Title
Bechtel	Cooper Bessemer	8856-M-260, rev 0	"Equipment Location Units 1&: Diesel Generator Building Pla of Elevs. 667'-0" & 660'-0""
Bechtel	Cooper Bessemer	8856-M-261, rev 0	"Equipment Location Units 1&: Diesel Generator Building Pla of Els. 723' & 711'-6", Sects A-A & B-B"
Cooper Bessemer	Bechtel	KSV-47-14 (dated 7-2-74)	"Cooling Water Schematic"
Cooper Bessemer	Bechtel	KSV-47-15 (dated 7-2-74)	"Jacket Water Schematic"
Cooper Bessemer	Bechtel	KSV-51-12 (dated 7-2-74)	"Fuel Oil Piping Schematic"
Cooper Bessemer	Bechtel	Diagram SK-5 (dated 3-11-74)	"Starting Air System"

15. G11-6.11 Panel heater will be manually controlled in accordance with spec. M30 sections 7.14.3a(2) and b(7).

16. G11-7.4 C-B proposes to use a metal stamped impression in an adjacent bulkhead for tubing terminal identification.

17. G11-7.6 C-B advises tubing runs to panels will have bottom entry.

18. G11-8.1.8 C-B uses permanent wiring (instead of plug-in bus) for 120V a-c power supply to instruments.

19. G11-8.1.9 C-B uses PVC raceway for internal panel wiring.

20. G11-8.4.1 C-B will use CR 2940 control switches on their panels.

21. G11-8.4.5 Panel sizes used by C-B are based on a vertical arrangement of indicating lights above control switch stations.

22. G11-9.1 C-B proposes to use engraved lamicoid or photo process aluminum nameplates.

23. G11-fig 43 C-B questioned the need for states NT sliding link terminal blocks in certain locations. Bechtel explained that they were to be used where external wiring is connected to panels to facilitate testing. C-B indicated that it would be impractical to use them for panel-to-panel wiring of the fuel control system. C-B drawings will be reviewed by Bechtel to determine the extent of required usage.

24. G11-8.1.1 C-B suggested teflon insulated wire or THW wire as a substitute for the panel wire specified.

<u>Comments - General</u>

1. The output rating of each diesel generator (see spec. sections 5.2) will be 4700 KW at 0.8 power factor. This is a continuous rating for up to 2000 hours at which time the unit is to be inspected and such maintenance work as may be required, is to be done. The 2000 hours is <u>not</u> a routine maintenance interval because it is entirely possible that no maintenance work may be required until more than one of these intervals has elapsed.

Q

MATERIAL REQUISITION

MATERIAL: DIESEL GENERATORS

SHOP INSPECTION ☒ IS ☐ IS NOT REQU

COST CODE: 4.095

JOB SITE DELIVERY DATE: Jo[illegible]
Units 1, 2

M/R STATUS

REV NO	DATE	REVISIONS	ORIGINATOR (NAME)	APPROVALS
A	12/4/[illegible]	Issued for Client Approval	E. Hanna	[illegible]
0	6/[illegible]/73	Issued for Bids	E. Hanna	[illegible]
1	8/24/73	Revised to add Addendums 1 and 2	E. Hanna	[illegible]
2	5/28/74	Issued For Purchase	C Petto	[illegible]

Job
Requisition No
8856-M-30
Rev

5.1.2 The diesel generators will be located on the plant site and will be independent of offsite power sources.

5.2 <u>Service Conditions</u>

5.2.1 The standby diesel generators shall be designed for the following conditions:

Number of units required: 4

Starting time to rated speed and voltage:	10 seconds max.
Rating per unit: Continuous	4000 KW
Overload (2000 hours)	= 4700 KW
Overload (200 hours)	= 4700 KW
Overload (30 mins.)	= 4700 KW
Overload (10 seconds)	= 4700 KW
Power factor, lagging	0.8
Frequency	60.0 Hertz
Voltage	4160 V
Phase	3 phase, wye
Neutral connection	External
Overload capacity (2 hours in any 24 hours)	4700 KW
Service	Standby Emergency
Speed	600 rpm
Largest motor HP to be started	2000 HP
Elevation at site	676 ft

Revision 5

Q

DESIGN SPECIFICATION

FOR

DIESEL GENERATORS

FOR THE

SUSQUEHANNA STEAM ELECTRIC STATION, UNITS 1 AND 2

PENNSYLVANIA POWER AND LIGHT COMPANY

ALLENTOWN, PENNSYLVANIA

SPEC. FILE NO. 17

ADDENDUM No. ______
F.C.R. No. ______
F.C.N. No. ______

Bechtel

San Francisco

REFERENCE COPY

Sheet 1 of 66

No.	DATE	REVISIONS	BY	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
5	4/21/81	Revised to incorporate 300 start tests & add. 1, 2, & 3 to Rev. 4 (DCA 5)	Ama	[illegible]	[illegible]	[illegible]	[illegible]	
4	8/11/78	REVISED TO INCORPORATE Rev. 3 ADDENDUMS 1, 2 & 3 AND REV'D 8856-Q41 Sh2 Rev 3	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
3	12/2/76	Revised per vendor supplied package	J Klee	[illegible]	[illegible]	[illegible]	[illegible]	
2	11/8/74	Revised To REFLECT VENDOR PACKAGE PURCH'D	Piette	[illegible]	[illegible]	[illegible]	[illegible]	
1	5/28/74	Issued for Purchase	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
0	6/13/73	Issued for Bids	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	
A	2-6-72	Issued for Approval	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	

SCALE | DESIGNED | DRAWN

BECHTEL

SUSQUEHANNA STEAM ELECTRIC STATION
UNITS 1 & 2
PENNSYLVANIA POWER & LIGHT COMPANY

JOB NO. 8856

SPEC/DES GUIDE NO. 8856-M-30 | 5

SHEET 1 OF 66

Supplies For Nuclear Power Generating Stations". IEEE - Std. 387-1972.

4.3 The selection of rating and performance characteristics is in accordance with AEC Safety Guide No. 9, March 10, 1971 "Selection of Diesel Generator Set Capacity for Standby Power Supplies".

4.4 In addition, the air receivers, heat exchangers, other pressure vessels and the diesel fuel oil system shall comply in all respects with the latest revision including addenda of Section III Class 3, of the ASME Boiler and Pressure Vessel Code with applicable code stamp and the Commonwealth of Pennsylvania Requirements for Boilers and Unfired Pressure Vessels. The Lube oil filter vessel shall be designed and fabricated in accordance with Section VIII of the ASME Code.

4.5 The Seller shall state which sections of the codes and standards listed herein apply.

4.6 In the event of any conflict in the above, the following priority sequence shall be followed, remembering that the more restrictive code or standard shall apply unless directed otherwise by the Buyer.

4.6.1 U.S. Department of Labor-Occupational Safety and Health Standards

4.6.2 Commonwealth of Pennsylvania Regulations

4.6.3 Local Agencies

4.6.4 Bechtel Specifications

5.0 SERVICE CONDITIONS AND PERFORMANCE

5.1 General

5.1.1 The equipment specified herein will serve a nuclear generating station consisting of two nominal 1100 MWe turbine-generator units. The diesel generators shall be designed to provide a reliable source of electrical power during an incident and for safe shutdown of the reactors.

5.1.2 The diesel generators will be located on the plant site and will be independent of offsite power sources.

5.2 Service Conditions

5.2.1 The standby diesel generators shall be designed for the following conditions:

Number of units required: 4

Starting time to rated speed and voltage:	10 seconds max.
Rating per unit: Continuous	4000 KW
Overload (2000 hours)	= 4700 KW
Overload (200 hours)	= 4700 KW
Overload (30 mins.)	= 4700 KW
Overload (10 seconds)	= 4700 KW
Power factor, lagging	0.8
Frequency	60.0 Hertz
Voltage	4160 V
Phase	3 phase, wye
Neutral connection	External
Service	Standby Emergency
Speed	600 rpm
Largest motor HP to be started	2000 HP
Elevation at site	676 ft

5.2.2 Design ambient Temperature at Diesel Generator Room:

a) Engine off (Min/Max)	72°/104°F
b) Engine Running: Max. At Generator	120°F
At Engine	160°F

cc: E. W. Figard
S. B. Kuhn
T. C. Dalpiaz
D. F. Roth
F. J. Czysz

October 13, 1989

MEMORANDUM

TO: D. J. Gandenberger

SUBJ: Diesel Piston Ring Break-In

My original assessment for the time required to seat newly installed piston rings was based on my previous experience with Fairbanks-Morse diesel engines; however, after lengthy discussions with Cooper-Bessemer senior management, I feel that our approach to seating piston rings must be changed.

Cooper-Bessemer states that the time to seat piston rings may take upwards of 100 hours of engine operation. Cooper strongly recommends that the minimum acceptable break-in for piston rings should consist of the following:

4 Hours	-	600 rpm unloaded
2 Hours	-	loaded to 25%
2 Hours	-	loaded to 50%
2 Hours	-	loaded to 75%
2 Hours	-	loaded to 100%

Following the 12 hour break-in the diesel engine should be inspected to determine cylinder liner and piston condition.

My personal position is that we at least follow the minimum requirements set by Cooper-Bessemer since it is logical and because they should be considered as the experts on this engine design.

J R Adams
J. R. Adams

JRA/pag

PP&L

cc: F. J. Czysz
T. C. Dalpiaz
D. J. Gandenberger
J. J. Graham
S. B. Kuhn
K. R. Mantz
T. J. Nork
R. L. Tutorow

October 15, 1989

MEMORANDUM

TO: E. W. Figard/G. J. Kuczynski

SUBJ: Acceptance Criteria for Diesel Visual Inspections

Acceptance criteria has been non-existent for visual inspections on the diesel generator. Acceptance criteria is necessary to establish a common dialogue and a standard set of ground rules. This set of acceptance criteria was developed in conjunction with Allen Lambert, Cooper-Bessemer Quality Assurance Manager, and will be invoked during and following the scheduled post maintanance and 24 hour surveillance test on the "C" diesel generator.

The acceptance criteria within this memorandum will be included in a future maintenance procedure so all future diesel inspections can benefit from lessons learned in the "C" diesel generator.

The following definitions are included to establish a common dialogue among members of the inspection team.

- Burnishing (*) - Smoothing surfaces through frictional contact between two parts. This contact produces a polished effect.
- Scuffing (*) - A form of adhesive wear that produces superficial scratches or a high polish on the rubbing surfaces. It is observed most often on inadequately lubricated parts.
- Scoring (*) - Marring or scratching of a smooth surface; most often caused by sliding contact with a mating member having a hard projection or embedded particle on its surface.

- o Tin Smearing - Tin that is wiped from one location and redeposited onto another location.

- o Inspection Team - Consists of the following members:
  - Maintenance Assistant Foreman
  - Maintenance Mechanic trained in diesel engine inspection.
  - Maintenance engineer responsible for diesel engine.
  - Quality control personnel as required.

Diesel engine internal acceptance criteria is delineated below and based upon the definitions stated above.

- o Cylinder Liner
  - Scoring is not permissible.
  - Scuffing is permissible, provided there is no raised metal between cylinder wall and scuff.
  - Burnishing is permissible, provided there is no discoloration evident. Discolored surface requires a boroscope inspection of the upper cylinder liner, through the fuel injection nozzle port in the cylinder head, and an engineering evaluation.

- o Piston
  - Outside diameter
    - o Casting defects are not permissible.
    - o Evidence of tin smearing requires an engineering evaluation.
    - o Scoring is not permissible.
  - Inside diameter
    - o Non-uniform carbon build-up is not permissible.
    - o Evidence of cracking is not permissible.
    - o Evidence of impact damage is not permissible.

- o Piston pin/articulated rod pin
  - Discoloration of pin is not permissible.
  - Scoring of pin is not permissible.
  - Scuffing of pin is not permissible.
  - Connecting rod
    - o Nicks or other forms of surface damage are not permissible.

The following items listed below are required to determine if the diesel engine is operating within existing guidelines.

- o Oil Samples
  Samples must be taken from the following points and submitted to the Hazleton lab for analysis:

  - Obtain an unfiltered sample from the engine sump prior to engine operation.
  - Obtain a sample of oil upstream of oil filter prior to shutting down diesel following post maintenance testing.

- o Diesel generator logs shall be taken in accordance with OI-024-004 and evaluated by maintenance at the conclusion of post maintenance testing.

- o During diesel operation a through walkdown of the diesel, its auxiliary skid, and observation of crankcase ventilation exhaust vent in the CST burm area is required as a minimum whenever logs are taken. Special emphasis shall be placed on engine noise, leaks or any smoke emanating from engine. Additionally any vapor issuing from the crankcase ventilation exhaust piping that is grey to black in color should result in operations stopping the engine.

Any item found to be not permissible during the internal inspection excluding the connecting rod will require the piston to be removed from its cylinder liner to determine cause.

NOTE:
(*) in definitions indicates that the ASM Metals Reference Book, second edition, was used to determine the correct definition.

J R Adams
J. R. Adams

Math M. Heidorn
M. M. Heidorn

JRA/pag

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REVIEW OF RING SET SD-94-KSV-9

COOPER KAYDON

TOP COMPRESSION RING
2ND COMPRESSION RING

TYPE: TAPER FACE/TWISTED

	Cooper spec.	Handbook data.
PART NO:	2-20R-005-001	A5701
WIDTH:	.2465/.2480	.2475/.2490
RADIAL THICKNESS:	0.430/0.450	0.455/0.470
END GAP:	0.080/0.105	0.054 MIN.
DIA. TENSION:	50-60 LBS (To close gap)	50-60 LBS
MATERIAL:	K-28, PARCO	K-28, PARCO
BACK CL. IN GROOVE:	0.020/0.050	0.054 MIN.
MIN. GROOVE DEPTH:	0.470	0.524
MIN. SIDE CLEARANCE:	0.007	0.008

3RD COMPRESSION RING
4TH COMPRESSION RING

TYPE: TAPER FACE/TWISTED

PART NO:	2-20R-354-001	80731
WIDTH:	.2465/.2480	.2475/.2490
RADIAL THICKNESS:	0.445/0.465	0.455/0.470
END GAP:	0.080/0.105	0.054 MIN.
DIA. TENSION:	34-44 LBS.	34-44 LBS.
MATERIAL:	K-IRON, PARCO	K-IRON, PARCO
BACK CL. IN GROOVE:	0.005/0.035	0.054 MIN.
MIN. GROOVE DEPTH:	0.470	0.524
MIN. SIDE CLEARANCE:	0.004	0.008

5TH OIL RING
6TH OIL RING
7TH OIL RING

CONFORMABLE DOUBLE HOOK SCRAPER, VENTILATED

PART NO:	2-20R-652-001	86188 RING 86139 SPRING
WIDTH:	0.372/0.3735	0.3715/0.373
RADIAL THICKNESS:	0.290/0.310	NO INFO
END GAP:	0.040/0.065	0.03375 MIN.
UNIT PRESSURE:	188 PSI	188 PSI
MATERIAL:	K6E PARCO	K6E PARCO
BACK CL. IN GROOVE:	NA	NA
MIN. GROOVE DEPTH:	0.544	NA
MIN. SIDE CLEARANCE	0.0035/0.0065	0.0025 MIN.

PISTON RING DIMENSIONAL DATA

D/G	PISTON	RING #	RADIAL WIDTHS (INCHES)	COMMENT
B	7L	1	-	Missing
		2	.450	Wrong Ring
		3	.454-.460	OK
		4	-	Missing
C	5R	1	-	Missing
		2	.437	OK
		3	.448	OK
		4	.449-.450	OK
D	8R	1	.429	OK
		2	.429	OK
		3	.445	OK
		4	.454	OK
?	A Exact Piston Location Unknown	1	.455-.460	Wrong Ring
		2	.453-.455	Wrong Ring
		3	.450-.456	OK
		4	.458-.463	OK
?	B Exact Piston Location Unknown	1	.428-.432	OK
		2	.455-.460	Wrong Ring
		3	(.435?)-.458	OK
		4	.451-.459	OK
A	1R	1	.425-.428	OK
		2	.428-.432	OK
		3	.448-.454	OK
		4	.450-.450	OK
A	2R	1	.424-.430	OK
		2	.422-.430	OK
		3	.442-.450	OK
		4	.450-.455	OK
A	7R	1	.440-.446	May Be Wrong Ring
		2	.443-.452	Wrong Ring
		3	.458-.460	OK
		4	.458-.461	OK
B	1R	1	.449-.456	Wrong Ring
		2	.462-.464	Wrong Ring
		3	.451-.456	OK
		4	.455-.458	OK

ATTACHMENT 6

PISTON RING DIMENSIONAL DATA

D/G	PISTON	RING #	RADIAL WIDTHS (INCHES)	COMMENT
B	5L	1	.436-.437	OK
		2	.437-.438	OK
		3	.456-.459	OK
		4	.459-.460	OK
A	3R	1	.429-.434	OK
		2	.432-.433	OK
		3	.464-.466	OK
		4	.457-.460	OK
New (from stock)		1	.436-.437	OK
		2	.437-.438	OK
		3	.457-.462	OK
		4	.459-.460	OK

fjc/mski14i(32)

File: DIESELS

TABLE SHOWING THE VERIFICATION OF THE MATERIAL PROPERTIES OF THE VARIOUS RINGS EXAMINED FROM SSES PISTON SETS.

HARDNESS READINGS

RING NUM.	'B' 5L	'B' 7L	'C' 5R	'D' 8R	NEW SET
1	40,40,42 Rc				
2	41,40.5,42,41			41,41,40 Rc	
3	77,76,79,79 Rg				
4	85,78,81,80			78,79 Rg	
5					
6					
7					

MICROSTRUCTURE

RING NUM.	'B' 5L	'B' 7L	'C' 5R	'D' 8R	NEW SET
1	NODULAR FE				NODULAR FE
2	NODULAR FE				NODULAR FE
3	FLAKE GRAPHI				FLAKE GRAPHI
4	FLAKE GRAPHI	FLAKE GRAPHI			FLAKE GRAPHI
5		FLAKE GRAPHI			
6					
7					

DIMENSIONS

RING NUM.	'B' 5L	'B' 7L	'C' 5R	'D' 8R	NEW SET
1				0.429"	0.437-0.436"
2		0.450"	0.437"	0.429"	0.438-0.437"
3		0.454-0.460"	0.448"	0.445"	0.457-0.462"
4			0.450"	0.454"	0.459-0.460"
5					
6					
7					

NOTE I. RINGS 1,2,3,AND 4 ARE THE COMPRESSION RINGS
NOTE II. RINGS 4,5,AND 6 ARE THE OIL RINGS
NOTE III. "GRAPHI" IS GRAPHITE
NOTE IV. RANGE OF DIMENSION FOR RINGS 1 AND 2 IS 0.430 TO 0.450"
NOTE V. RANGE OF DIMENSION FOR RINGS 3 AND 4 IS 0.445 TO 0.465"
NOTE VI. RANGE OF HARDNESS FOR RINGS 1 AND 2 IS 40-46 Rc
NOTE VII. RANGE OF HARDNESS FOR RINGS 3 AND 4 IS 72-88 Rg

11/14/89 14:37 C.B. RECIP.

TELECOPY
COOPER-BESSEMER RECIPROCATING (412) 458-8000 EXT. 4184
GROVE CITY, PENNSYLVANIA 16127

DATE: November 14, 1989	SENT BY: Billie Jean EXT: 3395
TO: L. Willertz	FROM: W. H. A. Lambert
COMPANY: P.P. & L.	NO. OF PAGES: (5)
FAX NO. 215-770-7830	OUR REF. NO. Our Ref: QCG-6516

SUBJECT: Piston/Liner Evaluation

MESSAGE

We are sending you herewith copies of various pages from the textbook "Engineers Handbook of Piston Rings...." by Koppers. A second set is in the mail in case the FAX reproduction of the photomicrographs is not adequate. This information covers "K" iron used in the third and fourth compression rings and "K28" used in the first and second. We do not have any published data on actual chemistry used.

W. H. A. Lambert /BJB
W. H. A. Lambert

WHAL/bjb

cc: File: 5-1
SO-0168/P2
K-5af17

CHAPTER VII

MATERIALS

This chapter provides detailed information for use in selecting a material for a specific piston ring application. For the most part, the materials have been grouped into families where their characteristics are similar. First, each material, or family of materials, is defined as to the method of manufacture - sand cast, centrifugally cast, pot cast or sintered. Next, further insight into the material is gained under the heading of "Characteristics." Here, in descriptive language the characteristics of the material from a metallurgical point of view are explained as they affect wear and mechanical properties. This information is augmented by a set of photomicrographs of the material at 100X and 500X and a set of mechanical property data including strength, elastic modulus, hardness, maximum allowable stress and impact strength. General fields of suitable usage for each material are given under the heading "Types of Service."

Many materials, including some highly specialized types, are required to cover the entire breadth of piston ring applications. The more popular types are described here.

K-Irons®

K-Irons are a family of statically cast grey irons which are used more widely than any other material for piston rings and seal rings. Cross-sections vary greatly with ring diameters ranging up to a maximum of ten feet. Because the mechanical properties of grey iron generally decrease with increasing cross-section, the chemistry of the iron is modified as cross-sectional area increases, to help to compensate for this change.

Characteristics

K-Irons are closely controlled, unalloyed, grey cast irons. These materials possess flake-type graphite evenly distributed throughout a matrix of pearlite, a eutectic of iron and iron-carbide. This combination of pearlite and flake-type graphite provides an excellent wearing material. The graphite acts as an internal lubricant for the ring as it wipes against a mating surface, keeping scuffing and wear to a minimum.

K-Iron rings possess properties tending to promote conformability, permitting the rings to follow slight out-of-roundness or taper in a cylinder, thus promoting a good seal.

For chromium plated oil control rings, K-Iron is modified slightly to produce a tighter type structure. This modification is an aid in producing smooth chromium plated rails associated with oil rings.

Types of Service

K-Irons are used for compression rings and oil rings in many types of internal combustion engines, steam engines, pumps, compressors and miscellaneous applications. The self-lubricating properties of K-Irons permit rings of these materials to be used unplated, even when lubrication is borderline. However, these materials also form good base metals for plated rings.

Diameter: Up to 120" (*3000 mm*).

K-IRONS
MECHANICAL AND PHYSICAL PROPERTIES
FOR COMPRESSION, CHROME COMPRESSION AND UNPLATED OIL CONTROL RINGS

PROPERTY	UNIT		MINIMUM DIMENSION OF FINISHED RING WIDTH OR WALL									
			.0625-.1250 (1.587-3.175)		OVER .1250 TO .3437 INCL. (3.175-8.730)		OVER .3437 TO .5625 INCL. (8.730-14.287)		OVER .5625 TO .750 INCL. (14.287-19.050)		OVER .750 (19.050)	
	IN.	METRIC	IN.	METRIC	IN.	METRIC	IN.	METRIC	IN.	METRIC	IN.	METRIC
Tensile Strength, min.	psi	kg/mm²	35,000	24.61	32,000	22.50	28,000	19.69	25,000	17.58	25,000	17.58
Transverse Rupture Strength, min.	psi	kg/mm²	70,000	49.22	64,000	45.00	56,000	39.37	50,000	35.15	50,000	35.15
Maximum Allowable Transverse Stress	psi	kg/mm²	55,000	38.67	50,000	35.15	44,000	30.94	40,000	28.12	40,000	28.12
Transverse Modulus of Elasticity at Stress Shown	psi x 10⁶ / psi	kg/mm² / kg/mm²	11-15 / 40,000	7734-10546 / 28.12	11-15 / 35,000	7734-10546 / 24.61	11-15 / 33,000	7734-10546 / 23.20	11-15 / 30,000	7734-10546 / 21.09	11-15 / 30,000	7734-10546 / 21.09
Izod Impact Strength (min.) .160 x .260 Unnotched Bar	in-lbs	mm-kg	1.5	17.3	1.5	17.3	1.5	17.3	1.5	17.3	1.5	17.3
Hardness Rockwell "G" Scale Brinell, 3000 kg Load 10 mm Ball	G BHN		77.5-91 211-275		72-88 194-260		68-84.5 182-241		64-81 172-228		60-78 162-210	
Coefficient of Expansion	in/in/°F (mm/mm/°C)		6.2 x 10⁻⁶ (11.16 x 10⁻⁶)		6.2 x 10⁻⁶ (11.16 x 10⁻⁶)		6.2 x 10⁻⁶ (11.16 x 10⁻⁶)		6.2 x 10⁻⁶ (11.16 x 10⁻⁶)		6.2 x 10⁻⁶ (11.16x10⁻⁶)	
Density	lb/in³	gr/cm³	.256	7.09	.256	7.09	.256	7.09	.256	7.09	.256	

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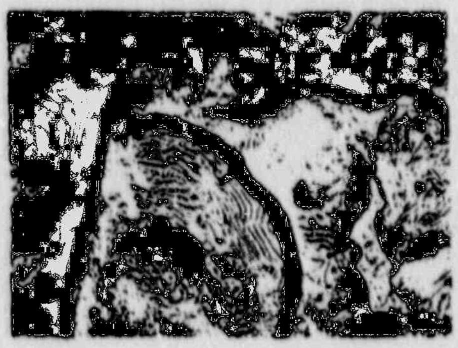
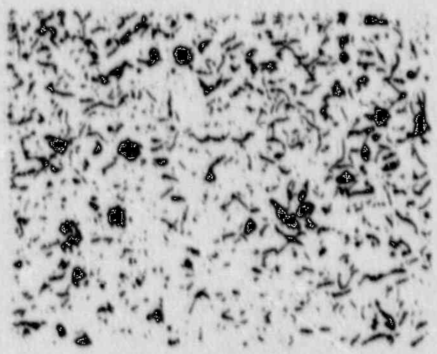
PHOTOMICROGRAPHS OF K-2 IRON

Typical K-2 Section—100X—Unetched
Graphite structure.

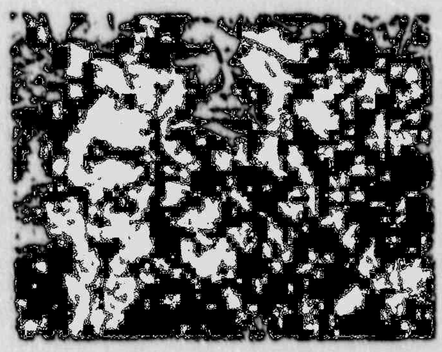


Typical K-2 Section—525X—Picral Etch
Fine pearlite matrix.

PHOTOMICROGRAPHS OF K-1 IRON



Typical K-1 Section—100X—Unetched
Graphite structure.



Typical K-1 Section—510X—Picral Etch
Fine pearlite matrix.

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K-16, K-27 and K-28 Ductile Irons

K-16, K-27 and K-28 are centrifugally cast ductile irons in which free graphite occurs in the form of tiny spheroids. The shape and distribution of the graphite occurring in this form so markedly enhance the mechanical properties that these materials are comparable to steel in strength while still retaining, to a great extent, the good wearing properties of cast iron. K-16 has a pearlitic matrix while K-27 and K-28 have martensitic matrices.

Characteristics

K-16, K-27 and K-28 are all very high strength ductile cast irons having minimum tensile strengths of 90,000 psi [*63.27 kg/mm²*] for K-16, 110,000 psi [*77.33 kg/mm*] for K-27 and 120,000 psi [*84.36 kg/mm²*] for K-28. They possess elastic moduli greater than the grey cast irons yet not as high as steel. While the self-lubricating properties of these materials are considerably better than those of steel they are not as good as grey cast iron and for that reason must be used with adequate lubrication. In many applications, the major wearing surface is chromium plated or plasma coated to obtain optimum wear characteristics.

K-16 is similar to K-Spun and may be used in lieu of that material. For piston rings having a wall dimension greater than 0.350 inches [*9mm*] K-16 should be specified in place of K-Spun. K-16 corresponds to the ASTM Type 80-60-03 ductile iron.

K-27 and K-28 are somewhat similar to Koppers F-88 and are recommended in lieu of F-88 when the ring wall dimension exceeds 0.350 inches [*9mm*]. K-27 is equivalent to ASTM Type 120-90-02 ductile iron. K-28 is a hardened version of K-27 having hardness of Rockwell "C" scale 40-46 as against Rockwell "C" scale 24-34 for K-27. K-28, being much harder than K-27, has found its best application where it bears against another hard surface such as chromium plate.

Types of Service

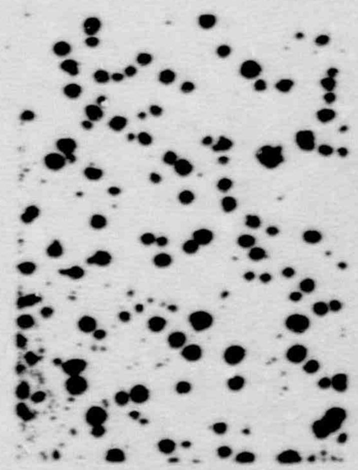
These materials are used primarily for piston rings in high output internal combustion engines where high strength and toughness are necessary attributes along with wear resistance, where grey irons are too weak, and steels present undesirable wearing characteristics. K-28 has found specialized service where one of the wear surfaces is hard, such as a chromium plated, nitrided or carburized cylinder liner.

Diameter: .750" to 24" [*20 to 610mm*]

K-16, K-27 and K-28
MECHANICAL AND PHYSICAL PROPERTIES

Property	Unit In.	Unit Metric	K-16 In.	K-16 Metric	K-27 In.	K-27 Metric	K-28 In.	K-28 Metric
Tensile Strength (min.)	psi	kg/mm²	90,000	63.27	110,000	77.33	120,000	84.36
Yield Strength (min.)	psi	kg/mm²	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Elongation in 2 inches	%		2		1		1	
Transverse Rupture Stress (min.)	psi	kg/mm²	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Maximum Allowable Transverse Stress	psi	kg/mm²	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Transverse Modulus of Elasticity at 70,000 psi (49.21 kg/mm²) Stress	psi x 10^6	kg/mm²	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Izod Impact Strength (min.)								
[illegible] unnotched bar	in-lbs	mm/kg	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible] unnotched bar	in-lbs	mm/kg			[illegible]	[illegible]	[illegible]	[illegible]
Hardness								
Rockwell "C" Scale	"C"			[illegible]		[illegible]		[illegible]
Brinell, 3000 kg, 10 mm ball	BHN			[illegible]		[illegible]		[illegible]
Coefficient of Expansion	in/in/°F	mm/mm/°C	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
Density	lbs/in³	gm/cm³	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]	[illegible]

PHOTOMICROGRAPHS OF K-16



100X Unetched
Spheroidal type of graphite

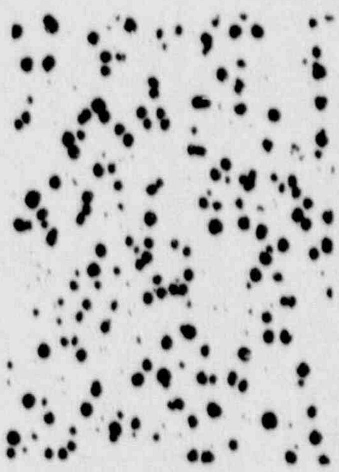
500X Nital Etch
Fine to medium pearlite

PHOTOMICROGRAPHS OF K-27

Centrifugally Cast - 100X - Unetched
Spheroidal type of graphite

Centrifugally Cast - 500X - Picral Etch
Tempered martensite matrix

PHOTOMICROGRAPHS OF K-28



Centrifugally Cast - 100X - Unetched
Spheroidal type of graphite

Centrifugally Cast - 500X - Picral Etch
Tempered martensite matrix

SGS Control Services Inc.

20 Lafayette Street
Carteret, New Jersey 07008
Tel. (201) 541-7200
Telex: 4754303 (SCRTUI)

December 26, 1989

A.W. Snyder
Penn. Power & Light
Chemical Laboratory
Cedar & Buttonwood Streets
Hazleton, Pa. 18301-0338

"FOAMING CHARACTERISTICS"
ASTM D-892
SAMPLES RECEIVED DECEMBER 23, 1989
SERVICE ORDER S-09357-5

SAMPLE NO.	MARK	SEQUENCE	INITIAL	10 MINUTE
24481	"A" EDG.L.O./8927524	I	Nil	Nil
		II	4	Nil
		III	Nil	Nil
24482	"B" EDG.L.O./8927525	I	8	Nil
		II	Nil	Nil
		III	Nil	Nil
24483	Storm Drum/8927547 NEW Superduty 40	I	Nil	Nil
		II	Nil	Nil
		III	Nil	Nil
24484	"D" EDG.L.O./8927527	I	Nil	Nil
		II	Nil	Nil
		III	Nil	Nil
24485	"E" EDG.L.O./8927528	I	Nil	Nil
		II	Nil	Nil
		III	Nil	Nil
24486	Chevron Delo 400 Sea 40 NEW	I	Nil	Nil
		II	Nil	Nil
		III	Nil	Nil

1) Samples from Susq. EDG's
2) No Sample Available from "C" Unit as it was out of Service for Repairs.
3) Storage Drum 8927547 is New Gulf Super Duty #40 Lubricant which will be used to fill "C" unit.
4) Contacted Jim Adams - Unit "C" has been filled with new Superduty 40 as of 12/27/89

David E. Lunceford
Laboratory Manager

6. Materials

6.1 *Petroleum Distillate*, conforming to the specifications given in Appendix X2. (**Caution:** Combustible. Vapor harmful. See Annex A1.1.)

6.2 *Toluene*, nitration, or equivalent. (**Warning:** Flammable. Vapor harmful. See Annex A1.2.)

6.3 *Acetone*, ACS reagent grade, or equivalent. (**Danger:** Extremely flammable. Vapors may cause flash fire. See Annex A1.3.)

6.4 *Petroleum Spirit, 60/80*, conforming to the IP specification or the Appendix X3. (**Caution:** Combustible. Vapor harmful. See Annex A1.4.)

7. Preparation of Apparatus

7.1 Thorough cleansing of the test cylinder and air-inlet tube is essential after each use to remove any additive remaining from previous tests which would seriously interfere with results of subsequent tests.

7.1.1 *Cylinder*—Rinse the cylinder in turn with petroleum distillate (**Caution:** Combustible. Vapor harmful. See Annex A1.1), then petroleum spirit (**Caution:** Combustible. Vapor harmful. See Annex A1.4), followed by directing a current of clean, dry air into the cylinder. Rinse the cylinder in turn with distilled water, then acetone (**Danger:** Extremely flammable. Vapors may cause flash fire. See Annex A1.3), and dry with a current of clean, dry air.

7.1.2 *Diffuser Stone*—Clean the diffuser stone by washing in turn with petroleum distillate, toluene (**Warning:** Flammable. Vapor harmful. See Annex A1.2), and petroleum spirit. Immerse the stone in about 300 mL of petroleum distillate. Flush a portion of it back and forth through the stone at least five times with vacuum and air pressure. After completing the final washing in the petroleum spirit, dry both tube and stone thoroughly by forcing clean air through them. Wipe the outside of the air-inlet tube, first with a clean cloth moistened with petroleum distillate, then with a clean dry cloth.

8. Procedure

8.1 *Sequence I—Without* mechanical shaking or stirring, decant approximately 200 mL of sample into a beaker. Heat to 49 ± 3°C (120 ± 5°F) and allow to cool to 24 ± 3°C (75 ± 5°F). See Option A for stored sample.

NOTE 6—Each step of the procedure described in 8.2 and 8.4, respectively, shall be carried out within 3 h after completion of the previous step. In 8.3, the test should be carried out as soon as compatible with the temperature specification and not more than 3 h after immersion of the cylinder in the 93.5°C (200°F) bath.

8.2 Pour the sample into the 1000-mL cylinder until the liquid level is at the 190-mL mark. Immerse the cylinder at least to the 900-mL mark in the bath maintained at 24 ± 0.5°C (75 ± 1°F). When the oil has reached the bath temperature, insert the diffuser stone and the air-inlet tube with the air source disconnected, and permit the stone to soak for about 5 min. Connect the air-outlet tube to the air volume measuring device. At the end of 5 min, connect to the air source, adjust the air flow rate to 94 ± 5 mL/min, and force clean dry air through the stone for 5 min ± 3 s, timed from the first appearance of air bubbles rising from the stone. At the end of this period, shut off the air flow by disconnecting the hose from the flow meter and immediately record the volume of foam; that is, the volume between the oil level and the top of the foam. The total air volume which has passed through the system shall be 470 ± 25 mL. Allow the cylinder to stand for 10 min ± 10 s and again record the volume of foam.

8.3 *Sequence II*—Pour a second portion of sample into a cleaned 1000-mL cylinder until the liquid level is at the 180-mL mark. Immerse the cylinder at least to the 900-mL mark in the bath maintained at 93.5 ± 0.5°C (200 ± 1°F). When the oil has reached a temperature of 93 ± 1°C (199 ± 2°F), insert a clean diffuser stone and air-inlet tube and proceed as described in 8.2, recording the volume of foam at the end of the blowing and setting periods.

8.4 *Sequence III*—Collapse any foam remaining after the test at 93.5°C (200°F) (8.3), by stirring. Cool the sample to a temperature below 43.5°C (110°F) by allowing the test cylinder to stand in air at room temperature, then place the cylinder in the bath maintained at 24 ± 0.5°C (75 ± 1°F). After the oil has reached bath temperature, insert a cleaned air-inlet tube and diffuser stone and proceed as described in 8.2, recording the foam value at the end of the blowing and settling periods.

8.5 Some lubricants with modern additives can pass their foam requirements when blended (with the antifoam properly dispersed in small particle sizes) but fail to meet the same require-

ments after two or more weeks' storage. (It appears that the polar dispersant additives have the potency to attract and hold antifoam particles, such that the apparent increased antifoam size results in decreased effectiveness to control foam in D 892.) However, if the same stored oil is merely decanted and poured into engines, transmissions, or gear boxes and those units operated for a few minutes, the oil again meets its foam targets. Similarly, *decanting* the stored oil into a blender, followed by agitation as described for Option A below, redisperses the antifoam held in suspension and the oil again will give good foam control in D 892. For such oils, Option A may be used. On the other hand, if the antifoam is not dispersed into sufficiently small particles when the oil is blended, the oil may not meet its foam requirements. If this freshly blended oil were vigorously stirred according to Option A, it is very possible that the oil would then meet its foam targets whereas the plant blend would never do so. Therefore, it is inappropriate and misleading to apply Option A for quality control of freshly made blends.

8.5.1 *Option A*—Clean the container of a 1-L (1-qt), high-speed blender[3] using the procedure given in 7.1.1. Place 500 mL of sample measured from 18 to 32°C (65 to 90°F) into the container, cover, and stir at maximum speed for 1 min. Because it is normal for considerable air to be entrained during this agitation, allow to stand until entrained bubbles have dispersed and the temperature of the oil has reached 24 ± 3°C (75 ± 5°F). Within 3 h following the agitation (Note 7), start with testing as specified in 8.2.

NOTE 7—In case of viscous oils, 3 h may not be enough to disperse the entrained air. If a longer time is required, record the time as a note on the results.

9. Report

9.1 Report the data in the following manner:

Test	Foaming Tendency ASTM D 892 IP 146 Foam Volume, mL, at end of 5-min blowing period	Foam Stability ASTM D 892 IP 146 Foam Volume, mL, at end of 10-min setting period
As received:		
Sequence I	...	...
Sequence II	...	...
Sequence III	...	...

Test	Foaming Tendency ASTM D 892 IP 146 Foam Volume, mL, at end of 5-min blowing period	Foam Stability ASTM D 892 IP 146 Foam Volume, mL, at end of 10-min setting period
After agitation: *(Option A, 8.5.1)*		
Sequence I	...	...
Sequence II	...	...
Sequence III	...	...

9.2 For the purpose of reporting results, when the bubble layer fails to completely cover the oil surface and a patch or "eye" of clear fluid is visible, the value shall be reported as "nil foam."

10. Alternative Procedure

10.1 For routine testing a simplified testing procedure may be used. This procedure differs from the standard method in only one respect. The total air volume used during the 5-min blowing period is not measured after the air has passed through the diffuser stone. This eliminates the volume measuring equipment and the airtight connections necessary to carry the exit air from the graduated cylinder to the volume measuring device, but requires that the flowmeter be correctly calibrated and that the flow rate be carefully controlled. Results obtained by this procedure shall be reported as D 892 - IP 146 (Alternative).

11. Precision

11.1 For the standard test (through 8.4), duplicate results by the same operator for foam volume at the end of the 5-min blowing period should be considered suspect if they differ by more than the amount indicated on Fig. 1 for "repeatability."

11.2 For the standard test (through 8.4): the results submitted by each of the two laboratories for foam volume at the end of the 5-min blowing period should not be considered suspect unless they differ by more than the amount indicated on Fig. 2 for "reproducibility."

11.3 For those oils which have been tested by Option A (8.5.1), no precision statement is available.

NOTE 8—The majority of the results in the cooperative work that led to Option A were nil foam; hence, no precision statement can be calculated.

[3] Waring Blender or equivalent.

Chapter VI
Attachment 10

SEA-CW-037
Page 192

OBSERVATIONS OF REMOVED PISTONS

D/G	Cylinder	Tin Smear on Non-Thrust Side of Liner	Blued Pin	End Caps	COMMENTS
C	5R	Piston involved in crankcase explosion. Non-thrust side tin smearing noted in previous inspection.	Yes.	Not seated.	
B	7L	Piston involved in crankcase explosion.	Slight bluing.	Not seated.	
D	8R	Yes.	Not blued, but pin has a 5 mil bend.		
A	1R	Yes.	Yes. Pin needed to be hammered out.	Not seated.	
A	2R	Yes.	Yes.	Not seated.	
A	7R	Yes.	Yes.	Not seated.	
B	1R	Yes.	No bluing, but pin is worn.*		
C	6L	Yes.	Scratched parallel to pin axis.	Not seated.	
B	7R	No tin smear. Burnishing due to end-cap rubbing.	Showed signs of wear. Pin was replaced.	Not seated.	Liner replaced. Piston replaced (non-thrust tin removed).
C	8R	Yes.	?	?	
A	3R	Good piston pulled for inspection. Some burnishing of tin on non-thrust side.	No, bushing needed to be scraped to obtain proper bluing.	?	

* Per Mike Schleigh of C-B.

fjc/tbk109c(9)

TELECOPY
COOPER-BESSEMER RECIPROCATING (412) 458-8000 EXT. 4184
COOPER-BESSEMER RECIPROCATING (412) 458-8000 EXT. 4184
GROVE CITY, PENNSYLVANIA 16127

DATE: October 16, 1989	SENT BY: Kathy EXT: 3395
TO: Frank Czysz	FROM: W. H. A. Lambert
COMPANY: P. P. and L.	NO. OF PAGES: 4 (One)
FAX NO. 215-770-7830	OUR REF. NO. QCG-6405

SUBJECT: P. P. & L. Report Draft

M E S S A G E
****

The following three pages are regarding the P.P. and L. crankcase explosion Oct. 7, 1989.

W. H. A. Lambert
Manager
Quality Assurance

QCG-6405

D R A F T

10/14/89

REPORT

Pennsylvania Power and Light Company
Standby Diesel/Generator
KSV-16-T S/N 7160 (Unit "C")
Crankcase Explosion - October 7, 1989

1.0 Introduction

1.1 On October 7, 1989 the "C" diesel was started at 22:28 hours on a twenty-four hour run. At 23:03 a fire alarm signal was transmitted to the Control Room and an operator observed smoke around crankcase doors following a crankcase explosion. The engine was shutdown. It was noted that this crankcase explosion was more severe than previously experienced with a large quantity of lube oil having been displaced and the liquid completely blown out of the crankcase manometer.

1.2 This engine had run 26 hours since the last "eighteen month inspection " which took place November 1988 and was restarted November 22, 1988. Total running hours of 796.9 have accumulated.

1.3 An internal inspection of the crankcase revealed major distress to the number 5 right piston and liner. During a site visit on October 12, 1989, the following observations were made by A. Lambert

2.0 Observations

2.1 The liner was very heavily damaged (burned and heat checked) in the lower half and for most of its circumference.

2.2 The piston skirt was extensively damaged with displaced skirt material tending to close over the lower oil ring which itself had suffered heavy wear on the "scraper" faces. Two "blow holes" towards the bottom of the skirt and located radially closer to one end of thepiston pin than the other end were in evidence. One hole was fairly round and about 5/16" in diameter and somewhat larger and irregular in shape. From the bottom of the larger hole a crack ran straight down to the bottom of the skirt (a distance of approx. 3/4"). About 2-1/2" from the larger hole and further around the skirt towards the piston pin bore there was a large vertical crack. The readily visible portion of this crack ran up from the skirt bottom for about 3". Towards the top end of the crack and for a distance of approximately 1-1/4",the piston skirt material was gouged (burned) out leaving a "canyon" with the crack in evidence at the bottom. The piston has been cut on the horizontal centerline of the piston bore, and sectioned vertically as well so it was not possible to see the complete piston. The rings were not inspected.

2.3 The piston pin had seized in its bushing and the bushing had rotated in the piston. The pin and bushing were forcibly removed from the piston and the bushing removed by cutting (slotting) through the bale at one end. At the other end the bale was cracked through indicating an area of greater distress than the opposite end. This cracked bale was in line with the "blow" holes and cracks in the piston skirt described in paragraph 2.2 above. Approximately 60% of the total bushing surface was destroyed by excessive heat. The remaining 40% was an irregular but connected area between the two bales and in the loaded part of the bushing. This area was indeed "clear" and had sustained absolutely no damage.

2.4 The lower liner seal was distorted when trapped water boiled off and turned to steam.

2.5 The piston pin end caps had no contact with the liner surface.

3.0 <u>Discussion</u>

3.1 The damage to the lower liner seal indicated that excessive heat was generated in the lower end of the liner.

3.2 Quality Control procedures at the C-B manufacturing facility would have required a piston skirt with "blow" holes as described in paragraph 2.1, to be scrapped. Therefore, it is concluded that a casting defect was present below the machined outer surface of the piston skirt. Some of the material subsequently displaced from this area of the skirt would have "fallen" into the crankcase and some, no doubt, "rolled" between the liner and piston skirt. This is the most likely explanation for the "canyon" (or gouge) observed in the vertical crack adjacent to the large "blow" hole.

3.3 The majority of the liner distress was observed to be in the lower portion indicating that contact with the piston was greatest at its skirt area.

3.4 Observations of the piston pin and bushing indicate very heavy loading in the upwards direction of the bales. This, together with the large undamaged portion of the normally loaded (down) area, is consistent with the rod pushing the piston upwards against a very heavy friction load - a load generated by heavy contact of the piston and liner.

4.0 Conclusion

4.1 The primary area of failure was that of the piston skirt. The root cause appears to stem from a sub-surface casting defect from which material loosened during engine operation and was trapped between the piston skirt and liner. This material was of sufficient dimension to roll up and down, destroy the lubrication and then generate sufficient heat to cause the piston skirt to expand and take up all available piston to liner clearance. With heavy rubbing of the piston skirt on the liner induced a further breakdown of the oil film occurred leading to more heat being generated to the point of incandescence which was the source of ignition for the crankcase explosion. (The lube oil vapor in the crankcase was the fuel which was ignited).

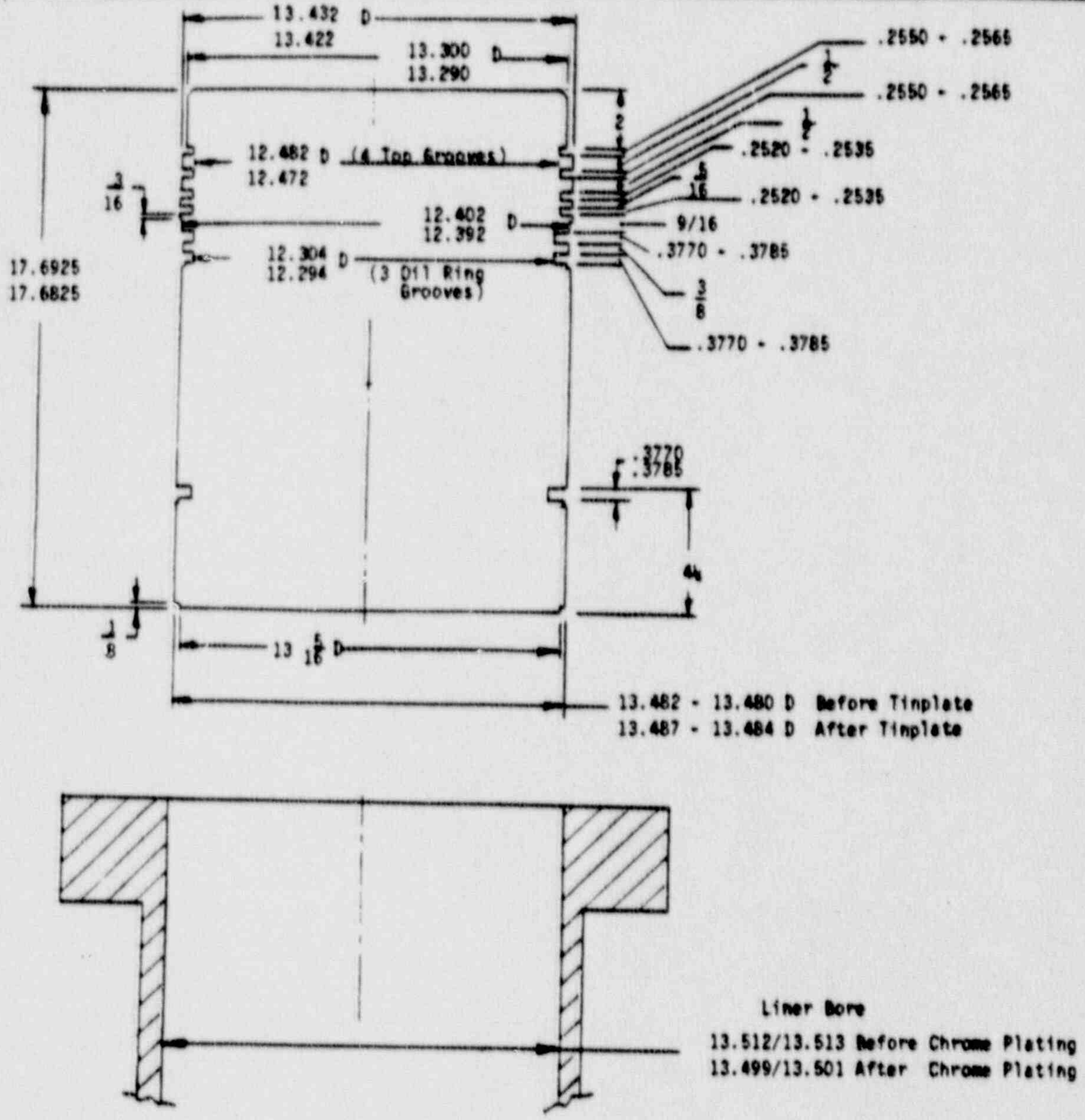
4.2 The damage to the piston pin and bushing is considered to be a secondary failure due to the abnormal loading described in paragraph 3.4.

W. H. A. Lambert/k

W. H. A. Lambert
Manager, Quality Assurance
and Nuclear Operations

cc: R. J. Brager
A. E. Bice
D. T. Blizzard
W. O. Ferguson
B. C. Guntrum - C-E
B. K. Hall
M. J. Helmich
J. M. Horne
W. Leishman - MV
R. A. Miklos
M. A. Schleigh
B. R. Sedelmyer
File: 5-1
SD-0188/P2
K5fa16

WHAL/k11



Ref Power Piston KSV-5-A
Liner-Power 61B-9-12A02

KSV PISTON/LINER - DETAIL

DRWN. BY BMcL

CHKD. BY

APPR. BY

COOPER ENERGY SERVICES

A COOPER-BESSEMER PRODUCT

REV. NO.		REVISION	DATE
ISSUE			NOV 20 '89

SCALE

SHEET 1 OF 1 SHEETS

QCS-6543

103 PRINTED IN U.S.A.

SUSQUEHANNA WORK AUTHORIZATION

WA NUMBER S74946

NEED DATE

I. PROBLEM DESCRIPTION

UNIT	SYSTEM	EQUIP NUMBER	LOCATION
0	24	0G501 C	A 44 676

WORK GROUP: MECH (circled) | ELEC | LABOR | MSER | I&C | CHEM | SUB GRP.

REMOVE 5 R PISTON TO INSPECT FOR CAUSE OF LINER SCUFFING.

Identified (Print): D Klum — Date: 7/13/87

Approved (Signature): Tom Wetmer — Date: 7-13-87

MODIFICATION RELATED ☐ YES ☒ NO PMR#

II. WORK CLASSIFICATION

QUALITY	☒ YES ☐ NO	ASME CODE	☐ YES ☒ NO
EQUIP QUAL	☐ YES ☒ NO	TECH SPEC	☒ YES ☐ NO
RELIABILITY RELATED	☐ YES ☐ NO		

ACCT. or ER NO. 742622 — LOC. or CODE 012

Work Group Supervisor: D K Mautz — Date: 7/14/87

III. WORK PLANNING

INITIAL INPUT STATUS & AP	PLAN CODE	PRIORITY CODE	WORK CODE	PLANT STATUS CODE
P/TC		3	2	ALL

A. INVESTIGATION

Work Group Foreman — Date

Operations Authorization — Date

B. WORK PLAN

SEE ATTACHED WORK PLAN

ERF NO A60029 — ERF REQ ✓YES ☐ NO — HP REVIEW ☐ YES ✓NO

RWP NO N/A

4 men 30 hrs — EST Man Hrs. 120 — RWP Man Hrs

Work Group Foreman: D Klum — Date: 7/15/87

Work Group Supervisor: S Moh — Date: 7-15-87

CONTACT QC @ X3777

C. POST MAINTENANCE TESTING REQUIREMENTS

PER ERF OPERATIONAL TESTING

Quality Control Review: B A Kulitsky — Date: 7-15-87

Operations Authorization: ERF A60029 — Date

IV. ACTION TAKEN - EQUIPMENT HISTORY SUMMARY

Inspected Piston and cylinder wall with Vendor Rep. present. The SCUFFed area on piston amounted to Just having the tin Coating rubbed OFF. The cylinder liner was polished by the Piston. witch is common. Overall the piston, Rings and liner looked very good.
Reinstalled piston And torque master rod to piston per MT-024-002.

87-2439

Work Completed: Bob Satterson — Date: 7-28-87

ERF SIGNED OFF ☐ NA ☑ YES — Date: 7/27/87

Work Group Review: Chet Borden — Date: 7/28/87

Quality Control Review: [illegible] — Date: 7-29-87

Operations Review: Kurt A Towell — Date: 7-29-87

V. PROBLEM CODE

Copy To Planner	PROBLEM	CAUSE	ACTION	PMIS
☐ Yes ☐ No	XX	XX	XX	[illegible]

SUSQUEHANNA SES WORK AUTHORIZATION

CONTINUATION SHEET

SECTION NO. TITLE	UNIT	SYSTEM	WA NUMBER
IIIB WORK PLAN	0	24	574946

I. GENERAL SCOPE.

"C" D/G 5R PISTON WILL BE REMOVED TO ALLOW FOR INSPECTION OF LINER TO DETERMINE POTENTIAL CAUSE OF LINER SCUFFING. THIS WORK IS BEING DONE IN CONJUNCTION WITH 18 MONTH SURVEILLANCE AND CYLINDER HEAD HAS ALREADY BEEN REMOVED PER WA 574076. AT THIS TIME IT IS NOT PLANNED TO PULL THE LINER OR THE ~~ARTICULATED~~ MASTER CONNECTING ROD, IF LINER REQUIRES REMOVAL, ADDITIONAL INSTRUCTIONS MAY BE REQUIRED.

II. REFERENCES.

A. MT-024-002 - EMERG. DIESEL PISTON/CONNECTING ROD/ EXPANSION SEAL REMOVAL AND INSTALLATION

III. PRECAUTIONS AND CONDITIONS.

A. SINCE THIS WORK IS BEING PERFORMED ALONG WITH NUMEROUS OTHER ACTIVITIES UNDER THE 18 MONTH SURVEILLANCE, IT MUST BE CLOSELY COORDINATED TO PREVENT PERSONNEL AND EQUIPMENT DAMAGE. (I.E. WHEN ENGINE MUST BE ROTATED).

B. MOST PREREQUISITE STEPS AND SEVERAL OF THE PROCEDURAL STEPS HAVE ALREADY BEEN PERFORMED UNDER OTHER WA'S. THE FOLLOWING WORK INSTRUCTIONS WILL CONTINUE FROM THAT POINT TO ALLOW PISTON REMOVAL AND INSTALLATION, THEN THE OTHER WA'S WILL REINSTALL THE HEAD, IMPULSE PUMP, ETC.

C. ASSISTANT FOREMAN WILL DETERMINE WHAT STEPS OF MT-024-002 ARE REQUIRED TO PERFORM THE TASK

SUSQUEHANNA SES WORK AUTHORIZATION

CONTINUATION SHEET	SECTION NO. TITLE	UNIT	SYSTEM	WA NUMBER
	IIIB Work Plan	C	24	574946

HE DESIRES. IF ADDITIONAL DISASSEMBLY OR REWORK IS REQUIRED THAT IS BEYOND THE SCOPE OF MT-024-002, ADDITIONAL INSTRUCTIONS MAY BE NECESSARY.

D. ALL STEPS NOT PERFORMED THAT REQUIRE QC INSPECTION SHOULD BE "NA'D" ON THE QUALITY DATA SHEET AND REASON GIVEN (IE. STEP NOT PERFORMED) AS TO WHY.

IV WORK FLOW

A. PER SECTION 6.4, OF MT-024-002, REMOVE 5R PISTON TO ALLOW FOR INSPECTION.

1. INSPECT LINER AND PISTON FOR POTENTIAL CAUSE OF SCUFFING.
2. IF REWORK OF EITHER IS REQUIRED, ADDITIONAL WORK INSTRUCTIONS MAY BE REQUIRED. ASST FOREMAN TO MAKE DETERMINATION. IF REPLACEMENT OF COMPONENTS IS REQUIRED AND CAN BE DONE IN ACCORDANCE WITH PROCEDURE, DO SO, IF NOT RETURN FOR ADDITIONAL INSTRUCTIONS.
3. IF FURTHER DISASSEMBLY IS REQUIRED FOR INSPECTION DO SO IN ACCORDANCE WITH PROCEDURE.

B. ONCE INSPECTION/REWORK IS COMPLETE, REINSTALL PISTON PER SECTION 6.9 OF MT-024-002.

1. IF OTHER COMPONENTS WERE REMOVED, REINSTALL PER APPLICABLE STEPS OF MT-024-002.

C. RECORD ALL WORK PERFORMED AND INSPECTION FINDINGS IN ACTION TAKEN.

Tony Lewis 7/15/82

EQUIPMENT RELEASE FORM

ERF No. A 60029

SYSTEM NO. 024 PERMIT NO. ______ (Ops Use Only) W.A. NO. S74446

REQUESTING GROUP SECTION

TECH SPEC / SAFETY RELATED ITEM: YES ✓ NO ____ QUALITY RELATED: YES ✓ NO ____

EQUIPMENT REQUESTED OG-501C STANDBY DIESEL GENERATOR

TIME REQUESTED: FROM 0700 7 / 15 / 87 (Time Mo Day Year) TO 1930 7 / 24 / 87 (Time Mo Day Year)

REASON To Remove 5R Piston To Inspect Piston and Cyl. Liner To Find Cause of Liner Scuffing.

PERMIT HOLDER N/A WORK GROUP MM

BLOCKING REQUESTED

Blocking Covered Under WA A71781 ERF A61777

BLOCKING REVIEWED BY ______ (Electrical / I&C Blocking)

TECH. SPEC. LIMITS

TECH. SPEC. NO. / CONDITION 3/4. 8.1 AC Sources

REQUIRED OPERATIONAL TESTING ST-024-001

REQUESTED BY Larry Weitman (Work Group Supervision) DATE 7-14-87

RELEASE EQUIPMENT TO WORK GROUP [signature] (Shift Supervision) 0715 / 7 / 15 / [illegible] (Time Mo Day Year)

ADDITIONAL TESTING ______

Torquing Data Sheet

	Required Torque to be determined by Supervision (optional)	Actual Torque Applied	Source of Torque Value (IOM, Drawing, Attach.)
Desired Torque Value	30 Ft LBS	30 Ft LBS	IOM 183
Upper Limit on Torque (Required to Control Leakage, etc.)		N/A	

Description of item or joint being torqued: Primary half of SR Piston Pin bolt antirotational brake nuts

Fastener size and grade: ______

Tool/Test Instrument	I.D. Number	Calibration Due Date
torque wrench	mm 149A	9-23-87
		6-23-87

Work Performed BY: Sattleson (Name) 7-15-87 (Date)

Torquing Data Sheet

	Required Torque to be determined by Supervision (optional)	Actual Torque Applied	Source of Torque Value (IOM, drawing, Attach.)
Desired Torque Value	20 Ft LBS	20 Ft LBS	IOM 183
Upper Limit on Torque (Required to Control Leakage, etc.)		N/A	

Description of item or joint being torqued Secondary half of Piston Pin Bolt anti-Rotation drake nuts

Fastener size and grade

Tool/Test Instrument	I.D. Number	Calibration Due Date
torque wrench	mg 149A	9-22-87 6-23-87

Work Performed BY: Sattkon 12-15-87
Name Date

Torquing Data Sheet

	Required Torque to be determined by Supervision (optional)	Actual Torque Applied	Source of Torque Value (IOM, drawing, Attach.)
Desired Torque Value	690 Ft LBS	690 Ft LBS	IOM 183
Upper Limit on Torque (Required to Control Leakage, etc.)		N/A	

Description of item or joint being torqued 5R CYL master Rod to Piston bolts

Fastener size and grade Big

Tool/Test Instrument	I.D. Number	Calibration Due Date
torque wrench	mm 107	6-24-87 9-24-87

Work Performed BY: Satterson (Name) 7-15-87 (Date)

EMERGENCY DIESEL PISTON/CONNECTING ROD/EXPANSION SEAL
REMOVAL AND INSTALLATION

QUALITY DATA SHEET

DIESEL 0G501C CYLINDER 5R

Quality Hold Point

Step	Verified by	Date	Description
6.7.5	N/A not Performed		Expansion seal split flange and flange cap screws are torqued to 60 ft. lbs.
6.8.3	N/A		Bearing cap studs are torqued to 615 ft. lbs.
6.8.8	N/A		Bearing cap stud nuts are torqued to stud stretch of 0.016" to 0.018".
6.8.9.d	N/A		Clearance between the detector and trip lever is 0.100".
6.9.9	D. Herring	7-16-87	Piston pin bolts are torqued to 690 ft. lbs.
6.9.11	D. Herring	7-16-87	Primary half of the piston pin drake nuts are torqued to 30 ft. lbs.
6.9.12	D. Herring	7-16-87	Secondary half of the piston pin drake nuts are torqued to 20 ft. lbs.
6.10.8	N/A not Performed		Two expansion seal cap screws are torqued to 60 ft. lbs.
6.10.9	N/A not Performed		Rod pin bolts are torqued to 1140 ft. lbs.
6.10.11	N/A not Performed.		Primary half of the rod pin drake nuts are torqued to 30 ft. lbs.

8707210274

QUALITY DATA SHEET

DIESEL 1G501G CYLINDER 5R

Step	Verified by	Date	Description
6.10.12	N/A	Not Performed.	Secondary half of the rod pin drake nuts are torqued to 20 ft. lbs.

SUSQUEHANNA SES WORK AUTHORIZATION

ACTION TAKEN CONTINUATION SHEET	SECTION NO. TITLE	UNIT	SYSTEM	WA NUMBER
	IV action taken	0	24	S-74946

NAME: Satterson, Hockenberg, Zielinski, Bainbridge

DATE	ACTION TAKEN
7-15-87	during cyl Liner insp under WA A71781 Found 5R Liner to be Scuffed in 4:00 oclock to 8:00 oclock area Standing on Cat walk Looking towards left Side. Removed 5R Piston Found Piston to be Scuffed in that area but only down to the point of having the Tin coating on it rubbed off. the Liner is NOT hurt at all only that the Piston has polished it. cleaned Piston + Liner. the Rings Looked very good yet. Reinstalled Piston. torque master Rod to Piston bolts See Data Sheets. Cooper Rep agreed with our conclusion. He also call the Factory to be Sure they agreeded, they did.

FORM AD-QA-502-3, Rev. 1 PART 1 — ORIGINAL PART 2 — COPY

PP&L Form 2454 (10/83)
Cat #973401

Chapter VI
Attachment 15

Dept. MAINT
Date 10/16 1989
Designed by JEA
Approved by

PENNSYLVANIA POWER & LIGHT COMPANY
CALCULATION SHEET

PROJECT A-D D/G
SIGNIFICANT OPERATING HOURS AND DATES

ER No.
Sht. No. ____ of ____

	A	B	C	D
TOTAL ENGINE HOURS	981.6	1033.1	796.9	965.5
HOURS SINCE 1/1/87	176.3	203.1	111.6	141.6
HOURS SINCE LAST 5 YEAR SURV. INSPECTION	101.5	126.3	90.4	91.2
DATE OF LAST 5 YEAR SURV INSPECTION	6/30/87	1/8/88	8/1/87	8/24/87
HOURS SINCE LAST 18 MO SURV. INSPECTION	49.3	39.6	26.7	23.6
DATE OF LAST 18 MO SURV INSPECTION	11/6/88	3/11/89	11/22/88	2/2/89
HOURS FROM 18 MO SURV TO CRANKCASE EXPLOSION	—	11.2	26.7	UNKNOWN
DATE OF LAST CRANKCASE EXPLOSION	—	9/16/89	10/7/89	1/84

NOTE (1): 'E' D/G RUN TIMES HAVE NOT BEEN CALCULATED

NOTE (2): 'B' D/G HAD A CRANKCASE EXPLOSION ON 1/86

NOTE (3): D D/G HAD A CRANKCASE EXPLOSION ON 1/84

 10-12-89

JIM,

THE 'A' DG HAS RUN 49 HRS 18 MIN. SINCE ITS LAST INSPECTION ON 11-6-88. FIRST START AFTER THIS INSPECTION WAS # 542.

DG 'A' HAS RUN 101 HRS 27 MIN SINCE ITS 5 YR. INSPECTION. HOUR COUNT BEGAN WITH START # 493.
(6/30/87)

A

Start #	Date	Time	Length	Loaded	Reason
460	12-26-86	8.15	1:14	YES	SO-024-001A
461	1-9-87	9.0	1:14	YES	SO-024-001A
462	1-21-87	8.6	1:15	YES	SO-024-001A
463	1-27-87	7.9	27	YES	SO-024-013
464	1-28-87	7.88	22	YES	SO-024-013
465	2-3-87	8.6	1:15	YES	SO-024-001
466	2-4-87	8.53	:25	YES	SO-024-013
467	2-11-87	8.46	:20	YES	SO-024-013
468	2-19-87	8.33	1:12	YES	SO-024-001
469	3-4-87	NA	:29	NO	TROUBLESHOOTING
470	3-4-87	NA	:07	NO	"
471	3-4-87	NA	:12	NO	"
472	3-4-87	NA	:24	NO	"
473	3-4-87	NA	:21	NO	"
474	3-4-87	NA	:08	NO	"
475	3-4-87	NA	:12	NO	"
476	3-4-87	8.3	1:03	YES	SO-024-001A
477	3-5-87	NA	:12	NO	UPLANNED EMERGENCY START
478	3-19-87	8.3	1:08	YES	SO-024-001
479	4-1-87	8.22	1:12	YES	SO-024-001
480	4-6-87	8.6	:19	YES	SO-024-013
481	4-15-87	7.9	1:13	YES	SO-024-001
482	4-30-87	8.23	1:11	YES	SO-024-001
483	5-7-87	8.09	1:12	YES	SO-024-013, SO-024-001
484	5-9-87	7.9	:09	NO	SO-024-013, SO-024-001
485	5-13-87	7.77	1:14	YES	SO-024-001A
486	5-19-87	8.1	1:10	YES	SO-024-001A
487	6-2-87	7.99	1:10	YES	SO-024-001A
488	6-6-87	8.1	1:16	YES	SO-024-001
489	6-8-87	8.3	1:13	YES	SO-024-001
490	6-8-87	8.07	4:26	YES	SO-024-001
491	6-28-87	12.1	:29	NO	POST MAINTENANCE TEST
492	6-28-87	7.7	1:09	YES	POST MAINT TEST
493	6-30-87	8.64	7:58	YES	" " "
494	7-6-87	8.13	6:59	YES	" "
495	7-6-87	7.82	1:17	YES	SO-024-001
496	7-20-87	8.12	1:09	YES	SO-024-001A
497	8-3-87	8.54	1:14	YES	SO-024-001A
498	8-17-87	8.34	1:22	YES	SO-024-001

1 of

A

Start #	Date	Time	Length	Loaded	Reason
499	8-31-87	7.68	1:12	YES	SO-024-001
[illegible]00	9-15-87	8.48	1:26	YES	SO-024-001
501	9-28-87	8.07	1:13	YES	SO-024-001
502	10-12-87	8.24	1:10	YES	SO-024-001A
503	10-26-87	7.86	1:20	YES	NA
504	11-8-87	8.16	1:15	YES	SO-024-001A
505	11-23-87	8.76	1:16	YES	SO-024-001
506	12-6-87	7.9	1:11	YES	SO-024-001A
507	12-21-87	7.98	1:20	YES	SO-024-001A
508	1-4-88	8.14	1:19	YES	SO-024-001A
509	1-14-88	8.16	1:30	YES	SO-024-001A AND VENDOR OBSERVATION
510	2-2-88	8.04	:09	YES	SO-024-001A
511	2-2-88	7.69	1:19	YES	MAINT. AND I&C INVESTIGATION
512	2-2-88	7.43	1:18	YES	SO-024-001A
513	2-15-88	8.25	2:03	YES	SO-024-001A
514	2-15-88	NA	:20	YES	OP-024-001
515	2-29-88	8.02	1:15	YES	SO-024-001
516	3-14-88	8.15	1:14	YES	SO-024-001
[illegible]7	3-28-88	8.16	1:29	YES	SO-024-001
518	4-9-88	8.23	27:35	YES	SO-024-001 & SE-024-A05
519	4-10-88	7.9	:12	YES	SE-224-A03
520	4-22-88	8.04	1:18	YES	SO-024-001A
521	4-28-88	7.8	3:11	YES	SO-024-001A
522	4-28-88	8.44	:50	YES	SE-224-A02
523	5-3-88	8.84	1:38	YES	SE-224-107
524	5-12-88	8.02	1:14	YES	SO-024-001A
525	5-13-88	8.2	:14	NO	INVESTIGATION OF ALARM FROM START #524.
526	5-25-88	8.22	1:16	YES	SO-024-001A
527	6-8-88	7.82	1:15	YES	SO-024-001A
528	6-22-88	8.32	1:15	YES	SO-024-001A
529	7-6-88	8.27	1:14	YES	SO-024-001A
530	7-28-88	8.7	1:40	YES	SO-024-001A
531	8-3-88	8.79	1:16	YES	SO-024-001A
532	8-27-88	9.29	:39	YES	TROUBLESHOOTING FOR I&C
533	8-27-88	9.1	1:10	YES	SO-024-001A
5[illegible]	9-10-88	8.2	1:19	YES	SO-024-001A
[illegible]	9-25-88	8.56	1:17	YES	SO-024-001A
536	10-17-88	8.52	3:19	YES	4 HR. ANALYZER RUN
537	10-11-88	8.2	1:19	YES	SO-024-001A

"A"

Start #	Date	Time	Length	Loaded	Reason
538	10-24-88		4.49		
539	11-6-88		:22		MAINT. TESTING
540	11-6-88		:05		MAINT. TESTING
541	11-6-88		:12		"
542	11-6-88	7.78	1:12		SO-024-001
543	12-5-88	8.46	NA		
544	12-5-88	NA	NA		FAILURE TO START
545	12-5-88	NA	NA	N	TROUBLESHOOTING
546	12-5-88	7.6	:11	N	TROUBLESHOOTING
547	12-5-88	8.1	1:15	Y	N/A
548	1-4-89		1:15		
549	1-29-89		:21		
550	2-2-89		1:21		
551	3-1-89		1:24		
552	3-11-89		:15		
553	3-22-89		:10	NO	QUICK START PER SO-024-013 & SO-024-001A
554	3-30-89		1:21		
555	4-29-89		:10		
556	4-29-89		<:01	N	
557	4-29-89		1:15	YES	
558	5-18-89		3:22		
559	5-18-89		:43		SE-124-A02
560	5-19-89		:57		
561	6-19-89		1:24		
562	7-6-89		:15		
563	7-16-89		:36		
564	7-16-89		1:30		SO-024-001A
565	8-15-89		1:18		
566	8-15-89		:29		
567	8-15-89		1:23		
568	9-3-89		1:15		
569	9-16-89		1:25		SO-024-001A
570	10-6-89		<:01		SE-024-A05
571	10-6-89	7.84	24:33		SE-024-A05

JIM,

THE 'B' DG HAS RUN 39 HRS. 38 MIN. SINCE ITS LAST INSPECTION ON 3-11-89. ITS FIRST START AFTER INSPECTION WAS # 537.

THE 'B' DG WAS RUN 126 HRS. 15 MIN. SINCE ITS 5 YR. INSPECTION. BEGAN COUNT ON DG START # 497.

'B' Diesel

Start #	Date	Time	Length	Loaded	Reason
449	12/30/86	7.4	1:17	Y	SO-024-001
450	1/6/87	7.7	1:16	Y	
451	1/13/87	8.52	1:18	Y	
452	1/20/87	8:18	1:10	Y	
453	1/27/87	8.42	1:20	Y	
454	1/28/87		:01	N	Post maintenance. DG tripped SCR
455	1/28/87		:02	N	Post maintenance
456	1/28/87		:04	N	" "
457	1/28/87		:02	N	" "
458	1/29/87		:03	N	" "
459	1/29/87		:05	N	" "
460	1/29/87		:01	N	" "
461	1/29/87	8.36	1:11	Y	SO-024-001
462	2/2/87	8.31	1:19	Y	SO-024-001
463	2/4/87	8.16	:21	Y	SO-024-013
464	2/10/87	7.08	1:15	Y	SO-024-001
465	2/11/87	8.44	:33	Y	SO-024-013
466	2/19/87	8.43	1:30	Y	SO-024-001
467	2/25/87	8.17	1:13	Y	SO-024-001
468	3/3/87	8.31	1:12	Y	"
469	3/11/87	8.13	1:08	Y	"
470	3/17/87	8.29	1:15	Y	"
471	3/17/87	8.46	:20	Y	SO-024-013 3GM
472	3/24/87	9.31	1:08	Y	SO-024-001
473	3/31/87	8.28	1:15	Y	"
474	4/6/87	8.3	:18	Y	SO-024-013
475	4/14/87	8.2	:39	Y	SO-024-001 Fuel oil leak, governor
476	4/14/87	8.2	:16	Y	Post maintenance
477	4/14/87	8.19	1:15	Y	SO-024-001
478	4/30/87	8.22	1:11	Y	"
479	5/8/87		1:20	N	Post Maintenance SCR tuneup
480	5/9/87	7.8	1:10	Y	SO-024-001
481	5/9/87	7.9	:05	N	SO-024-013
482	5/18/87	8.14	:03	N	"
483	6/9/87	8.63	1:13	Y	SO-024-001
484	7/9/87	8.26	1:14	Y	"
485	8/7/87	8.87	1:16.	Y	"
486	9/15/87	8.81	:38	Y	Post Maintenance I+C

[redacted]

Diesel Start #	Date	Time	Length	Loaded	Reason
487	9/13/87	8.03	1:17	Y	SO-024-001
488	10/3/87	8.55	1:15	Y	"
489	11/2/87	7.94	1:45	Y	SE-124-B02
490	11/2/87		:48	Y	"
491	11/3/87	9.1	:55	Y	SE-124-207
492	11/12/87	8.3	1:16	Y	SO-024-001
493	11/16/87	8.0	10:15	Y	SO-024-001, 4 hr. run, hot web deflectio
494	1/4/88		<:01	N	Trip on Turbo Thrust Bearing Fa
495	1/4/88		<:01	N	"
496	1/4/88	8.64	1:31	Y	Post maintenance,
497	1/7/88	8.35	1:30	Y	SO-024-001
498	1/13/88	8.3	2:12	Y	", vendor insp., trip on Con. Rod Brg.
499	1/13/88	8.32	1:21	Y	"
500	2/8/88	8.47	1:17	Y	"
501	3/11/88	8.36	1:15	Y	"
502	3/18/88	8.15	1:59	Y	SO-024-B05
503	3/25/88	9.64	1:17	Y	SO-024-001
504	3/25/88	8.4	16:11	Y	SE-024-B05, Fuel Oil leak
505	3/28/88	9.54	24:32	Y	SE-024-B05
506	4/21/88	8.33	1:20	Y	SO-024-001
507	5/4/88	8.0	:28	Y	SE-224-B02
508	5/5/88	8.71	1:02	Y	SE-224-207
509	5/21/88	9.71	1:18	Y	SO-024-001
510	6/20/88	8.88	1:16	Y	"
511	7/1/88	8.57	1:32	Y	", SE-124-B03
512	7/1/88	8.25	:08	N	SE-024-B04
513	7/21/88	8.28	:25	N	SO-024-001, VR failure
514	7/21/88	8.45	1:31	Y	"
515	8/19/88	7.92	1:17	Y	"
516	9/14/88	9.8	1:25	Y	"
517	10/4/88	8.7	1:21	Y	"
518	10/15/88	6.69	:18	N	Post maintenance
519	10/16/88	7.87	:14	N	" "
520	10/19/88	9.03	1:08	Y	SO-024-001
521	11/11/88	9.85	1:13	Y	"
522	12/12/88		<:01	N	Failed to start
523	12/12/88	8.86	:16	N	Post maintenance
524	12/12/88	8.21	1:27	Y	SO-024-001
525	1/3/89	9.02	6:38	Y	", analyzer run

	Date	Time	Length	Loaded	Reason
526	1/3/89	10.18	:15	N	Post mainterance
527	1/3/89	9.85	:12	N	" "
528	1/29/89	15.01	:21	N	SO-024-001, slew start
529	1/29/89	9.4	:07	N	SO-024-013
530	1/30/89	10.04	:10	N	Post maintenace
531	1/30/89	7.78	1:21	Y	SO-024-001
532	2/6/89	9.97	5:43	Y	SO-024-001, Hot web
533	3/11/89		:06	N	Post maintenance
534	3/11/89		:41	N	" "
535	3/11/89		:34	N	" "
536	3/11/89	8.02	1:14	Y	SO-024-001
537	3/22/89	8.38	:06	N	SO-024-013
538	4/4/89	8.30	1:11	Y	SO-024-001
539	5/3/89	7.14	1:11	Y	"
540	5/23/89	8.2	1:35	Y	"
541	6/26/89	8.23	1:14	Y	"
542	7/26/89	8.37	1:14	Y	"
543	8/11/89		:12	N	Post maintenance
544	8/11/89		:12	N	SE-024-B04
545	8/11/89		1:24	Y	SO-024-001
546	9/4/89	8.2	1:25	Y	"
547	9/16/89	8.17	1:24	Y	SE-024-B05, Crankcase explosion
548	9/23/89	8.80	1:29	Y	OP-024-001
549	9/23/89	8.78	1:15	Y	SO-024-001
550	9/24/89	8.09	24:38	Y	SE-024-B05
551	10/6/88	8.0	1:08	Y	SO-024-001B

 10-12-89

DG 'C' RAN 26 HRS 42 MIN. SINCE ITS LAST INSPECTION (11-22-88). HOUR COUNT BEGAN WITH START # 526.

DG 'C' RAN 90 HRS. 22 MIN SINCE ITS 5 YR INSPECTION (8-1-87). HOUR COUNT BEGAN WITH START # 494.

"C"

Start #	Date	Time	Length	Loaded	Reason
471	12-26-86	7.76	1:16	YES	SO-024-001C
472	1-21-87	8.12	1:16	YES	SO-024-001C, SE-124-C03
473	1-27-87	7.9	:25	YES	SO-024-013
474	1-27-87	7.34	:20	YES	SO-024-013
475	1-28-87	7.88	:24	YES	SO-024-013
476	2-4-87	*	:05	NO	POST MAINTENANCE TESTING
477	2-4-87	*	:09	NO	POST MAINTENANCE TESTING
478	2-4-87	*	:02	NO	" "
479	2-4-87	*	:00	NO	" "
480	2-4-87	*	:02	NO	" "
481	2-4-87	7.6	1:20	YES	SO-024-001C
482	2-11-87	7.92	:20	YES	SO-024-013
483	3-3-87	8.08	1:16	YES	SO-024-001
484	4-2-87	8.01	1:20	YES	SO-024-001C, SE-224-C03
485	4-6-87	8.3	:18	YES	SO-024-013
486	5-3-87	8.02	1:12	YES	SO-024-001
487	5-7-87	8.03	1:13	YES	SO-024-013, SO-024-001
488	5-10-87	7.8	1:17	YES	SO-024-001
489	5-18-87	8.28	:06	NO	SO-024-013, SO-024-001
490	6-9-87	7.98	1:19	YES	SO-024-001
491	6-11-87	8.07	1:14	YES	SO-024-013
492	7-8-87	9.3	7:00	YES	SO-024-001C
493	7-31-87	—	:00	NO	POST MAINT.
494	8-1-87	10.2	:07	NO	POST MAINT. RUN
495	8-1-87	8.85	1:21	YES	POST MAINT. RUN
496	8-1-87	9.2	2:56	YES	POST MAINT. RUN
497	8-1-87	8.2	1:09	YES	SO-024-001
498	9-1-87	8.91	1:12	YES	SO-024-001C
499	10-1-87	8.14	1:17	YES	SO-024-001
500	10-21-87	8.03	1:29	YES	SO-024-001
501	10-31-87	7.7	1:05	YES	SE-124-C02
502	10-31-87	8.2	1:15	YES	SE-124-C07
503	11-25-87	7.8	:20	NO	SE-028-C01
504	11-25-87	7.6	:06	NO	SE-028-C01
505	11-25-87	8.23	:14	YES	SO-024-001C
506	12-23-87	7.89	:11	YES	SO-024-001C
507	12-23-87	8.11	1:13	YES	SO-024-001C
508	1-14-88	8.3	3:47	YES	SO-024-001C
509	2-12-88	8.2	1:18	YES	SO-024-001C

1 of

"C"

Start #	Date	Time	Length	Loaded	Reason
510	3-14-88	8.24	1:14	YES	SO-024-001C
511	4-7-88	8.5	:29	YES	SO-024-001, SE-024-C05
512	4-7-88	7.8	:07	NO	SO-024-001, SE-024-C05
513	4-7-88	7.3	24:37	YES	SO-024-001C, SE-024-C05
514	5-14-88	10.0	1:25	YES	POST MAINT. TESTING
515	5-14-88	8.02	:08	NO	POST MAINT. TESTING
516	5-14-88	7.85	—	—	" " "
517	5-14-88	7.88	1:20	YES	SO-024-001C
518	6-13-88	8.02	1:18	YES	SO-024-001C
519	7-13-88	8.40	1:17	YES	SO-024-001C
520	8-11-88	8.73	1:30	YES	SO-024-001C
521	9-9-88	7.9	1:15	YES	SO-024-001C
522	9-21-88	8.18	:05	NO	SO-024-001C
523	10-10-88	7.8	1:27	YES	SO-024-001C
524	10-18-88	7.5	3:35	YES	SO-024-001C – 4 HR ANALYZER RUN
525	11-7-88	7.04	5:03	YES	SO-024-001C – 4 HR ANALYZER RUN
526	11-22-88	20.26	2:20	YES	POST MAINT. START
527	11-22-88	7.8	:50	NO	" " "
528	11-22-88	7.6	1:21	YES	SO-024-001C
529	12-22-88	8.4	1:19	YES	SO-024-001C
530	1-20-89	7.61	1:09	YES	SO-024-001C
531	1-29-89	8.45	:13	NO	SO-024-013, SO-024-001C
532	2-17-89	8.2	1:00	YES	SO-024-001C
533	2-17-89	8.52	1:40	YES	SO-024-001C
534	3-11-89	8.4	:10	NO	SO-024-013, SO-024-001C
535	3-23-89	8.32	:10	NO	OP-024-001; 3.6; POST MAINT TEST ON VOLTAGE REG.
536	3-23-89	6.32	:13	NO	" "
537	3-23-89	9.04	:12	YES	SO-024-001C
538	3-23-89	6.95	:12	NO	TROUBLESHOOTING TRIP OF START # 537
539	3-23-89	8.64	:38	YES	" " " " "
540	3-23-89	8.95	1:09	YES	SO-024-001C
541	4-20-89	7.8	:09	NO	SE-024-C04
542	4-20-89	7.8	1:37	YES	SO-024-001
543	5-19-89	8.2	2:57	YES	SO-024-001C
544	5-19-89	8.2	:47	YES	SE-124-C02
545	5-19-89	8.15	1:20	YES	SE-124-107
[illegible]	6-19-89	8.05	1:13	YES	SO-024-001C
547	7-30-89	8.00	:09	YES	SO-024-001
548	7-30-89	8.09	:10	YES	TROUBLESHOOTING

"C

Start #	Date	Time	Length	Loaded	Reason
549	7-31-89	10.12	:12	YES	POST MAINT START AFTER WORK ON GOV.
550	7-31-89	9.87	:17	YES	TROUBLESHOOTING
551	8-1-89	7.9	1:44	YES	SO-024-001C
552	8-1-89	NA	:08	NO	SE-024-004
553	8-30-89	7.94	1:22	YES	SO-024-001C
554	9-16-89	8.15	1:22	YES	SO-024-001C
555	10-7-89	8.45	:39	YES	SE-024-005

JIM,

DG 'D' RAN 23 HRS 1 MIN SINCE LAST INSPECTION ON 2-5-89. THE FIRST START AFTER THE INSPECTION WAS #531.

DG 'D' RAN 91 HRS. 11 MIN. SINCE THE 5 YR INSPECTION. BEGAN HOUR COUNT FROM START #494

D Diesel

Start #	Date	Time	Length	Loaded	Reason
465	12/26/86	8.17	1:10	Y	SO-024-001
466	1/7/87	8.0	:08	N	SE-024-D04
467	1/7/87	8.43	1:15	Y	SO-024-001
468	1/21/87	8.25	1:16	Y	"
469	1/27/87	8.3	:26	Y	SO-024-013
470	1/27/87	8.04	:18	Y	"
471	1/28/87	8.30	:24	Y	"
472	2/3/87	7.92	1:13	Y	SO-024-001
473	2/4/87	7.94	:26	Y	SO-024-013
474	2/7/87		:27	N	Unplanned Emergency start
475	2/8/87	8.23	:09	N	Post Maintenance
476	2/11/87		:17	N	" "
477	2/11/87	7.59	1:15	Y	SO-024-001
478	3/11/87		:47	N	Post Maintenance
479	3/11/87		:25	N	" "
480	3/11/87		:09	N	" "
481	3/11/87		:04	N	" "
482	3/11/87	7.8	1:17	Y	SO-024-001
483	4/7/87	7.79	1:10	Y	"
484	5/7/87	8.37	1:09	Y	"
485	5/7/87	7.80	:29	Y	Post Maintenance
486	5/9/87	8.3	:11	N	SO-024-013
487	5/18/87	8.25	:03	N	"
488	6/5/87	8.92	1:08	Y	SO-024-001
489	6/11/87	8.3	1:16	Y	SO-024-013
490	6/25/87	8.68	1:08	Y	SO-024-001, SE-224-D03
491	6/25/87	8.13	:35	Y	SE-224-D03
492	7/24/87	8.4	1:14	Y	SO-024-001
493	8/3/87	8.51	5:45	Y	SO-024-001, hot web
494	8/29/87		1:01		Trip on turbo bearing failure
495	8/29/87	8.76	:21		Post Maintenance
496	8/29/87	8.69	1:13		" ", Trip on Hi Vibes, hi JW
497	8/29/87	8.7	1:31	Y	" "
498	8/29/87	8.63	1:12	Y	SO-024-001
499	9/28/87	8.7	1:12	Y	"
500	10/5/87	8.64	:16	Y	Post Maintenance
501	10/30/87	8.31	1:29	Y	SO-024-001
502	11/3/87	8.72	1:29	Y	", SE-124-D02
503	11/3/87	8.1	:32	Y	SE-124-D02

1 of 3

D

Start #	Date	Time	Length	Loaded	Reason
504	11-3-87	8.5	:55	Y	SE-124-207
505	12-3-87	8.5	1:03	Y	SO-024-001
506	12-3-87	8.79	:06	Y	MAINT. CHECK ON OIL LEAK
507	12-3-87	8.8	1:12	Y	SO-024-001
508	12-30-87	7.88	1:11	Y	SO-024-001D
509	1-12-88	8.68	3:25	Y	SO-024-001D
510	2-10-88	8.45	1:13	Y	SO-024-001D
511	3-11-88	8.9	1:14	Y	SO-024-001
512	3-14-88	8.6	26:29	Y	SE-024-D05
513	4-8-88	9.02	1:12	Y	SO-024-001D
514	5-5-88	8.5	:22	Y	SE-224-002
515	5-5-88	8.53	1:00	Y	SO-224-207
516	5-8-88	8.54	1:10	Y	SO-024-001
517	6-26-88	8.84	1:05	Y	POST MAINT. RUN & SO-024-001D
518	7-1-88	8.66	1:11	Y	SO-024-001
519	7-28-88	8.42	1:29	Y	SO-024-001D
520	7-28-88	8.00	:06	Y	SE-024-D04 & SE-124-D03 LOCA START TEST.
521	8-28-88	9.09	1:18	Y	SO-024-001
522	9-26-88	8.85	1:18	Y	SO-024-001D
[illegible]3	10-15-88	8.8	1:24	Y	SO-024-001D
524	11-14-88	8.61	1:04	Y	LOADED TO 2D BUS
525	12-14-88	8.6	1:29	Y	SO-024-001D
526	1-4-89	8.53	1:10	Y	SO-024-001D
527	1-4-89	8.26	1:49	Y	SO-024-001D
528	1-12-89	7.96	4:38	Y	SO-024-001D
529	1-29-89	—	—	—	FAILED TO START
530	1-29-89	11.9	:22		POST MAINT. START
531	2-5-89	8.85	1:55	Y	POST MAINT., & SO-024-001
532	3-11-89	8.4	1:16	Y	SO-024-013 & SO-024-001D
533	3-22-89	8.47	:12	N	QUICK START PER SO-024-013 & SO-024-001D
534	4-10-89	8.38	1:13	Y	SO-024-001D
535	5-7-89	8.64	1:18	Y	SO-024-001D
536	5-21-89	8.6	2:06	Y	SO-024-001D
537	5-21-89	8.6	:43	Y	SE-124-D02
538	5-22-89	8.1	2:03	Y	SE-124-207
539	5-22-89	8.2	2:41	Y	SO-024-001
[illegible]0	5-22-89	8.7	:50	Y	SE-124-D02
541	5-27-89	8.59	:17	N	SO-024-001 VERIFY START TIME
542	6-22-89	8.9	1:19	Y	SO-024-001D

Start #	Date	Time	Length	Loaded	Reason
543	7-6-89	8.9	1:20	Y	SO-024-001D
544	8-14-89	9.85	1:26	Y	SO-024-001D
545	9-2-89	NA	:26	NA	TP-024-083
546	9-2-89	NA	:12	NA	TP-024-083
547	9-2-89	7.1	1:51	Y	SO-024-001D
548	9-16-89	8.47	1:24	Y	SO-024-001D
549	9-26-84	8.6	:30	Y	SE-024-005

10-14-89

THE 'E' DG HAS RUN 39 HRS 28 MIN SINCE ITS INSPECTION ON 8-10-88.

John P. Schroeder

E

Start #	Date	Time	Length	Loaded	Reason
1	5-30-87	7.5	1:22	Y	SO-024-014
2	6-7-87	7.56	:16	Y	SO-024-001
3	6-7-87	7.05	1:23	Y	SO-024-001
4	6-8-87	7.36	1:15	Y	SO-024-001E
5	6-28-87	6.4	1:19	Y	SO-024-001E
6	6-30-87	NA	<:01	N	SO-024-001E
7	6-30-87	6.27	1:11	Y	SO-024-001E
8	7-8-87	7.59	1:20	Y	SO-024-001
9	7-31-87	7.72	1:20	Y	SO-024-001E
10	8-3-87	8.2	1:15	Y	SO-024-001E
11	8-30-87	7.50	1:19	Y	SO-024-001E
12	9-29-87	7.28	1:23	Y	SO-024-014
13	10-31-87	7.21	:30	Y	SO-024-001
14	10-31-87	6.24	1:33	Y	SO-024-001
15	10-31-87	7.0	:15	N	SE-024-E04
16	10-31-87	6.96	2:01	Y	SO-024-001
17	10-31-87	6.8	:25	Y	SE-125-A02
18	10-31-87	6.59	1:15	Y	SE-124-107
19	11-16-87	6.69	1:15	Y	SO-024-001
20	12-14-87	8.8	1:14	Y	SO-024-001E
21	1-4-88	7.51	:11	Y	SO-024-001
22	1-4-88	7.67	1:08	Y	SO-024-001
23	1-5-88	7.2	1:22	Y	SO-024-001
24	1-14-88	NA	<:01	N	NA
25	1-14-88	7.3	:05	Y	TROUBLESHOOTING
26	1-14-88	7.4	1:13	Y	SO-024-001
27	2-11-88	8.7	:20	Y	SO-024-014
28	2-11-88	7.0	1:12	Y	SO-024-014
29	2-11-88	7.38	1:43	Y	SO-024-014
30	3-19-88	7.55	1:26	Y	SO-024-001
31	3-26-88	6.67	:38	Y	SO-024-001
32	3-26-88	6.62	2:24	Y	SO-024-001
33	4-17-88	7.11	1:14	Y	SO-024-001E
34	4-29-88	7.6	:21	Y	SE-224-C03
35	4-29-88	6.5	:47	Y	SE-224-C02
36	4-29-88	N/A	:10	N	SE-204-107
37	5-3-88	8.37	1:43	Y	SE-224-107
38	5-17-88	8.6	1:23	Y	SO-024-014
39	5-31-88	7.66	1:15	Y	SO-024-001

Start #	Date	Time	Length	Loaded	Reason
40	6-26-88	7.7	1:18	Y	SO-024-001E
41	7-25-88	8.6	4:10	Y	SO-024-014
42	8-11-88	7.8	<:01	N	TROUBLESHOOTING
43	8-11-88	7.8	1:20	Y	SO-024-014
44	8-14-88	7.86	1:23	Y	SO-024-001
45	9-9-88	8.24	1:20	Y	SO-024-014
46	10-3-88	7.65	1:19	Y	SO-024-001
47	10-17-88	6.[illegible]	1:16	Y	SO-024-001
48	10-24-88	7.9	1:45	Y	SO-024-001
49	11-7-88	7.65	1:15	Y	SO-024-001
50	12-9-88	8.5	:25	N/A	SO-024-014
51	12-9-88	9.1	1:32	Y	SO-024-014
52	1-6-89	7.45	1:22	Y	SO-024-014
53	1-12-89	7.5	1:14	Y	SO-024-001
54	1-29-89	7.53	1:26	Y	SO-024-001
55	2-6-89	7.6	1:18	Y	SO-024-001E
56	3-12-89	7.83	1:16	Y	SO-024-001
57	5-11-89	6.0	1:20	Y	SO-024-014
58	5-20-89	7.3	:08	N/A	SE-024-E04
59	5-20-89	7.1	2:15	Y	SO-024-001E
60	5-20-89	7.2	:57	Y	SE-124-B02
61	5-22-89	7.0	1:38	Y	SE-124-207
6_	6-7-89	6.57	1:15	Y	SO-024-001E
63	7-3-89	7.82	1:37	Y	SO-024-001
64	7-17-89	8.2	1:25	Y	SO-024-001E
65	7-31-89	6.2	1:28	Y	SO-024-001E
66	8-1-89	6.58	1:14	Y	SO-024-001E
67	8-2-89	6.2	1:23	Y	SUBSTITUTE FOR 'B' DG
68	8-14-89	9.88	2:35	Y	SO-024-014
69	8-26-89	7.29	:05	Y	SO-024-001
70	8-26-89	7.63	:01	N	TROUBLESHOOT
71	8-27-89	7.6	1:15	Y	SO-024-001
72	9-18-89	7.98	:23	N/A	TP-024-084
73	9-18-89	N/A	:05	N/A	TP-024-084
74	9-18-89	8.08	1:16	Y	SO-024-001
75	9-23-89	7.56	1:15	Y	SO-024-001
76	10-8-89	7.6	1:15	Y	SO-024-001

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ENGINE AND VEHICLE RESEARCH DIVISION
TELECOPIER 512/522-2019

5 December 1989

Mr. Frank Czysz
Senior Project Engineer
Pennsylvania Power & Light Company
Two North Ninth Street
Allentown, PA 18101

Subject: **SwRI Preproposal No. EVR-649, "Piston Temperature Mapping and FEA in a KSV-16 Engine"**

Dear Mr. Czysz:

SwRI is pleased to respond to your request for cost and time estimates to measure piston temperatures and perform FEA in your Cooper-Bessemer KSV-16 engine. We are aware of the cold start-up piston seizure problem and feel that we can help solve it as outlined below.

BACKGROUND

Pennsylvania Power and Light Company has experienced several engine failures caused by piston and liner scuffing. Initial investigations indicate that thermal gradients during start up may be the root of the problem. It is desirable to measure piston and liner temperatures in order to perform thermal stress analysis of the piston and liner assembly as well as determine if piston to cylinder wall clearance is large enough to accommodate thermal expansion of the piston during a rapid start up.

Piston temperature profiles are necessary to use computer models and FEA programs for piston design analysis. The measurement of piston temperature in a reciprocating engine has historically been a very time consuming and expensive process. Several conditions exist in an engine that measurement equipment must be protected against. Acceleration forces at TDC in an engine the size of the KSV-16 is near 100 G's at 600 rpm. Operating temperatures inside the crankcase can range over 200°F. To allow complete mapping of piston temperature, several measuring locations are required in the piston and data must be obtained under transient engine operating conditions. The most widely used current measurement technique is to use long sensor leads from the piston along a bar linkage arrangement to the engine block. This method is very difficult to install and maintain. Another type of equipment uses two sets of coils, one of which is attached to the piston and the other to the crankcase. This type requires careful alignment of the coils and is subject to alignment-related failures.

SwRI has developed a telemetry-based system that withstands the harsh environments mentioned above. The device is attached to the underside of a piston and temperature data is transmitted to a receiving antenna in the engine oil pan. The key element of this device is a tiny power generator which utilizes the reciprocating motion of the piston to generate electricity thus allowing the transmitter to be self-powered.

Thick-film hybrid circuit construction techniques have been used to keep the package size as small as possible. The weight of the device, for an engine of this size, will be less than 100 grams. We propose to use this device to obtain the necessary data to allow piston thermal and stress design analysis for the subject engine.

TEST PLAN

The test plan is divided into two separate tasks and cost and time estimates are provided for each.

Task A. Piston Temperature Measurement

One piston will be instrumented with sensors placed in seven locations. A receiving antenna will be installed in the engine oil sump and connected to a receiver and PC-based data logging system. A custom power generator will be constructed to function at 600 rpm in the KSV-16 engine. Sensors will be installed in locations needed to use our ANSYS FEA program.

The engine will be mapped during a cold start-up sequence. Each temperature sensor is scanned once every 16 seconds. This will allow about 6 complete scans during a 90 second start-up. Data will continue to be monitored until the engine has stabilized at operating temperature.

Task B. Finite Element Analysis

Finite Element Analysis (FEA) is an approach to solving real engineering problems which has in the last decade become an extremely powerful tool with the advances in computers. In an FEA the geometry of a structure, part, or assembly of parts are mathematically modelled on a computer. This geometry which may be very complex is cut up into little "finite elements" which may be in the form of lines, rectangles, triangles, bricks, or tetrahedrons. The properties of these elements can be very accurately predicted with mathematical equations. The resulting finite element model, if properly constructed, can very accurately predict what is occurring in an actual part. FEA can be used to predict, stresses, strains, deflections, heat flow, and many other characteristics of a part.

In the case of this subject engine, FEA will be used to try to reproduce mathematically what takes place in the parts of concern; namely the piston, liner, and possibly the piston pin and pin end caps; during the start-up of the engine. An FEA model of a piston and liner will be constructed. Initially a two dimensional axisymmetric model will be constructed. This will provide some basic insight as to the nature of the heat flow and resulting deformation. A piston, however, is not axisymmetric because of the wrist pin and the internal cooling fins and sometimes the combustion bowl. If

the results of the axisymmetric model are not conclusive, a three dimensional solid model of the piston and liner will be constructed. It will most likely be a quarter section model bounded by symmetric planes.

A transient thermal analysis will first be performed in which the piston and liner starting at normal temperatures, will be subjected to a high temperature in the area of the combustion chamber that is representative of the average actual temperatures during the start-up process. This analysis will predict the temperatures in the piston and liner during this start-up process. From this temperature data, the model will predict the deformation and stresses of the piston and liner due to the temperatures and loads. The loads that are applied can be any combination of combustion pressure, side loads from the connecting rod, and inertial loads. From these results, it should become obvious where the problem areas are as far as possible interferences, when they occur in the start-up process, and under what loading conditions they occur.

As with any computer analysis, the results are only as good as the input data. In this case, it is very important, for the accuracy of the model to know what the boundary conditions are, in particular, the temperatures at the surfaces of the piston and liner, and the physical loads on the parts. The temperatures in the combustion chamber can be predicted by a computer model known as TRANSENG. If actual piston temperature data is not measured, this model will be used. It is preferable that actual temperature data be used. The physical loads will be estimated based upon the engine, geometry and operating conditions.

Upon the identification of the problem area, corrective actions can be determined and tested with the FEA model. Several different corrections can be tested easily and quickly on the computer without risking potentially catastrophic and expensive failures on the actual engines. Once an acceptable correction has been tested and identified, then they could be implemented on the engines.

The FEA software package that will be used is ANSYS by Swanson Analysis Systems, Inc., Houston, Pa. This is a leading company in this field and has a longstanding reputation of providing reliable software. This software meets the applicable NRC quality assurance specifications.

COST AND SCHEDULE

The estimated cost to set-up the instrumentation on the subject engine, instrument one piston, and acquire piston temperature data will be $135,000. This task will take twelve weeks to complete. The majority of this time is to design and construct the power generator for this size engine. Task B will also require about 12 weeks to complete and can be conducted in conjunction with task A using actual piston temperature data or independently using the TRANSENG modeling data. The estimated cost of this task will be $50,000.

The costs estimated for this project are estimates for planning purposes only. Upon request, we will send you a formal SwRI proposal which will describe the proposed work, the proposal contractual terms, and the project cost. For your information, we are attaching to this letter a sample copy of the contract we will send you in the formal proposal.

MANAGEMENT AND EXPERIENCE

Through the years of contract research, SwRI has developed a project manager system for conducting each research project. The project manager provides one point of contact for the sponsor. This project manager is responsible for the technical, schedule, and cost facets of each project. The system ensures timely response to the sponsor's needs, and simplified communications. The project manager for this program will be selected upon proposal acceptance.

Professional record sheets for key personnel available to assist in the accomplishment of this project are included in the attachments. Organization charts for the Institute and for the Engine and Vehicle Research Division and related reference material are also included.

If you have any questions, please call me at (512) 522-3064.

Sincerely,

Robert W. Burrahm
Senior Research Scientist
Engine Evaluations
Department of Engine Research

/jg

Approved:

Shannon Vinyard
Director
Department of Engine Research

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Report by J. R. THOMAS

Approved by S. G. DEXTER

Date 15TH DECEMBER, 1989

INVESTIGATION FOR PENNSYLVANIA POWER & LIGHT CO. OF CRANKCASE EXPLOSIONS IN COOPER-BESSEMER KSV-16 STAND-BY ENGINES AT SUSQUEHANNA STEAM ELECTRIC STATION BERWICK, PENNSYLVANIA

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5th December, 1989

SUMMARY

Ricardo visited the Susquehanna Steam Electric Station at the request of PP&L, to inspect damaged components following two crankcase explosions and, if possible, to indicate the cause(s) of failure.

As a result of the evidence seen and information supplied later by PP&L a clear cut cause of failure has not been identified. However, Ricardo consider it likely that distress of piston pins in their bushings has led to the failure, the increase in piston pin friction having initiated piston skirt seizures. Poor combustion or overfuelling are not suspected, nor is piston ring scuffing.

The report describes and discusses the evidence considered by Ricardo, indicates possible causes of distress at piston pin bearings and then presents conclusions and recommendations for further investigations.

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INVESTIGATION FOR PENNSYLVANIA POWER & LIGHT CO. OF CRANKCASE EXPLOSIONS IN COOPER-BESSEMER KSV-16 STAND-BY ENGINES AT SUSQUEHANNA STEAM ELECTRIC STATION BERWICK, PENNSYLVANIA

1. INTRODUCTION

At the Susquehanna Steam Electric Station there are 4 stand-by diesel generating sets powered by Cooper-Bessemer KSV-16 engines. A fifth set with a KSV-20 engine has been recently installed in a separate power house.

Crankcase explosions have recently occurred in two of the KSV-16 engines during routine testing. Three other such explosions had occurred in earlier years, one during commissioning during 1983.

Following the two recent failures, Ricardo were contacted by PP&L and asked to inspect the installation, examine failed parts and report on their findings as quickly as possible. A visit was paid to the power station and to PP&L's Laboratories at Hazleton, Pa, on 7th November 1989. Three failed pistons and two failed liners were seen and the installation of one of the stand-by sets was examined.

This report presents Ricardo's findings based on the information obtained during that visit, plus additional facts supplied later by PP&L.

2. THE STAND-BY ENGINES AND OPERATION

The engines are Cooper-Bessemer KSV-16 diesel engines, operated on No. 2 Diesel Fuel. The following are the main details:

Bore 13.5 ins

Stroke 16.5 ins 16 cylinders at 45° Vee angle

Rated Speed 600 rev/min Mean piston speed 1650 ft/min (8.38 m/s)

Continuous Rating 4000kW (electrical)

Maximum Rating (short term 2000 h max total) 4720 kW (electrical)

Bmep (continuous) 213 lbf/in^2 (maximum) 250 lbf/in^2

Constant Pressure Turbocharging

Intercooler water temp (out) 112°F (44.5°C)

Normal coolant temp (out) 170°F (77°C)

Normal oil temp (from engine)	165°F (74°C)	
Normal oil pressure	50 lbf/in^2	
Oil pump capacity	530 gall/min	
Oil volume in system	1400 gall.	Lube Oil: Gulf SD 40

Nominal peak firing pressure at continuous rated load 1600 lbf/in^2 (110 bar)

The engine has an underderslung crankshaft; precision-type steel-backed shell bearings; master and slave (articulated) connecting rods, palm ended with bolted-on piston pins; cast iron one piece pistons with bronze piston pin bushes and oil-fed crown gallery with cocktail shaker cooling. The piston skirts are tin plated from the second oil scraper ring groove downwards; porous chromium plated iron liners; 4 taper faced plain iron compression rings and 3 iron, hooked, spring-backed oil scraper rings (one below the piston pin); 4 valve cylinder heads; high mounted individual fuel pumps with short high pressure pipes and multi-hole injector nozzles; nozzle opening pressure 3500 lbf/in^2 (240 bar).

Under stand-by conditions the engines are kept primed and warm with continuous lube oil circulation at 50 lbf/in^2 and 120-130°F. Coolant is continuously circulated at 120-130°F.

A large number of safety warning and shutdown systems (18) are fitted, including temperature sensors at every crankshaft bearing. All shutdowns are disabled during emergency conditions except overspeed, low engine oil pressure and generator differential.

The engines are kept in stand-by mode ready for automatic emergency start and load application if necessary. They are capable of running at rated speed within 10 sec of start initiation, and of carrying 4000 kW within 90 sec.

At monthly intervals each engine is given a test start and load application; 4000 kW must be obtained within 90 seconds. The engines are then run at this load for 1 hour, prior to unloading and shutdown. The shutdown procedure involves a 5 minute cooldown period at no load. During the monthly tests, the load is ramped onto the engines manually as the engines run in parallel with the main power station output; 4000 kW is generally achieved before 90 seconds.

Every 18 months the engines receive an inspection which includes a 22 hour run at 4000 kW plus 2 hours at 4700 kW. Internal inspection does not include the removal of any running components. Until recently, oil changes were carried out at these 18 month inspections, but now a more comprehensive oil sampling and analysis programme is carried out and oil changes will be made only when such analysis shows it to be necessary.

Cylinder head replacement is scheduled at 5 yearly intervals.

Main engine operating pressures and temperatures are displayed on the generator set control panel adjacent to each engine. Monitoring is carried out manually at hourly or 4-hourly intervals; individual cylinder exhaust temperatures are logged during the 18 month trial but not during monthly runs. Max and min accepted readings are pre-printed on the log sheets. No formal trend monitoring is undertaken, but logs are actually carefully scrutinised by experienced observers.

3. HISTORY OF FAILURES

Five crankcase explosions have occurred as follows:

1. 29 Nov 1981 D-unit Bearing failure in lube oil pump during initial start-up, causing a hot spot.

2. 14 Jan 1984 D-unit Severe 2L piston skirt pick-up and overheat. Put down to inadequate lubrication because of fuel oil dilution.

3. 18 Jan 1986 B-unit Severe 5L piston pick-up and overheat. One piston bolt found loose. Put down to loss of cooling oil via loose bolt.

4. 16 Sep 1989 B-unit Severe 7L piston skirt pick-up and overheat. Mainly non-thrust side. Cause unknown.

5. 7 Oct 1989 C-unit Severe 5R piston skirt pick-up and overheat. Initially attributed to authorised "repair" of casting defect on skirt surface.

The crankcase explosions in cases 2 to 5 occurred at various stages during test runs; not particularly early in the run-up and load application sequence.

Piston breakages did not occur and the distressed pistons continued to reciprocate intact until the engines were shut down manually following the explosions.

The report by PP&L on event 2 was shown to Ricardo. It showed the following:

Considerable metal transfer from skirt to liner surface and build-up of smeared and dragged oxides had occurred. Deposits inside the skirts showed that very high temperatures had occurred. Metallurgical examinations of the piston skirt from event No. 2 indicated that local skirt temperatures had reached 1350°F (730°C). The piston pin had also lost hardness in places, requiring a similar temperature. The lead phase of the lead bronze pin bushing had melted, indicating temperatures above about 450°F (250°C).

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PP&L reported that in several cases of piston failure, heavy contact and material transfer had occurred between the end caps, which close the bore of the piston pin bushing in the piston body, and the liner. These cast iron end caps are pressed into the piston body and are normally located comfortably below the piston running surface.

Following all failures, careful crankcase-side inspections of all other cylinders were made to ensure no other cylinders or components were damaged. Cooper-Bessemer Reciprocating were closely involved in ensuring the engines were returned to service correctly and safely.

A number of KSV engines are performing similar stand-by duties at other power stations as well as a range of other industrial and marine duties. PP&L indicated that there has been no history of similar failures elsewhere.

4. DISCUSSION OF OBSERVATIONS MADE 7TH NOV 1989 (AND LATER INFORMATION)

4.1 Practices

Ricardo first enquired about the installation and modes of operation of the diesel engine sets at the Susquehanna Steam Electric Station, together with details of maintenance procedures, data logging, record keeping etc. The history and events leading up to the crankcase explosions were examined, together with the oil analysis information that is available.

PP&L showed that in October 1985 it was found that all 4 units were running with about 1750 lbf/in^2 Pmax at 4000 kW output. (This was found in engine analyser tests carried out by Cooper, which form part of the 18 month engine inspection). The injection timing was retarded on all engines by 4-6° crank. In one case this required retiming of the camshaft by 2° but shimming of the pumps was carried out on the other engines.

Engine oil analysis may not have given a true estimate of the wear metals in the oil until late in 1989. This is because samples were being taken after the oil filters. Since June 1989, sampling from before the filters has been carried out and spectrographic analysis is being made by 'Gulf-Check'. Ricardo agreed that the new analyses should be suitable to guide maintenance, and advised PP&L to monitor trends on the major wear metals and additive elements reported rather than wait for Gulf to announce a problem or the need for an oil change. Gulf should be asked to explain the factors which indicate the alkalinity or TBN of the oil, so that this important quality can be approximately checked at a glance. Ricardo suggested that individual cylinder exhaust temperatures should be checked during monthly test runs. Changes in these temperatures or their scatter pattern give useful indications of individual cylinder condition or deteriorating performance.

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The maintenance and record keeping by PP&L are largely governed by the regulations and formalities of the US Nuclear Regulatory Commission. Ricardo gained the impression that these activities are carefully and thoughtfully carried out at the Susquehanna SES. The PP&L engineers are, however, heavily dependent upon Cooper-Bessemer Reciprocating for the evaluation of engine health. The identification and advice on any rectifications or adjustments needed are generally from Cooper. This is a satisfactory and safe situation so long as PP&L staff continue to monitor their records and understand the implications of work proposed, in the light of earlier work done.

4.3 <u>Description of Components Seen</u>

Ricardo were able to examine the pistons, liners and some of the piston rings from events 4 and 5, i.e. the 7L piston from B-unit and the 5R piston from C-unit.

In the descriptions which follow, the identification of thrust and non-thrust sides is that given by PP&L. Ricardo were unable to verify these from the dismantled components.

The most significant features of the two pistons were their similarity to each other and to the description of the piston in the PP&L report of event 2. This is said despite the fact that the piston from event 4 was much less severely damaged than those from events 2 and 5, particularly on the thrust side and piston pin/bearing surfaces. The character and likely history of the damage was considered similar. The piston skirts were heavily damaged over most of their surfaces, deeply torn and scuffed so that the lower scraper rings were mostly trapped by smeared material. Very little tin plating was still visible as the skirts were blackened due to oxidation, with large areas of smeared and torn black material re-adhered to the skirt surfaces. The lower edges of the skirts were the most recently damaged with bright iron visible. The insides of the skirts were also mainly blackened, with the brightness of the tin plating visible only on the undersides of the piston pin bosses. The blackening inside the skirts was at least partially a deposit of carbonised lub oil.

There was a strong contrast between the skirts and the crown portions of both pistons. Whilst heavy scoring and smearing damage was visible on the inter-ring lands and on the faces of the compression rings, this was clearly secondary damage and there was no evidence of piston crown overheating or initial ring distress. There was moderate carbon build-up on the top lands, with light carbon and staining on the intermediate lands also. Top ring carbon was moderate for the hours run, but not excessive. The compression rings were free, except where trapped by smeared material from the skirt. There was no evidence on their side faces of pinching in the grooves, although there was some wear of the top ring lower faces. The rubbing faces of the compression rings were badly scored due to riding over debris attached to the liners, but they did not show witness to excessive temperatures. The oil cooling galleries in the piston crowns appeared clean, again

showing no evidence of excessive crown temperatures.

Contact between piston-pin end caps and the liners had occurred, with material transfer and the development of hot spots on the caps. The end caps were not in situ during this inspection but it was reported that some were found to be dislodged when the engine was stopped.

The Piston pin and bush from event 5 were badly distressed due to overheating, with both oxide and carbon build-up between pins and bushes. The 7L pin and bush from event 4 were only very slightly marked but there was heat staining over more than half of the surfaces.

The running surfaces of the liners were badly damaged, with areas of adhered black material covering the chromium plated surface. Again, both liners showed a basically similar condition. Between the patches of adhered material, the porous chromium surface appeared intact and in good condition. There was little sign of the chromium becoming detached from the main body of the liner. The patches of material build-up covered the lower portion of the liners, from the second scraper ring position at tdc, downwards. Above this position the liners were relatively undamaged and the chromium surfaces showed no evidence of primary distress. The build-up of adhered material on the lower portions of the liner surfaces formed distinct patterns with bands of burnt-on material less than a half-inch wide in a "tartan-like" formation. This pattern appeared to have formed relatively early in the failure and in many cases had been obliterated by material later dragged and smeared over the bands.

There were differences observed amongst the basic similarities between power cylinder components following the two failures:

The skirt and liner distress and the consequent secondary damage, had been more severe on the 5R piston from C-unit (event 5). In this case the damage around the full 360° of piston and liner was more or less uniform, but with slightly more dragging on the non-thrust side, particularly on the compression ring interlands. In this case also, the top rings were bedded over their full width. The piston pin had seized so firmly into its bush that the bush was turning in the piston body. The "tartan" pattern was evident almost all around the liner.

Part of the lower skirt of this piston from event 5 had been cut out for metallurgical examination of a casting defect. There were 2 "blowholes" which did not pass right through the skirt wall and examination later showed these had been initially filled with solder, prior to tin-plating of the skirt. Coopers have confirmed that this particular piston casting had been repaired in this way, according to an internally approved procedure.

The 7L piston and liner from the B-unit (event 4) were not so extensively damaged. The main piston skirt and land damage was on the non-thrust side and the "tartan" marking in the liner was more or less confined to the thrust and non-thrust positions. The subsequent spread of damage had nevertheless extended around much of the liner. The

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bedding (or secondary wear) of the compression rings was not so extensive. Piston pin and bush damage was also lighter, but there had nevertheless been overheating, mainly towards one end, leading to heat staining and some carbon/oxide build-up. In this case there were possible indications of slight fretting between the pin and the palm end of the rod.

It is apparent that during event 4 the skirt distress and overheating produced an explosion slightly earlier in the course of the spread of the seizure than occurred in event 5.

One other piston was seen by Ricardo. During inspection of the C-unit following event 5, the lower portion of another liner was seen from within the crankcase to have piston material adhering to the liner surface on the non-thrust side. This power cylinder had then been removed. The liner had been found to have tin smeared onto it, but after removing this with Scotchbrite the liner was adjudged to be suitable for further service. A new piston was fitted. The piston removed was seen to have extensive scuffing on the non-thrust side only. This scuffing had removed most of the tin plating from between the lower 2 oil scraper rings (above and below the piston pin) leaving a bright torn iron surface. Below the lower ring, the skirt was slightly scratched only. The thrust side of the skirt was very smoothly bedded with light scratches. All rings were free and showed no evidence of primary distress. Smeared material had scored and marked the ring faces, however, and the rails of the sharply hooked scraper rings were jaggedly torn. Carbon deposits around the piston crown were moderate with some staining down to the third inter-land. The oil cooling gallery was clean and free from deposits. The piston pin boss showed no heat staining on the lower sides. The pin and bush were not seen.

PP&L had carried out some careful diagnostic metallurgical and metallographical investigations on various portions of the failed piston components. These portions were those subject to severe seizure and metal transference relatively late in the course of failure, and material identification as well as surface structures were clearly seen. Together with the heat staining and carbon deposit inside the skirts of the failed pistons, these examinations confirmed that piston skirt temperatures had reached high enough levels to initiate crankcase explosions. They did not, however, indicate events which might have been taking place during the early phases of the failures. Ricardo suggested other regions which should be examined to try to discover whether any piston oil scraper ring material (iron) was being transferred onto the chromium liner surface prior to material from the skirt surface (tin). Similarly, it was suggested that the smeared material on the piston pin end caps should be examined to see if the initial contact was with the chromium liner surface, or whether in fact it was with tin already transferred onto the liner from just above or below the piston pin bore.

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4.3 <u>Discussion of Findings</u>

After the visit to the Susquehanna SES and the Hazleton Laboratories, Ricardo considered their findings. The following two related statements (4.3.1 and 4.3.2) are regarded as basic. The comments made in 4.3.3 to 4.3.15 were also derived from information gathered during the visit but a more complete picture emerges in section 4.4 from information later supplied by PP&L.

4.3.1 The PP&L conclusions that piston skirt distress, scuffing and subsequent metal transfer and high friction had led to high skirt temperatures, subsequently causing the crankcase explosions of events 2, 4 and 5, were quite justified. (Ricardo have seen no evidence from event 3).

4.3.2 The skirt distress originates in the skirt region itself. It does not result from combustion-induced distress in the top land/top rings zone of the pistons

4.3.3 The chromium plated surface of the cylinder liners appears good and well-suited to the duty.

Note: Chromium is not an easy surface for marginally-lubricated sliding surfaces at high temperatures. A porous (fissured) surface is therefore essential to retain lubricant to prevent ring scuffing. Cooper appear to achieve an excellent and consistent surface, to the naked eye. Inadequacies in chromium surface condition would be expected to be evident first in scuffing of the top ring. There is no evidence that this has occurred.

4.3.4 The ring pack appears extremely stringent in downward-scraping oil control. The 4 compression rings are all taper-faced and then there are a further 3 spring-backed oil scraper rings having very narrow and sharply downward-hooked rails on each ring.

For a chromium plated liner surface, and a duty where load application is quite rapid after initial start-up, Ricardo would not expect such a stringent ring pack.

PP&L were asked to obtain piston and liner dimensions for examination of basic clearances and roundness by Ricardo in the light of these observations on oil control.

4.3.5 The thickness of the tin plating on the piston skirt is not known by Ricardo. Provided it is not too thick, the tin plating gives good compatibility with the liner and helps initial embedding of debris or particles encountered during running-in. PP&L were asked to find the thickness used on the KSV engine.

4.3.6 The skirt surface produced is extremely smooth.

4.3.7 The condition of the third piston, seen in the early stages of failure, suggests that skirt pick-up is initiated near the top of the skirt on the non-thrust side.

4.3.8 The design arrangements for oil escape from the scraper rings are not very generous. There are no drain holes from 2 of the 3 grooves and little relief below the bottom groove.

4.3.9 The condition of the piston pins and bearings after failure was consistent with the general temperature to which the surrounding skirt regions had been heated by the seizures. The damage following event 5 was much worse than after event 4. It was not possible to detect any clear pattern and it appeared that most of the distress and damage was consequent upon the main skirt failure.

4.3.10 The contact between piston pin bore end caps and the cylinder liners is reported to have occurred even where piston skirt failure has not taken place. Loosening of the end caps must therefore occur from time to time, without excessive heat input to the surrounding zone. The oil drain holes are adequate to prevent any pressure build-up between the piston pin and the end cap. Contact between liner and end cap is undesirable because the correct alignment of a "slipped" end cap could not be guaranteed, but such contact would not necessarily lead to metal plucking and high temperatures because the caps are made of cast iron. The major hot spots and welded material on the end caps of the badly seized pistons are most likely to be due to contact with piston skirt material already welded onto the liner during the course of the failure. Metallurgical examinations would confirm this if the material welded first to the iron surface is mainly tin.

Ricardo believe that removal of the end caps would not be detrimental to engine oil control or durability.

4.3.11 The unusual "tartan" pattern of heated and oxidised material adhering to the liners probably occurs due to piston ring vibrations caused after the failure is well under way, by rings passing over spots of adhered tin on the liner surface, and/or when rings are trapped at some point by smeared material.

4.3.12 Any seizures are the result of a breakdown in lubrication for some reason. Therefore the lubricating oil quality must be examined to ensure it has not contributed.

Apart from examination of oil specification and cleanliness, PP&L should check what other KSV engines are running on the Gulf SD 40 oil used at Susquehanna SES, and indeed what other similar medium speed diesel engine experience there is with this oil. Ricardo recommend discussions with users as well as with the oil manufacturer.

4.3.13 PP&L are satisfied that the fuel quality is correct, and no fuel-related problems have been identified in these failures.

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4.3.14 The skirt repair carried out by soldering-up blowholes on the piston which seized in event 5 is not thought to have contributed to the failure. Ricardo would not criticise this repair procedure, provided that the quality control safeguards are seen to be adequate. (Location of fault, size and frequency of fault etc., etc.).

4.3.15 The mode of operation of the stand-by engines is not considered too severe and hence a main cause of the crankcase explosions. The circulation of warm oil and coolant when the engines are in stand-by mode ensures that they are well prepared for start-up, and the quick starting and rapid application of load cannot be considered too arduous for the design of the engine.

The operating procedures and maintenance practices at Susquehanna SES appeared to be well and thoughtfully carried out.

The installation of the engines and auxiliary systems gave no cause for concern.

4.4 Information Made Available After the Visit

PP&L have provided other information to Ricardo since the visit on 7th November. This information is discussed below:

4.4.1 Piston to Liner Clearances

Dimensions and tolerances for the piston and cylinder liner diameters have been examined. The piston skirt is cylindrical and the nominal cold clearance is 0.006 to 0.0085 ins radially (0.044% to 0.063% of the cylinder bore). Ricardo would consider these clearances normal to generous for an iron piston in an iron liner, and well suited to a duty where full load must be applied quickly after initial start-up.

4.4.2 Thickness of Tin Plating

The tin plating thickness on the finished skirt is 0.001 to 0.0035 ins. At its maximum, this thickness may be considered quite large, in view of the skirt spalling that has taken place. The tin plating does, however, make the skirt highly tolerant to dirt particles and brief moments of marginal lubrication, such as could occur on the first revolution or so after start-up. The pick-up of this tin coating, which is occurring at Susquehanna, and its transfer to the liner surface appears to be initiated by some exceptionally high loading of the skirt onto the liner, perhaps also with marginal lubrication, over many cycles. Once a small region of the tin coating adheres to the liner, piston rings would be dragged over it and the distress would spread rapidly, with high temperatures.

4.4.3 Piston Pin and Bush Distress

PP&L reported 3 further cases of tin-smearing seen on liners during an underside inspection. The pistons were removed from these cylinders

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and in each case local heated spots were seen on the piston pins. There was also a clear relationship between the degree of skirt distress and the amount of piston pin heating (shown by "blueing" of the piston pin surface). The tin removal had occurred on the non-thrust side of the piston skirt, leaving a bright surface of scored and torn iron and tinplate between the lower two oil scraper rings. The only other damage seen was scoring of the piston ring faces as a result of passing over the smeared material on the liner face. This information is regarded by Ricardo as very significant, in explaining the earlier piston failures. High friction at the piston pin bearing clearly results in increased side loading on tilting of the piston within the liner bore. Two-cycle engines present particularly difficult piston pin bearing lubrication problems because the load on the bearing is continuous in one direction throughout the cycle. Where piston pin/bushing distress occurs, the increase in friction usually results in piston skirt distress also, due to very high side thrusts. In four-cycle engines, the load reverses at the piston pin due to inertia forces at non-firing tdc, so lubrication is replenished each cycle and piston pin/bearing problems are rare except during engine development.

The damage to the piston pin and bush surfaces of those pistons which have led to crankcase explosions was so extensively developed that it had not been possible to identify whether it had preceded or followed the piston skirt failure. The new evidence shows that pin distress occurs with skirt distress and that, even where no heat build-up has taken place in the failing skirt, there is heat build-up occurring at the piston pin/bushing interface. There is strong evidence, therefore, that local breakdown of the lubrication is taking place in piston pin bearings which gives rise to the piston skirt seizures which have ultimately caused crankcase explosions. (It is considered unlikely that piston skirt pick-up, which had not been so heavy as to lead to significant local heating of the skirt surface would lead to overload of the piston pin bearing).

To support the suggestion that piston pin/bearing failures are the cause of the problem, it is necessary to identify an initial cause for such failures. Possible causes could be:

a) - inadequate design

b) - failure of lubricant supply or dirty oil

c) - incorrect manufacture (dimensions, roundness, surface finish, material specifications etc.)

d) - overload

These will be briefly discussed next.

4.4.3a Design

The use of a palm-ended connecting rod bolted to the piston pin allows the piston firing load to be carried by the whole length of the piston pin. Ricardo have not analysed the design and loadings of the KSV piston pin and bushing. The use of a piston pin which is 38% of the bore diameter and mainly solid, with the above bushing feature, indicates a design which is entirely adequate for the firing pressures of less than 2000 lbf/in^2 occurring in the KSV engine. This observation can be confirmed by detailed design analysis which Ricardo are able to carry out if required. The pin hardness specification and bushing material are believed satisfactory, but can be assessed.

It is possible that particular features of the piston design could give rise to local distortions at the piston pin bushing under high cylinder firing pressure loads, leading to loss of bearing clearance and local seizures. This has not generally been seen on these engine, but could occur in certain pistons if the gallery core had shifted in some way, for example. In such cases failure would have been expected earlier in the engine lives, particularly since the Pmax was higher than intended during the first years of operation at Susquehanna. If the design is marginal in terms of local distortions this could be shown by detailed Finite Element Analysis of the piston structure, under realistic engine mechanical and thermal loads. Such analysis requires considerable experience in the correct application of mechanical and thermal loadings, and Ricardo have successfully carried out this type of analysis on many occasions.

4.4.3b Lubrication

The most significant factor here is that no failed pistons have shown any loss of oil cooling in the piston crown. Cooling galleries have been oily and clean. Total failure of the oil supply up the connecting rods is therefore not suspected. Some kind of intermittent and short term interruption of the oil supply would not cause piston crown overheating because oil remains trapped and shaken in the cooling gallery for several cycles, but could lead to local failure of lubrication and cooling at the piston pin/bushing interface if it persisted for several cycles.

Grooving of the crankshaft bearings allows for "continuous" feed of lubricant at engine oil pressure to the connecting rod drillings. During engine running the feed up the rod is not truly continuous because of the very strong inertia forces acting upon the column of oil in the conn rod drilling. Nevertheless, the oil feed normally occurs during part of every engine revolution.

Components seen from events 2, 4 and 5, together with the incipient skirt failures recently reported, give no indications of any blocking of oilways to the piston pin bearings. It is understood that there are no check valves in the oil drillings at the large end of the connecting rod, which have given rise to problems on some engines.

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The passage of oil into and through the piston pin bearings requires correct running clearances and adequate drain paths. Ricardo did not observe that the design is inadequate in these respects but a review of the design is recommended. Blocked drain holes from the spaces at the ends of the pin have not been reported.

Oil cleanliness is not suspect in any cases under investigation at PP&L.

No causes for local interruption of lubrication supply to the piston pin bearings have been discovered at present.

A review of the actual lub oil in use at Susquehanna is advised, as in any case such as this.

4.4.3c Manufacturing Defects

PP&L have been advised to carry out careful dimensional checks on the piston pin features in the cases where incipient failure has been detected. These checks should be extended to include evaluation of materials, hardnesses and surface finish. No such defects as might have led to failure have been visually evident in components seen by Ricardo.

4.4.3d Overload

There are no indications that engine load was excessive just prior to any of the failures or incipient failures. Nor have fuel pumps or injectors shown that the individual cylinders were subject to overloading. Absence of evidence of high piston crown temperatures confirms that overload is not a likely cause of failure. Furthermore, the high Pmax conditions prevailing during the early life of the engines did not lead to a spate of such failures at that time

It is possible that the early period of running at high Pmax could have initiated piston pin bushing local distress which has since progressed slowly leading to intermittent failures. If this were the case, it would be possible now to find pins with local marking or heat staining, but from which no skirt distress had yet developed. Ricardo recommend some random examinations of "good" pistons.

4.4.4 Piston Ring Wear

PP&L reported that on all 3 pistons suffering incipient skirt seizure and on one other "good" piston withdrawn for inspection, the top ring bedding was over the full face width of the ring. The bedded area of the lower rings was progressively narrower from the second ring downwards.

Each of the engines has run for about 1000 hours. This would not normally be expected to lead to sufficient ring wear to produce full face bedding of the top ring. It should be remembered, however, that

each engine has also accomplished over 200 starts with rapid load application following. This could have contributed to higher than normal wear rates. PP&L are advised to monitor ring wear in order to estimate the wear rate and likely replacement period. Meanwhile, checks should be made of all air cleaner panels and intake air ducting to ensure no unfiltered air enters the engine. In the same way, the cleanliness of the starting air system should be confirmed.

Ricardo advise PP&L to discuss the ring wear with Cooper and also with other Nuclear users of the KSV engine. However it is not considered to be a factor directly leading to the crankcase explosions. (High piston side loads should not result in increased ring loading, but abnormal piston tilting could affect piston ring attitude to the liner face).

4.5 **Further Discussions**

Whereas the examinations carried out by Ricardo at Susquehanna SES and the Hazleton Laboratories did not enable a cause of the piston skirt failures to be identified, the information subsequently supplied by PP&L enables Ricardo to identify piston pin/bushing distress as the likely route to failure.

The cause of any such piston pin/bushing distress is not yet clear.

- The detailed inspection by PP&L of the components where incipient failures were discovered should identify any quality, manufacturing or material defects. None have been apparent in investigations so far.

- Cooper should be asked to investigate any relationships which may exist between failed pistons, pins or bushings and their manufacturing batches or dates.

- Ricardo have not been able to observe any defects in the design of the piston pin area of the KSV engine either from components seen or from general arrangement drawings of the engine. It is not believed that the overall design is faulty, but confirmatory checks of the pin stiffness and loadings and of the design clearances and tolerances could be quickly carried out. If further examination of incipient failures indicates that particular local distortions of the piston pin bushing may be occurring, a detailed analysis of the piston structural design would be needed to confirm that the design may be marginal, and to indicate design improvements. Ricardo have considerable experience of such analysis, where the correct mechanical and thermal inputs are essential.

- Reasons for inadequate or intermittent lubrication of the piston pin bushing cannot be identified at present. This cannot be eliminated as a source of the failures however. PP&L should carefully examine the issue of the lubricant itself as indicated in 4.3.11. Ricardo have accepted PP&L's assurances that so far there has been no evidence at all of crankshaft

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bearing problems on their engines. If there are any discontinuities in the oil feel to the piston pins in these engines, it is probable that the crankshaft bearings would also exhibit some evidence, either in the form resulting from loss of oil film thickness or from cavitation. PP&L are advised to check both large end and main bearings adjacent to a failed cylinder if there are any doubts on this matter. Ricardo would be able to comment on any results observed.

5. **CONCLUSIONS**

The following conclusions are drawn by Ricardo on the basis of the evidence seen during the visit to PP&L on 7th November 1989, and of information subsequently supplied by PP&L.

5.1 The crankcase explosions have been initiated by excessively high piston skirt temperatures which have resulted from severe piston skirt seizures, material transfer to the liner surface and very high friction.

5.2 The piston skirt distress has not been consequent upon overheating, overloading or scuffing of the piston crown or of the compression rings and ring belt. (Note: skirt failures often result secondarily from top ring scuffing or top land carbon grabbing. This is not the case for the PP&L failures).

5.3 Incipient skirt seizures detected and examined at PP&L have confirmed the evidence from the badly overheated piston that the skirt distress is initiated on the non-thrust side, probably near to the top of the skirt portion.

5.4 It is known that high friction in the piston/piston pin bearing can give rise to excessive piston side thrust leading to scuffing, metal transfer and seizure of the piston skirt against the cylinder liner. Such high friction at the piston pin bearing can result from loss of the correct conditions of lubrication and/or geometry between the piston pin and bearing bush. High cylinder pressures may contribute.

Overheating (blueing) of the piston pins has been seen in all cases of incipient piston skirt seizure at PP&L. It is considered likely that the origin of the piston failures which have occurred, was piston pin/bush distress leading to high friction in this bearing.

Severe piston pin and bush overheating had occurred in the pistons which had seized and led to explosions. In these cases, the heavy overheating and damage was considered to be secondary, and the initial pin/bush distress condition was not detectable.

5.5 The fact that many pistons operate successfully, both at PP&L and in other installations of the KSV engine at the same rating, suggests that piston pin/bush distress arises where some small variation in dimensions, clearances, surface finishes or lubrication occurs. These

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critical variations, whatever they are, could have been aggravated earlier in the lives of the PP&L engines when they were operated at high max firing pressures. Careful examination of the piston pin/bushing design and tolerancing is needed to discover the key factors. In the absence of any indications from such an examination a detailed structural analysis of the piston and pin/bushing may be necessary.

5.6 Under abnormal conditions of high piston side loading, where spalling of the tin-plating on the piston skirt may occur, the thickness of the tin-coating (.001 to .0035 ins) could lead to excessive material transfer and very rapid spread of the damage. Nevertheless, under normal and quasi-normal operating conditions (e.g. rapid start and load application) the tin plating provides a surface with forgiving features of conformability and embeddability.

5.7 The casting defect and repair found on the 5R piston skirt of the C-unit which failed (Event 5) is not thought to have contributed to this failure.

5.8 The oil scraping features of the piston ring pack are rather stringent. It is expected that the piston would be more tolerant to high skirt thrust forces if changes were made to allow more oil on the liner surface. Such changes need not increase engine oil consumption, provided that adequate oil drain features from the oil control ring grooves are incorporated. The existing piston skirt clearances are generous, and probably apt for the high rate of load application which these engines must accept.

5.9 There is evidence of higher than normal wear rates on the faces of some upper compression rings. The reasons for this require further investigation, but it is not thought that this factor has contributed to the skirt failures which led to crankcase explosions.

5.10 There do not appear to be faults in the installation or the mode of operation of the standby diesels at PP&L's Susquehanna SES which have directly led to the crankcase explosions which have occurred.

6. RECOMMENDATIONS

6.1 Detailed measurements should be made of the piston pin and bore dimensions on pistons which have been found with incipient skirt failure, and on others which show no distress. Other important piston and ring dimensions should also be checked.

6.2 Checks should be made to ascertain any "batch" or "date of manufacture" relationships amongst those pistons which have failed or in which incipient failure is suspected.

6.3 Analyser Records should be examined to see if failed or failing pistons are always those subject to higher than average firing pressures.

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6.4 Design analysis of the piston pin/bushing of the KSV engine should be carried out. 2 stages would be appropriate:

a) To examine basic dimensions, clearances, finishes, tolerances etc. and to calculate pin bending and ovalisation, plus mean loadings and the lubrication supply and drain features.

b) If the findings from a) and 6.1 are quite normal, a detailed structural analysis of the piston, bushing and pin should be carried out to examine possible distortions leading to local distress between pin and bushing.

6.5 Piston motion analysis would demonstrate the effect of high friction at the piston pin bushing on piston side loadings.

6.6 For early action to reduce sensitivity to skirt side loads, consideration should be given to increasing skirt lubrication by some or all of the following changes, in conjunction with Cooper and other KSV operators:

- Remove the bottom scraper ring
- Replace the upper two scraper rings with those of the more conventional symmetrical rail type and improve oil drainage
- Remove the end caps from the piston pin bores
- Review the use of an all-taper faced compression ring pack

6.7 Confirm with Gulf Oil that there is working experience with SD 40 oil in similar medium speed engines and duties; in particular in other KSV engines and/or engines with Chromium plated liners, tin plated pistons etc. Talk with other operators. Examine the oil specification more closely.

6.8 Longer term changes to piston design, perhaps consequent upon 6.4b), could include revision of the piston ring pack to 3 compression rings, elimination of the bottom ring groove, smaller tin plating thickness and the introduction of some small-scale waviness in the piston skirt surface finish.

6.9 PP&L should further monitor and plot trends in engine data generated during routine tests, oil analyses etc. Certain key data for such plots can be identified so that engine health trends can be closely tracked.

VII. ROOT CAUSE ANALYSIS

1.0 The "B" Diesel Failure

1.1 A single root cause of the "B" diesel, 7L cylinder could not be conclusively determined from the investigation's current findings. Basing this discussion solely on the metallurgical, oil and operating parameter analysis that were completed to this point, we can chart the following "most likely" failure progression for this cylinder.

A. Heavy use of this engine (it has the most running hours and starts of any engine) has caused rings to be worn, tin and iron to be transferred to the liner and end caps to work out towards the liner wall and freeze in place.

B. With the end caps scraping the wall, two things happen:

(1) Heat is built up in them, causing them to expand and bow outwards increasing the pressure against the liner.

(2) The cap scrapes oil from the liner surface reducing the lubrication locally. Metal is transferred back and forth between the cap and liner, roughening both surfaces which increases friction on the rings riding over this surface.

C. The liner surface becomes so hot that remaining oil vaporizes and all lubrication is lost in the area scraped by the end cap.

D. Debris continues to be generated by all sliding surfaces (rings, piston, caps and liner) which either cold welds to the cooler surfaces or is collected by the rings to make a powdery abrasive mixture which accelerates wear on all sliding parts contacted.

E. Eventually so much extra heat is generated that the cooling oil flow through the piston pin and into the head area cannot extract the heat fast enough to control the temperature. The piston begins swelling by the added heat load and eventually fills the normal gap designed into the piston liner assembly.

F. At this time some of the parts have become red hot by the friction and ignition of vaporized oil/air mixtures takes place producing the crankcase explosion. Heaviest damage and overheating is concentrated on the lower half of the piston.

G. Continued operation of the engine for some 5 minutes after the explosion generates additional piston heating and overloading of the piston pin. The extra load on the pin and heat conducted inwards from the extremely hot piston outer surface causes the ends of the pin to become slightly oxidized. In this failure, the pin was tight, but could be rotated by hand with the articulating rod still attached to it.

1.2 Although it is likely that the end caps were contributing elements to the failure process, it is impossible to say that they were the "root cause" with any certainty. The 7R cylinder and piston were examined at the same time as the 7L failure and appeared to be in the early stages of a similar failure. The end caps showed heavy scraping on one end and light burnishing on the other. The 7R liner showed some burnishing and slight roughening at contact points with the end caps.

1.3 The questions that remain to be answered are:

A. Why does metal get transferred to the liner surface if it is covered with a layer of oil?

B. What caused the end caps to migrate out and be forcibly pinned against the liner?

C. Why are the compression and oil rings showing what appears to be excessive wear?

D. If oil quality is questionable, why are other bearing surfaces not experiencing degradation like the piston/liner interfaces? (e.g., the articulating pin, the crank bearings, etc.)

2.0 The "C" Diesel Failure

2.1 While the "C" diesel 5R piston failure appears to the casual observer to be similar in nature to the "B" 7L failure, there are differences that point to a different source of overheating. The following failure progression seems to fit the observations:

A. The pin increases in temperature by frictional heating. This could be caused by loss of clearance between pin and bushing due to improper fit-up or loss of lubrication due to collection of debris on oil holes or bubble entrapment.

B. As heat builds up, bushing-to-pin clearances close further, reducing lubrication, boiling oil from the surfaces and increasing friction further.

C. Locally, temperatures of the pin reach 1200°F or more, decreasing the surface hardness to a very low annealed state and reducing the yield strength to a very low value. Because the heating is localized, the pin shape becomes irregular (i.e., a distorted cylinder) which helps increase localized loading on the bushing.

D. The bushing also sees excessive temperatures, and in fact, may reach the melting point of the eutectic Sn-Pb phase or 361°F. Because the bushing is confined to the space between the piston and the pin, the bushing can only expand in the longitudinal

direction towards the end caps. Contact of the bushing with the end cap inside was observed by scratches and bronze transfer to that surface. The end caps were moved outward to a point where the outside surface of the caps were slightly scored and overheated locally.

E. With the expansions and overheating, the bushing and piston pin became bound together and caused abnormal movement of the piston against the liner. The combination of frictional heat and scraping of piston caused additional oil film loss, further increasing frictional heat and metal scraping.

F. The heat generated cannot be totally removed by the oil flow, so the piston starts heating up and expands radially towards the liner.

G. As the gap between the liner and piston goes down, the friction between them goes up and more heat is generated until the piston seizes in the liner.

H. At this point, temperatures in the piston skirt are so high that the oil vapor/air mixture ignition temperature is reached, and the crankcase explosion occurs.

2.2 Although the initiation of the failure in this case appears to be related to the piston pin overheating and excessive damage in the pin/bushing area, the reason for the overheating is not obvious, so likely causes are:

A. Inadequate lubrication.

B. Air pockets trapped in the oil passages which prevent oil flow.

C. Poor fit-up between bushing and pin.

D. Bent pin.

E. Overloading of pin by excessive combustion pressures.

VIII. INSPECTIONS/TESTING FOR RELIABILITY

The diesel generators are considered Operable when they are capable of performing their specified functions. The diesel generators are designed to provide backup power to safety related equipment in the event of a loss of offsite power supplies. The worst case load demand is the DBA consisting of a LOCA in one unit and a simultaneous forced shutdown of the other unit with a total loss of offsite power. Four diesel generators are shared by two units. Three of four diesel generators are sufficient for the worst case of DBA. Susquehanna SES has increased our diesel reliability by the addition of a fifth diesel generator (E). Since the crankcase overpressurizations, the diesel generators have been subjected to Technical Specification Testing. The following is summary of major testing performed since the recent overpressure events:

- o <u>24-Hour Load Capability Test</u> - Diesels "A," "B," "C," and "D" have been tested to 4700 KW for two hours followed by 22 hours at 4000 KW.

- o <u>LOCA/LOOP Test</u> - Verified capability of diesels to start and energize auto connected LOCA load through sequencing timers. Diesel "E" was substituted for "D."

- o <u>LOOP Test</u> - Verified capability of diesels to start on a LOOP signal and energize buses with permanently connected loads. Diesel "E" was substituted for "D."

- o <u>One Hour Monthly Surveillance Test</u> - Verified capability of diesels to load to 4000 KW within 90 seconds of starting and operate at 4000 KW for a minimum of one hour. Administratively, the diesels are operated up to four hours at 4000 KW with a minimum duration of two hours. The extended run time enables bearing "self-healing" properties to occur and provides added assurance of load carrying capability.

The above testing has been successfully completed. The tests are more demanding on diesel generator capability than worst case DBA loading. This demonstrates the ability of the diesel generators to perform their specified functions. Therefore, they are considered operable per Technical Specifications. As an added indication of the engine ability to perform as designed, the number of hours each engine has been run has been totaled since the crankcase overpressurizations to December 8, 1989. The moderate number of hours which includes several start/stop cycles provides further assurance of diesel capability. The run times are summarized below.

Diesel Generator A	-	42.5 Hours
Diesel Generator B	-	59.1 Hours
Diesel Generator C	-	65.9 Hours
Diesel Generator D	-	47.1 Hours
Diesel Generator E	-	17.8 Hours

In order to assure reliable operation in the near term, an interim inspection has been performed on the "A," "B," "C," and "D" engines. The inspections

identified a number of engine components that were conservatively judged to warrant replacement. The following is a summary of piston and liner replacements since the piston failures on the "B" and "C" engines:

INSPECTIONS

	Date	Replacements
Diesel Generator B	09/23	7L/R Piston/Liner*
	10/26	1R Piston/Liner
Diesel Generator C	10/22	5R Piston/Liner*
		6L Piston*
Diesel Generator D	11/04	8R Piston
Diesel Generator A	12/01	1R, 2R, 7R Piston

*Replaced due to "B" and "C" diesel failures.

With the inspections of all pistons and liners and the replacement of the above components as well as some ring assemblies and other components related to pistons, the engines are considered reliable. Diesel Generator "E" has significantly less hours of operation and does not require component inspection in the near term.

In order to look at the questions of reliability from an additional and completely independent review our Systems Engineering Group was asked to perform the following; identify the most limiting Design Basis Accident (DBA), define the number of diesels required and length of time the diesels would be required to run if the DBA were to occur, determine, based on available data, the overall performance of the diesels, and determine, based on available data, if a high probability of success is assured should the DBA occur.

An assessment was performed to determine the trend of diesel performance, and to determine whether there was a high likelihood of diesel success if LOOP were to occur. Systems Engineering Group conclusion contained in PLI-62326 is there is no increase in total plant damage frequency.

Beginning with Diesel Generator "E" in January all diesels will undergo their 18-month inspections sequentially currently planned through August of 1990. More extensive inspections and component replacements will assure reliable diesel generator operation until the next inspection cycle.

Several Testing methods are being pursued to test the diesels in a less severe manner. Testing, in this manner, will still demonstrate the engine's ability to perform design functions. The one hour monthly test at 4000 KW has already been administratively changed to two/four hours as previously mentioned. The 24 hour surveillance has been changed to include a warm up period prior to testing to the 4700 KW overload capability of the engine. Both changes are per manufacturer recommendations to reduce engine stress. Technical

Specification changes will be proposed to further reduce excess stress and wear on the engines. These changes in periodic testing will extend component lifetimes for added reliable operation.

In summary, the diesel generators have successfully completed all Technical Specification Testing. The testing proves the ability to perform design functions. Therefore, the diesel generators are considered operable per Technical Specifications. To assure continued reliable operation engines "A" through "D" have been inspected and worn piston/cylinder components have been replaced. The upcoming 18-month inspections, combined with worn component changeout, improved periodic tests will assure reliable operation in the future.

IX. APPENDICES

1.0 Ongoing Evaluations

1.1 Piston Pin Bluing By High Temperatures Generated at the Piston Bushing Interface - A blued pin needs to be sectioned to determine the extent of damage to the structure of the pin, particularly the hardened surface. Structural and hardness changes of the microstructure can give the investigator a good idea of the temperatures reached in all areas of the pin. Relating this to surface coloration will give us data needed to determine if the absence of bluing is good enough reason to believe that the pin is not heat damaged. This is necessary because we have measured the straightness of pins that were not showing evidence of bluing and they were found bent a few mils. The theory is that overheating is causing the bending by reducing the yield stress of the pin base metal.

1.2 Liner/Piston Inspection Techniques

1.2.1 The "FAX" replication technique has been used by C-B as a quality control method for determining the porosity of the chrome liner surface. It is also able to reveal if the pores are filling up with metal and if the liner is showing any scoring or abnormal wear patterns. We have taken replicas of various areas of "A" diesel liners opened for a recent inspection. These replicas have not been analyzed yet, but will be done prior to the upcoming "E" diesel inspection in January 1990.

1.2.2 Determination of the elemental composition of the metal contained in the pores is important for two reasons:

(1) It may determine if and how the pores can be cleaned and if the liner can be refurbished or if it must be replaced.

(2) It will tell us what metal is wearing away and how far the damage has progressed.

2.0 Anticipated Changes in:

2.1 Maintenance Procedures

A review of the C-B recommended maintenance practices and PP&L's maintenance procedures was made. A listing of the C-B requirements and how we address them are contained in Attachment 1. A review of the C-B requirements and our implementation of them has not shown anything that could lead to the crankcase explosions.

2.2 Inspection Practices

2.2.1 Criteria for visually examining liners without removing them:

(1) Borescopes from above.

(2) Visual from below.

(3) Replication technique for wear.

2.2.2 Examinations for piston head cracking.

2.2.3 Examination of injection port head region for cracking.

2.2.4 Oil sampling practices and critical parameters.

2.2.5 Debris sampling in:

(1) Oil sump.

(2) Oil filters.

2.2.6 Running parameters:

(1) Peak pressures

(2) Temperatures of oil, cooling water.

2.3 Engine Testing

2.3.1 4700 KW Rating

There was some concern about why we have engines rated for 4700 KW. Attachment 2 contains Memo ME-1148 which concludes the 4700 KW rating was to be consistent with Design Guide 9, Section C. The regulatory requirement is to have the predicted loads not exceed the smaller of the 2,000-hour rating or 90 percent of the 30-minute rating at the operating license state of review.

The predicted loads at that point in time were 4057 KW. Bechtel recommended alternate C of C-B's October 15, 1973 letter to uprate the engine's overload to 4700 KW or 4230 KW at 90 percent of the 30-minute rating).

Review of IOM Requirements vs. PP&L Procedural Requirements

Scope - The requirements in section 15 of IOM 183 were compared with the requirements in PP&L's procedures (MT, SE, OP, SM, and SO).

Other sections in the IOM were reviewed, including drawing notes, for any recommendations which could affect cylinder lubrication. Additionally, Cooper's service bulletins were reviewed for recommendations which could affect cylinder lubrication.

D. A. Wales

Monthly SO-024-001

Cooper Requirements	Comments / Suggestions / Questions
- Log operating parameters during each monthly run	Define which parameters
- Check for signs of water, oil, and air leaks	Also check crankcase vent discharge color? (engine is usually run at night)
- Check for excessive sparking of generator brushes	
	Monthly run should be 4 hours per LEW recommendation (PLI-35179)
	Loading is done in <90 seconds after 10-20 minute warm-up (why 90 seconds?) (TS requirement)
- Monthly oil check during periodic test from flowing line: viscosity, dirt, water, wear materials, acidity. Compare with previous analyses	Check anti-foam additives and other additives? Define limits
- Verify operation of FO transfer pump before running engine	Not necessary, since running the engine will show this
- Verify post-lube pump auto operation record pressure and confirm within normal limits	Define limits
- Record turbo LO pressure after engine shuts down	Define limits
- Five minutes after engine stops record and confirm turbo LO pressure is zero	
- Check governor oil level after shutdown	
- Check for signs of external water, oil, and air leaks after shutdown	
	Check lube oil level before, during, and after run; high level during and after run would indicate foaming and/or JW/FO dillution; level per dwg KSV-1-5 sh 1

Annual SM-024-002; SM-024-A01, B01, C01, D01; SM-024-A02, B02, C02, D02

Cooper Requirements	Comments / Suggestions / Questions
- Procedures are 18 month rather than annual	Acceptable per bulletin 694
- Photographically record analyzer traces	
- Pre and Post maintenance runs should not be less than 4 hour's continuous loaded operation	
- Analyze for unusual conditions	Define acceptance criteria
- Compare vibration survey with previous survey	Define what needs to be surveyed and acceptance
- Check bearings for excessive heat by holding hand near bearings	Can Cooper define a more scientific method?
- Measure cold crankshaft web deflection	
	IOM states "replace elements in lube oil filter and clean strainer as necessary." Does this mean replace elements or replace elements as necessary? SM-024-002 states "replace if required."
- Procedure states to clean crankcase breather and coat with oil. IOM says to clean with solvent and dry.	
- Check operation and calibration of all control and safety shutdown devices. Inspect tubing.	
- Check auxiliary drive vibration damper for dents	
- Check for moisture at indicator cocks on post maintenance runs.	
- Measure and record generator rotor to stator air gap	
- Check overspeed governor set point per bulletin 697	
	Check peak firing pressure and load balance per bulletins 679, 679-A, 711, 711-A
- Clean air inlet casing to generator stator	

Five Year Requirements SM-024-002, SM-024A01, B01, C01, D01

Cooper Requirements	Comments / Suggestions / Questions
- Done every 6 years instead of 5 years	Not clear if this is acceptable per bulletin 694
- Clean collector rings, commutator, brushes, and brush rigging	
- Check rotor pole bolts or keys for tightness	
- Inspect generator frame and all fasteners	
- Inspect exciter	
- Check mechanical condition of all switch gear and relays	May have been covered by SM-024-A02, B02, C02, D02 which have expired
	Inspect FO drain hoses to cylinder head (bulletin 612 - does not state frequency)

Miscellaneous Requirements

Cooper Requirements	Comments/Suggestions/Questions
- 3000 start inspections; No procedure yet	Need acceptance criteria and procedur Need procedure for cylinder liner removal and installation Need procedure for installing piston rings
- Check piston pin bushing for at least 80% contact	Need procedure for removing and installing piston pin bushing. Bulletin 717 states blue check is not required for factory installed bushing
- Air start valve face and seat should be inspected every time a cylinder head is removed	Perform piston inspections every time a piston is removed
- Thoroughly clean and check cylinder head for cracks or wear when head is removed. Dye check the entire face of the cylinder head.	
- Use positioning tool to install cylinder head per bulletin 678	
- Flush and clean intercoolers every 4 to 6 months	
- Crankcase manometer to be filled to "0" mark with SAE #10 oil	
- Bulletin 711-A states 29°-30° BTC for fuel injection pump timing	MT-024-008 Step 6.5.2 states 28° vs. 30° recommendation Need documentation that 28° is acceptable
- Use lube oil meeting the requirements of Cooper standard SE-114.	

TO: [redacted] DATE: 12-8-89

FROM: Tim Wales 2dW

JOB: NUMBER: ME-1148 COPIES TO: Walt Rhoades A6-2

FILE: S024 REPLY: NO

SUBJECT: KSV-16 Diesel Generator Rating

In response to your request to determine our diesel generator rating for diesels A through D, I have located the following references:

1. Telex from Parsons Peebles (N. Monnolly) to PP&L (Barry Skoras) dated 7-13-82 stated generator ratings:

   4000 kW, 0.8 PF, 5000 kVA continuous
   4400 kW, 0.8 PF, 5500 kVA 2 hours/24 hr
   4700 kW, 0.8 PF, 5875 kVA 2000 hrs/year

2. Letter from Portec EP (L. C. Madison) to Cooper (Harris Johnson) dated 10-19-73 states that the proposed generator will be suitable for:

   4000 kW continuously
   4700 kW for 2000 hours
   5050 kW for 200 hours
   5100 kW for 120 hours

3. Letter from Cooper (R. A. Miklos) to PP&L (G. D. Miller) dated 8-15-88 states:

   4000 kW continuous
   4700 kW is 2000 hour overload stated on EP outline drawing D09243. The 10% two hour overload limit is the standard DEMA overload limit, which EP included on all generators. Their method of conveying special overload limits was to list them on the outline drawing.

4. Vendor drawing D09243 (FF105800, Sh. 7901) states:

   4000 kW continuous
   4700 kW (2000 hr)

5. BT Spec 8856-M-30, Rev. 5 for procurement of the DGs, Section 5.2.1 states:

   continuous = 4000 kW
   overload (2000 hours) = 4700 kW
   overload (200 hours) = 4700 kW
   overload (30 mins.) = 4700 kW
   overload (10 seconds) = 4700 kW

6. BLP-3776 dated 4-23-74 from Bechtel (John F. West, Jr.) to PP&L (W. A. Frederick) stated:

   Per Design Guide 9 Section C - Regulatory Position states, "At the operating license stage of review the predicted loads should not exceed the smaller of the 2000 hour rating or 90 percent of the 30 minute rating of the set."

   The BLP noted the corresponding ratings for the diesels now on order are 4000 kW and 3960 kW respectively, and as such are exceeded by the predicted load of 4057 kW. Therefore, the letter noted that the diesels must be uprated. It recommended accepting Alternate C of Cooper's October 15, 1973 letter which quoted an extra of $8,095 to uprate the four diesels to 4700 kW for 2000 hours (or 4230 kW at 90% of the 30 minute rating). This uprating would give spare capacity for future contingencies.

   The price increase covered the supply of largercoolers for the jacket water, air and lube oil systems as well as larger exhaust silencers for all four engines. Neither the engines nor the generators would be changed. However, the size of the excitation system was increased for $1,450 per generator in accordance with a Cooper letter dated 10-22-73. (This 10-22-73 letter cannot be located in SRMS.)

7. Note that with a predicted load of 4057 kW, the minimum 30 minute rating acceptable would have been 4508 kW, per Design Guide 9.

8. Letter from Cooper (H. A. Johnson) to Bechtel (R. C. Higgins) dated 10-15-73 submitted three proposals for uprating the engines. They all included a 4700 kW overload rating. The higher cost options, which were not chosen, included even higher upratings for shorter intervals.

9. Telex from Cooper (T. W. Kearns) to Bechtel (E. B. Poser) dated 4-19-82 states the maximum rating of the diesel generator sets is 4700 kW.

10. Reg. Guide 1.108 section C.2.a.3 requires two hours of the 24 hour diesel test to be at the two hour rating of the diesel generator. From the items above, it is concluded that the two hour DG rating is 4700 kW.

11. Reg. Guide 1.108 section C.2.c.2 requires the diesel to assume load "at the maximum practical rate" for periodic diesel testing during normal plant operation.

<u>Conclusion</u>

The continuous rating for diesel generators A through D is 4000 kW. The short-term rating for any time interval 2000 hours or shorter is 4700 kW.

Maintenance Intervals

Per Cooper's Engineering Standard SA-77, equivalent base load operating hours are determined by multiplying actual hours by factors related to load.

$$He = (Ha_1 f_1 + Ha_2 f_2 + \dots Ha_n f_n)$$

where:

He = Equivalent hours (12,000 maximum)

Ha = Actual hours at a particular load

$$\log_{10} f = \frac{(P-200)^2}{1250} \qquad P \geq 200$$

$$\text{for 16 cylinders, } P = 29.24 \times \frac{kW}{RPM}$$

P = Brake Mean Effective Pressure, psi

For Susquehanna, assume that in an 18 month period a diesel will receive two 24 hours tests with 2 hours at 4700 kW
22 hours at 4000 kW

and

183 four hour tests at 4000 kW (start frequency with ≤ 0.96 reliability)

and

then is required to run at 4700 kW for 31 days

Total loading: Ha_1 = 748 hours at 4700 kW for 148 hours
Ha_2 = 776 hours at 4000 kW

$$P_1 = 29.24 \frac{(4700)}{600} = 229$$

$$P_2 = 29.24 \frac{(4000)}{600} = 195$$

$$\log_{10} f_1 = \frac{(229-200)^2}{1250} \Rightarrow f_1 = 4.7$$

f_2 = 1.0 from Fig. 1 of SA-77

He = (748)(4.7) + (776)(1.0) = 4292 hours which is << 12,000

So our 18 month inspection interval is adequate per this standard.

tgw/me1032c(32)

2.3.2 Remote Emergency Shutdown Switch

After the "B" engine had a crankcase explosion, the need for a remote emergency shutdown switch arose. After this explosion, the engine continued to run since the explosion did not activate any of the normal shutdown trips. The explosion filled the room with smoke making it difficult for the operator to exit. He called the control room to shut down the engine. The control room only has a normal shutdown; hence, after they initiated a shutdown, the engine ran for its 5 minutes cool down cycle. It was decided that a remote emergency shutdown switch was needed. Operators would like to have this switch in the control room. A study will be initiated to determine the best place for this switch and circuit design considering separation and Appendix R criteria.

3.0 QA EVALUATION OF C-B

A review of Cooper's Quality Program including NQA audits of Cooper, Bechtel audits of Cooper, and Bechtel Shop Inspection Reports was conducted. In addition, NQA also reviewed all available NCRs and RDRs, QA Manual reviews, the February 1986 SPAR activity, PP&L Source Inspection Reports, vendor correspondence, and an NQA-Procurement report of major problems encountered with ESQ during 1987. This review was conducted with special interest in the physical components and processes in the area of the piston, liner, connecting rods, and their assembly. This review was conducted by senior members of the NQA organization.

The complete review is contained in PLI-62439, January 3, 1990. This review generally concludes the following:

"In summary, we feel that Energy Services Group of Cooper Industries' (ESG) QA Program is satisfactory. Past assessments of the implementation of their QA Program has generally demonstrated a lack of attention to detail. However, we have seen improvement in receipt inspection results which could indicate an improvement in QA Program implementation. This may be a result due, in part, to increased PP&L QA involvement."

The improved receipt inspection results referred to are summarized in Figure 3.1. The "Specific Results and Conclusions" section is currently under review in engineering. We do intend to research, with Cooper, specific shop records that might be available on the failure of the 4R piston of the "C" engine during shop acceptance testing. However, our present opinion is this event was caused by improper fit-up of the piston pin to bushing contact surface area and was corrected when the new piston/pin assembly was installed. This event does serve to further highlight the importance of the pin/bushing contact area. It is one we are desirous of assuring the best possible fit. All ("A" through "D") pistons to piston pin assemblies will be "blue" checked during the upcoming 18-month inspections to assure ourselves that optimal conditions exists.

COMPARISON OF RECEIPT DISCREPANCES AGAINST TOTAL RECEIPTS

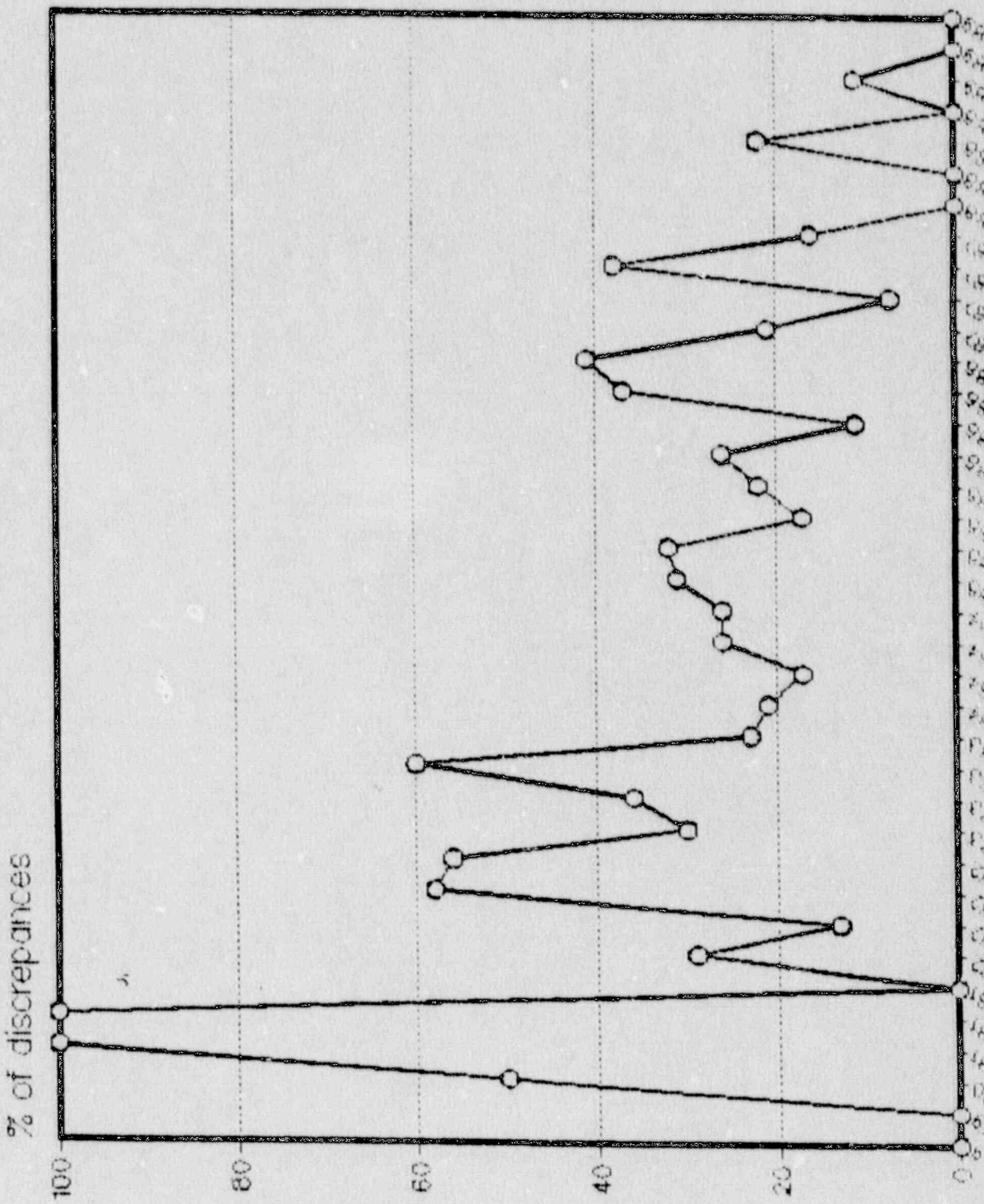


Figure 3.1

4.0 Attachments

4.1 Chemical Analysis of Diesel Parts

Established 1924

HENRY J. YEAGER LABORATORIES

CHEMIST AND METALLURGIST
ROSS AND MARKET STREETS
P. O. BOX 4063
LANCASTER, PA. 17604

TELEPHONE: 717-397-9614 FAX / 717-397-2580

SAND CONTROL FOUNDRY CONSULTANT TENSILE TESTING

CERTIFICATE OF ANALYSIS

Received from Penna. Power & Light Lab. # 213952

Identifying Mark P.O. S-06182-5 Five Samples Reported 10/19/89

	Sample 1 B Piston	Sample 2 C Piston	Sample 3 B Oil Ring	Sample 4 C Oil Ring	Sample 5 B Compression Ring
Silicon	1.57	1.66	1.81	1.79	2.51
Manganese	.989	.960	.621	.630	.621
Phosphorus					
Sulfur					
Graph. Carbon					
Comb. Carbon					
Total Carbon	3.22	3.18	3.42	3.47	3.39
Chromium	.079	.162	.320	.334	.101
Nickel	.154	.422	.054	.098	.082
Molybdenum	.035	.125	.368	.357	.030
Copper	.151	.145	.062	.064	.068
Magnesium					
Iron	93.65	93.18	92.88	92.79	92.72

Very truly yours,

HENRY J. YEAGER LABORATORIES

Per ....

Stewart M. Beyerle
CHIEF METALLURGIST

Established 1924

HENRY J. YEAGER LABORATORIES
CHEMIST AND METALLURGIST
ROSS AND MARKET STREETS
P. O. BOX 4063
LANCASTER, PA. 17604

TELEPHONE: 717-397-9614 FAX # 717-397-2580

SAND CONTROL FOUNDRY CONSULTANT TENSILE TESTING

CERTIFICATE OF ANALYSIS

Received from Penna. Power & Light Lab. # 213953
Identifying Mark P.O. S-06183-5 Two Samples Reported 10/19/89

	Sample 6 B End Cap	Sample 7 C End Cap
Silicon	2.23	1.86
Manganese	.419	.874
Phosphorus		
Sulfur		
Graph. Carbon		
Comb. Carbon		
Total Carbon	3.28	3.19
Chromium	.077	.083
Nickel	.051	.141
Molybdenum	.012	.053
Copper	.143	.158
Magnesium		
Iron	93.20	93.48

Very truly yours,

HENRY J. YEAGER LABORATORIES

Per ....

Stewart M. Beyerle
CHIEF METALLURGIST

4.2 Oil Analysis for All SSES Diesels

4.2.1 June Oil Test Reports for "A," "B," "C," "D," and "E" Diesels

20-50-06

Released - SAS
JUL 13 1989

Mr. C. E. Burke

7-11-89
Date

OIL TEST REPORT

Susquehanna S.E.S. STATION

	6701089	6711089	6721089	6731089	6741089	
Station Sample Number	89-12296	89-12298	89-12300	89-12302	89-12304	
Date Taken	6-10-89					
Date Received	6-26-89					
Type of Oil	Super Duty 40					
Type of Equipment	EDG-A	EDG-B	EDG-C	EDG-D	EDG-E	
Supplier	Gulf Oil Co					
Viscosity at 100°F. S.S.U.	771	718	780	771	779	
Viscosity at 210°F. S.S.U.						
Flash, °F. PMCC	410	420	410	425	400	
Fire, °F.						
Acidity, Mg. KOH/GM	0.37	0.76	0.46	0.43	0.43	
Moisture, % Dist.	0.07	0.08	0.06	0.06	0.03	
Sediment, %						
Steam Emulsion Number						
Viscosity Index						
Carbon Residue, %	1.06	1.04	1.08	1.03	1.19	
Ash in Oil, %	0.80	0.84	0.78	0.78	0.82	
Pour, °F.						
Interfacial Tension, dynes/cm.						
Color						
Oxidation Inhibitor Content, %						
Rotary Bomb Oxidation Test, Min.						
Pentane Insoluble, %	0.02	0.02	0.02	0.00	0.00	
Benzene Soluble, %						
Ferrography	see 1 attachment					
metals	see 2 attachment					
Meets Manufacturer's Specs.						
Suitable for Further Use	✓	✓	✓	✓	✓	
See Comments Below						

Recommendations:

Copy to: A. Rieder D. Gandenberger
E.W. Figard J.M. Hettinger
T.J. Nork

Bob Bason

R.W. Snyder/ps

DISSOLVED METALS IN OIL

M C.E. Burke

Date 28 June 1989

Station Susquehanna SES

	6701089	6711089	6721089	6731089	6741089
Station Sample Number	89-12296	89-12298	89-12300	89-12302	89-12304
Date Taken	6/10/89				
Date Received	6/26/89				
Type of Oil	Super Duty 40				
Type of Equipment	EDG 'A'	~~EDG 'B'~~	EDG 'C'	EDG 'D'	EDG 'E'
Oil Supplier	Gulf Oil Co.				
Metals-Parts Per Million (PPM)					
Iron (Fe)	3.6	3.2	4.1	2.7	2.2
Copper (Cu)	3.3	3.0	2.7	2.3	2.1
Chromium (Cr)	<1.	<1.	<1.	<1.	<1.
Lead (Pb)	<1.	<1.	<1.	<1.	<1.
Tin (Sn)	<2.	<2.	<2.	<2.	<2.

COMMENTS: Typical — very little soluble wear metal present

COPY TO: A. Rieder
E.W. Figard
D. Gandenberger
J.M. Hettinger
T.J. Nork

Paul [illegible]
Chem Lab

Susquehanna S.E.S.

ATTACHMENT 6

Lab Sample Number	Station Sample Number	Date Taken	Date Received	Type of Equipment	WPC	PLP	Comments
701089	89-12298	6/10/89	6/26/89	EDG 'A'	1.8	44.4	ok
711089	89-12298	6/10/89	6/26/89	[illegible]	2.9	3.4	ok
721089	89-12300	6/10/89	6/26/89	EDG 'C'	2.5	12.0	ok
731089	89-12302	6/10/89	6/26/89	EDG 'D'	5.6	7.1	ok
741089	89-12304	6/10/89	6/26/89	EDG 'E'	3.5	8.6	ok

PLP = % Large Particles = $\frac{L-S}{L+S} \times 100$

WPC = Wear Particle Conc. = $\frac{Lg + Sm}{Sample\ Size}$

Chem Lab Paul Sup...

4.2.2 July Oil Test Reports for "A," "B," "C," "D," and "E" Diesels

ATTACHMENT

DISSOLVED METALS IN OIL

M C.E. Burke

Date 26 July 1989

Station Susquehanna SES

	733L089	734L089	735L089	736L089	737L089
Station Sample Number	89-15451	89-15217	89-15215	89-15219	89-15221
Date Taken	7/17/89	7/17/89			
Date Received	7/20/89				
Type of Oil	Super Duty 40				
Type of Equipment	EDG 'A'	[redacted]	EDG 'C'	EDG 'D'	EDG 'E'
Oil Supplier	Gulf Oil Co.				
Metals-Parts Per Million (PPM)					
Iron (Fe)	3.5	2.8	3.5	2.4	1.9
Copper (Cu)	3.2	2.3	2.2	1.9	1.4
Chromium (Cr)	<1.	<1.	<1.	<1.	<1.
Lead (Pb)	<1.	<1.	<1.	<1.	<1.
Tin (Sn)	<2.	<2.	<2.	<2.	<2.

COMMENTS: OK

COPY TO: A. Rieder
E.W. Figard
D. Gandenberger
J.M. Hettinger
T.J. Nork

Paul Syer
Chem Lab

Released - [illegible]

AUG 14 1989

Mr. C.E. Burke

8-9-89
Date

OIL TEST REPORT

Susquehanna S.E. STATION

	733L089	734L089	735L089	736L089	737L089	
Station Sample Number	89 15457	89 15217	89 15215	89-15219	89-15221	
Date Taken	7-17-89	7/14/89			→	
Date Received	7-20-89				→	
Type of Oil	Super Duty 40				→	
Type of Equipment	EDG-A	EDG-B	EDG-C	EDG-D	EDG-E	
Supplier	Gulf oil co				→	
Viscosity at 100°F. S.S.U.	725	680	726	716	721	
Viscosity at 210°F. S.S.U.						
Flash, °F. Pmcc	410	405	395	405	425	
Fire, °F.						
Acidity, Mg. KOH/GM	0.40	0.70	0.42	0.45	0.17	
Moisture, % Dist	0.01	TRACE	TRACE	TRACE	0.02	
Sediment, %						
Steam Emulsion Number						
Viscosity Index						
Carbon Residue %	1.15	1.12	1.12	1.10	1.20	
Ash in Oil, %	0.80	0.86	0.81	0.81	0.78	
Pour, °F.						
Interfacial Tension, dynes/cm.						
Color						
Oxidation Inhibitor Content, %						
Rotary Bomb Oxidation Test, Min.						
Pentane Insoluble, %	0.03	0.00	0.03	0.00	0.00	
Benzene Soluble, %						
Ferrography	See ① Attachment				→	
metals	See ② Attachment				→	
Meets Manufacturer's Specs.						
Suitable for Further Use	✓	✓	✓	✓	✓	
See Comments Below						

Recommendations:

Copy to: A. Rieder D. Gandenberger
E.W. Figard
T.J. Nork J.M. Hettinger

B. Fegley

A.W. Snyder / BS

Susq. hanna S.E.S.

ATTACHMENT ②

Lab Sample Number	Station Sample Number	Date Taken	Date Received	Type of Equipment	WPC	PLP	Comments
733L089	89-15451	7/17/89	7/20/89	EDG 'A'	2.2	36.4	ok
734L089	89-15217	7/14/89	7/20/89	[redacted]	3.1	⊖3.2	ok
735L089	89-15215	7/19/89	7/20/89	EDG 'C'	2.9	10.3	ok
736L089	89-15219	7/14/89	7/20/89	EDG 'D'	6.0	6.7	ok
737L089	89-15221	7/17/89	7/20/89	EDG 'E'	3.7	8.1	ok

PLP = % Large Particles = $\frac{L-S}{L+S} \times 100$

WPC = Wear Particle Conc. = $\frac{Lg + Sm}{\text{Sample Size}}$

Chem Lab ____________

4.2.3 August Oil Test Reports for "A," "B," "C," "D," and "E" Diesels

LO-SQ-DG

Mr. C. E. Burke

24 Aug. 1989
Date

AUG 25 1989

OIL TEST REPORT
Susquehanna S.E. STATION

	829LO89	830LO89	831LO89	832LO89	833LO89	
Station Sample Number	89-17797	89-17798	89-17799	89-17800	89-17801	
Date Taken	8-14-89					
Date Received	8-16-89					
Type of Oil	Super Duty HD					
Type of Equipment	EDG-A	[redacted]	EDG-C	EDG-D	EDG-E	
Supplier	Gulf Oil Co					
Viscosity at 100°F. S.S.U.	733	711	735	737	743	
Viscosity at 210°F. S.S.U.						
Flash, °F. PMCC	400	390	400	390	395	
Fire, °F.						
Acidity, Mg. KOH/GM	0.53	1.00	0.51	0.58	0.52	
Moisture, % Dist.	Trace	Trace	Trace	Trace	Trace	
Sediment, %						
Steam Emulsion Number						
Viscosity Index						
Carbon Residue, %	1.22	1.04	1.18	1.08	1.37	
Ash in Oil, %	0.82	0.83	0.80	0.80	0.85	
Pour, °F.						
Interfacial Tension, dynes/cm.						
Color						
Oxidation Inhibitor Content, %						
Rotary Bomb Oxidation Test, Min.						
Pentane Insoluble, %	0.00	0.01	0.02	0.04	0.02	
Benzene Soluble, %						
Ferrography	See attachment					
metals	See attachment					
Meets Manufacturer's Specs.						
Suitable for Further Use	✓	✓	✓	✓	✓	
See Comments Below						

Recommendations:

Copy to: A. Rieder, E. W. Figard, T. J. Nork, D. Gandenberger, J. M. Hetlinger

P. E. Sine

A. W. Snyder/PS

ATTACHMENT 4

DISSOLVED METALS IN OIL

M C.E. Burke

Date '24 Aug. 1989

Station Susquehanna

	8291089	8306089	8311089	8321089	8331089
Station Sample Number	89-17797	89-17798	89-17799	89-17800	89-17801
Date Taken	8/14/89				
Date Received	8/16/89				
Type of Oil	SuperDuty 40				
Type of Equipment	EDG 'A'	EDG 'B'	EDG 'C'	EDG 'D'	EDG 'E'
Oil Supplier	Gulf Oil Co.				
Metals-Parts Per Million (PPM)					
Iron (Fe)	3.7	3.1	4.0	2.8	2.3
Copper (Cu)	3.5	2.8	2.7	2.2	2.0
Chromium (Cr)	<1.	<1.	<1.	<1.	<1.
Lead (Pb)	<1.	<1.	<1.	<1.	<1.
Tin (Sn)	<2.	<2.	<2.	<2.	<2.

COMMENTS: Typical

COPY TO: A. Rieder
E.W. Figard
D. Gandenberger
J.M. Hettinger
T.J. Nork

Paul Super
Chem Lab

FERROGRAPHY

Susquehanna S.E.S.

ATTACHMENT 2

Lab Sample Number	Station Sample Number	Date Taken	Date Received	Type of Equipment	WPC	PLP	Comments
829L089	89-17797	8/14/89	8/16/89	EDG 'A'	2.6	30.8	ok
830L089	89-17798	8/14/89	8/16/89	~~EDG~~ 'B'	3.3	3.0	ok
831L089	89-17799	8/14/89	8/16/89	EDG 'C'	3.3	9.1	ok
832L089	89-17800	8/14/89	8/16/89	EDG 'D'	6.4	6.3	ok
833L089	89-17801	8/14/89	8/16/89	EDG 'E'	3.3	9.1	ok

PLP = % Large Particles = $\frac{L-S}{L+S} \times 100$

WPC = Wear Particle Conc. = $\frac{Lg + Sm}{\text{Sample Size}}$

Chem Lab Paul Burger

4.2.4 September Oil Test Reports for "A," "B," "C," "D," and "E" Diesels

Mr. C.E. Burke

20.34.06
9-28-89
Date

OIL TEST REPORT

Susquehanna S.G. STATION

	929L089	930L089	931L089	932L089		
Station Sample Number	89-20174	89-20176	89-20178	89-20180		
Date Taken	9-11-89	→	→	→		
Date Received	9-25-89	→	→	→		
Type of Oil	Super Duty 40	→	→	→		
Type of Equipment	EDG-A	EDG-B	EDG C	EDG D		
Supplier	Gulf Oil Co	→	→	→		
Viscosity at 100°F. S.S.U.	728	696	756	747		
Viscosity at 210°F. S.S.U.						
Flash, °F. PMCC	405	415	400	415		
Fire, °F.						
Acidity, Mg. KOH/GM	0.60	0.61	0.38	0.40		
Moisture, % Dist.	TRACE	TRACE	TRACE	TRACE		
Sediment, %						
Steam Emulsion Number						
Viscosity Index						
Carbon Residue, %	1.18	1.06	1.05	1.09		
Ash in Oil, %	0.82	0.87	0.81	0.83		
Pour, °F.						
Interfacial Tension, dynes/cm.						
Color						
Oxidation Inhibitor Content, %						
Rotary Bomb Oxidation Test, Min.						
Pentane Insoluble, %	0.05	0.01	0.01	0.04		
Benzene Soluble, %						
Ferrography	SEE ATTACHMENT ①	→	→	→		
metals	See ATTACHMENT ②	→	→	→		
Meets Manufacturer's Specs.						
Suitable for Further Use	✓	✓	✓	✓		
See Comments Below						

Recommendations:

Copy to: A. Reider D. Gandenberger
E.W. Figard J.M. Hattinger
T.J. Nork J. Adams

B. Fegley
Chem. Lab.

A.W. Snyder
A. W. Snyder

DISSOLVED METALS IN OIL

To C.E. Burke

Date 27 Sept. 1989

Station Susquehanna SES

	929L089	930L089	931L089	932L089	
Station Sample Number	89-20174	89-20176	89-20178	89-20180	
Date Taken	9/11/89				
Date Received	9/25/89				
Type of Oil	Super Duty 40				
Type of Equipment	EDG 'A'	[illegible]	EDG 'C'	EDG 'D'	
Oil Supplier	Gulf Oil Co.				
Metals-Parts Per Million (PPM)					
Iron (Fe)	3.6	2.8	3.4	3.2	
Copper (Cu)	3.1	2.5	2.4	2.0	
Chromium (Cr)	<1.	<1.	<1.	<1.	
Lead (Pb)	<1.	<1.	<1.	<1.	
Tin (Sn)	<2.	<2.	<2.	<2.	

COMMENTS:

COPY TO: A. Rieder
J. Adams
E.D. Figard
D. Gandenberger
J.M. Hettinger
[illegible]

Chem Lab A.W. Snyder / TJJ

FER[illegible]GRAPHY

Susquehanna S.E.S.

ATTACHMENT

Lab Sample Number	Station Sample Number	Date Taken	Date Received	Type of Equipment	WPC	PLP	Comments
929L089	89-20174	9/11/89	9/25/89	EDG 'A'	2.3	30.4	
930L089	89-20176	9/11/89	9/25/89	EDG 'B'	2.9	3.4	
931L089	89-20178	9/11/89	9/25/89	EDG 'C'	3.1	9.7	
932L089	89-20180	9/11/89	9/25/89	EDG 'D'	5.6	7.1	

PLP = % Large Particles = $\frac{L-S}{L+S} \times 100$

WPC = Wear Particle Conc. = $\frac{Lg + Sm}{Sample\ Size}$

Chem Lab A. W. Snyder /TBD

4.2.5 Oil Testing Report Comparing "New" Oil and "Used" Oil - "A" Diesel

DEC 18 '89 10:54 PPD CHEM LAB P02

Mr. BE Rhoads

12-11-89
Date

OIL TEST REPORT

Susquehanna SE STATION

	[illegible]	[illegible]	[illegible]			
Station Sample Number	89 9911	89-[illegible]	USED			
Date Taken	5-7-89	8-2-89	12-1-89			
Date Received	12-7-89	12-7-89	12-7-89			
Type of Oil	[illegible]	→	[illegible] →			
Type of Equipment	1K2030 B-S18 comp	1K205 A S18 comp	[illegible] sample			
Supplier	Gulf oil co	→	→			
Viscosity at 100°F, S.S.U.	571	554	758			
Viscosity at 210°F, S.S.U.						
Flash, °F. COC	470	465	440	[illegible]		
Fire, °F.	510	500	490			
Acidity, Mg. KOH/GM	0.74	0.84	0.48			
Moisture, %						
Sediment, % 50/50	0.00	0.00	0.00			
Steam Emulsion Number						
Viscosity Index						
Carbon Residue, %			1.09			
Ash in Oil, %			0.00			
Pour, °F.						
Interfacial Tension, dynes/cm.						
Color						
Oxidation Inhibitor Content, %						
Rotary Bomb Oxidation Test, Min.						
Pentane Insoluble, %						
Benzene Soluble, %						
Ferrography	See attached					
Foam			8.6			
Burnt Odor			✓			
Meets Manufacturer's Specs.						
Suitable for Further Use						
See Comments Below						

Recommendations:

Copy to: A. Rieder J.M. Hettinger
J.R. Adams T.J. Noak
E.W. Firred D. Roche

[illegible]

DEC 18 '89 10:55 PPD CHEM LAB P03
ATTACHMENT W

DISSOLVED METALS IN OIL

M B.E. Rhoads

Date 13 Dec. 1989

Station Susquehanna SES

	12DD6089				
Station Sample Number					
Date Taken	12/1/89				
Date Received	12/9/89				
Type of Oil	Super Duty 40				
Type of Equipment	OB EG 1A Dip Sample				
Oil Supplier	Gulf Oil Co.				
Metals-Parts Per Million (PPM)					
Iron (Fe)	2.6				
Copper (Cu)	1.7				
Chromium (Cr)	<1.				
Lead (Pb)	<1.				
Tin (Sn)					

COMMENTS:

COPY TO: A. Rieder
S.R. Adams
E.D. Figard
D. Gaudenberger
J.M. Hollinger
T.S. Nork

Chem Lab

FERROGRAPHY

Susquehanna S.E.S.

Attachment

Lab Sample Number	Station Sample Number	Date Taken	Date Received	Type of Equipment	WPC	PLP	Comments
1158L089	87-7911	5/7/89	12/7/89	1K208B Turbine [illegible]	13.8	49.3	
1199L089	89-16829	9/2/89	12/7/89	1P206A CSG	12.0	45.0	
2000L089	OG Sol A	12/6/89	12/7/89	Dip Sample	12.0	48.3	

PLP = % Large Particles = $\frac{L-S}{L+S}$ X 100

WPC = Wear Particle Conc. = $\frac{Lg + Sm}{\text{Sample Size}}$

Chem Lab ____________________

4.3 Fuel Oil Test Report

FO-SQ

Mr. [redacted]

12-13-89
Date

Fuel Oil Test Report

SUSQUEHANNA SES Station

	179DEC89					
Combustion Turbine Sites	C-DAY TANK					
Station Sample Number	89-25683					
Date Taken	11-2-89					
Date Received	12-7-89					
Type of Fuel	#2					
Supplier						
Refiner						
Specific Gravity at 60°F.	0.8571					
Deg. A.P.I. at 60°F.	33.6					
Water, % by Volume V/S	0.00					
Sediment, % by Volume V/S	0.00					
Distillation, °F.						
10%	426					
50%	510					
90%	598					
Cetane Number (Calculated)	45					
Conradson Carbon Residue in 10% Bottoms,%	0.005					
Ash in Oil, %	0.000					
Pour, °F.	-10					
Color, A.S.T.M.	<3.5					
Viscosity at 100°F., S.S.U.	—					
Sulfur, %	0.12					
Flash, °F.	146					
Heating Value, BTU/Gal. (CALC.)	138,748					
Manganese, ppm	—					
HEATING VALUE BTU/LBS. (CALC)	19,443					
HEATING VALUE BTU/LBS. (DETERMINED)	19,415					
Meets Refiners Specifications	✓					
See Comments Below						

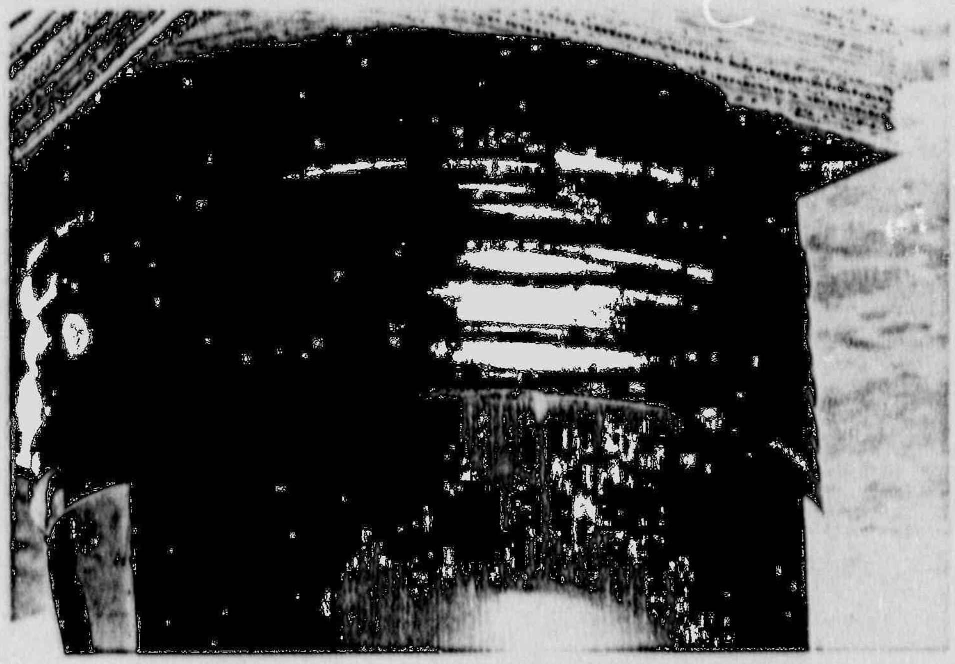
Comments:

Copy to: JIM ADAMS SSES

Bob Baum
Chem. Lab.

A. W. Snyder/BS
A. W. Snyder

4.4 Discussion of Findings From "D" Diesel Inspection



a

b

Figure D-1. (a) Non-thrust side of 8R piston showing wear between No. 1 and No. 2 compression rings. (b) Head view of 8R piston.

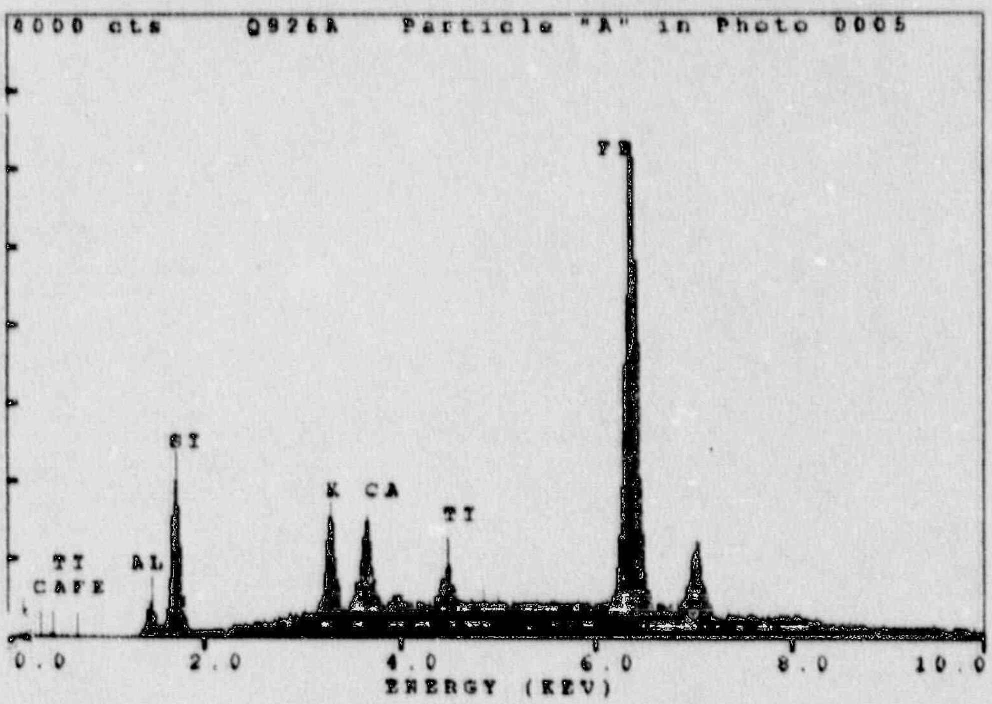
c

d

Figure D-1. (c) Thrust (south) side of 8R piston showing no deposits on head. (d) East (generator end) of 8R piston - note complete removal of tin coating on left side.

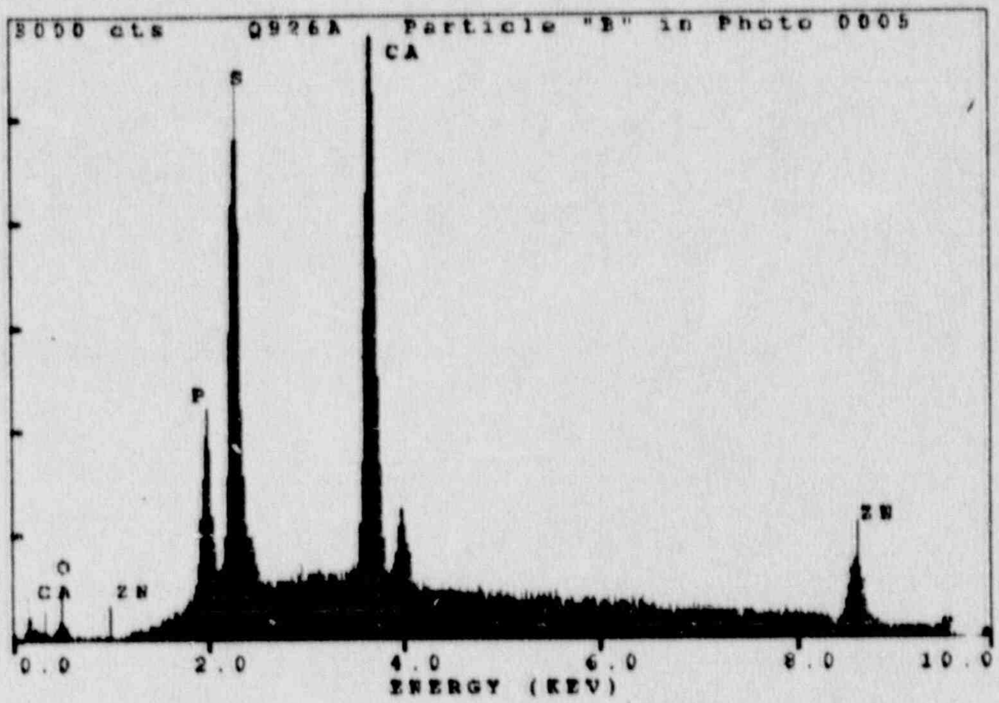
Figure D-1. (e) Non-thrust (north) side of 8R piston showing removal of tin and scoring of piston surface.

a



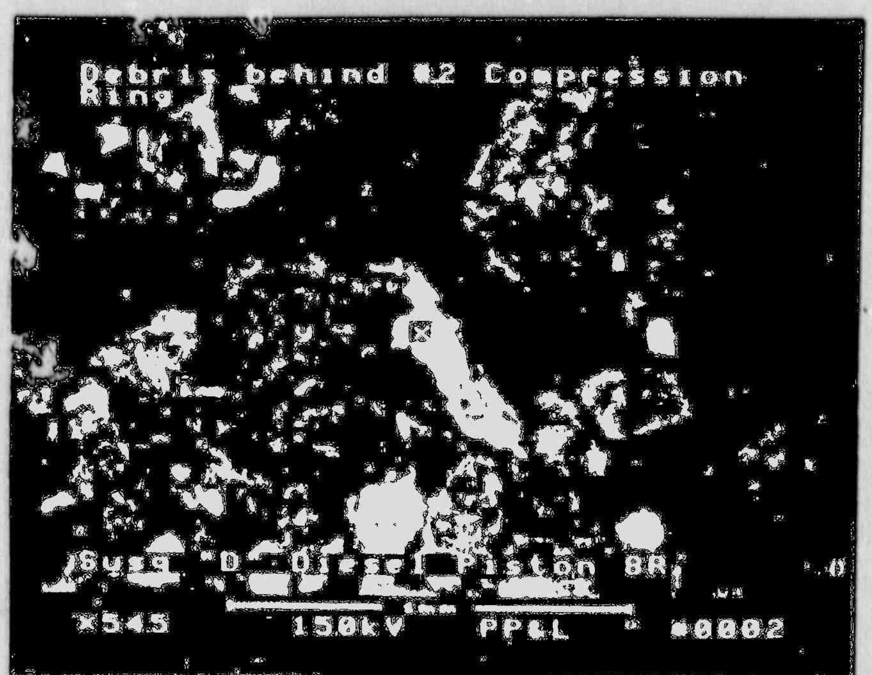
b

Figure D-2. (a) Caked deposits from the top compression ring of the 8R cylinder. (b) EDS analysis of deposits from area "A."

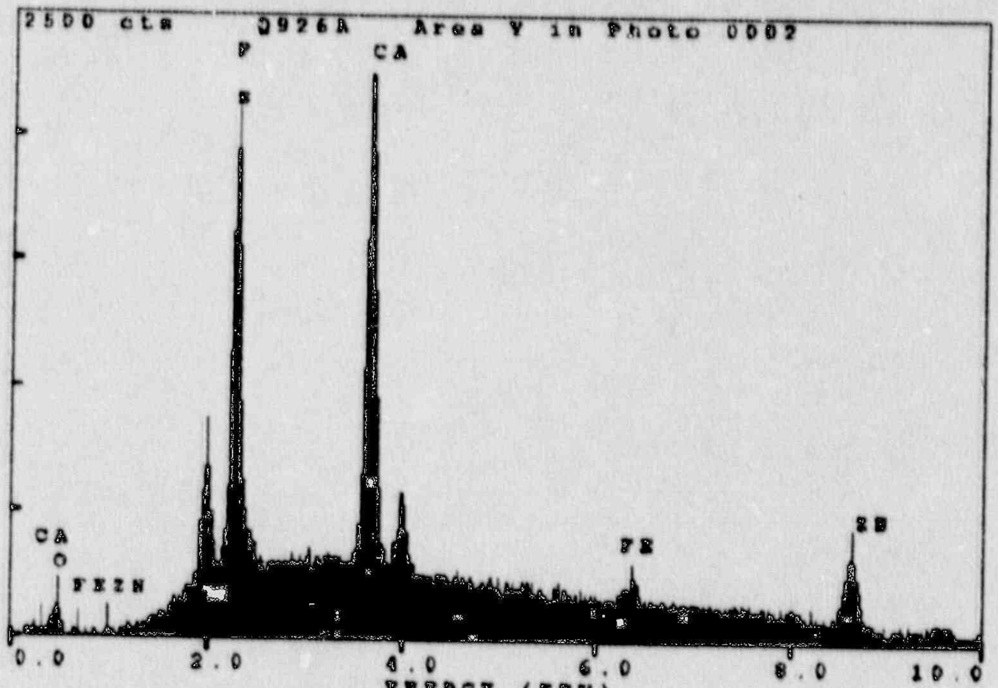


C

Figure D-2c. EDS analysis from area "B" of Figure D-2A.

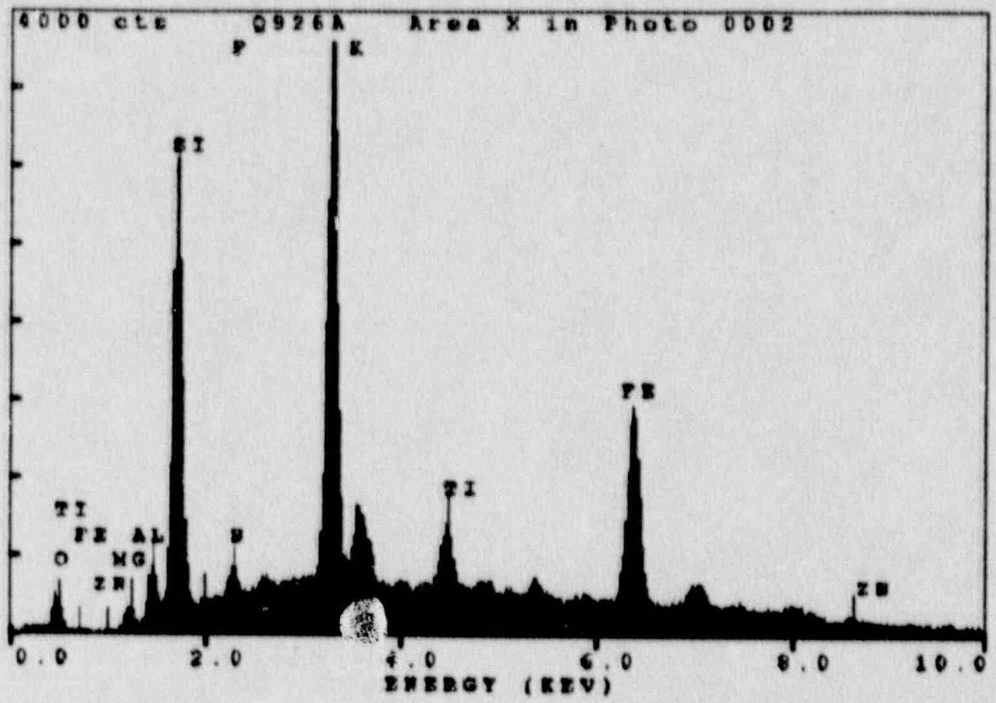


a



b

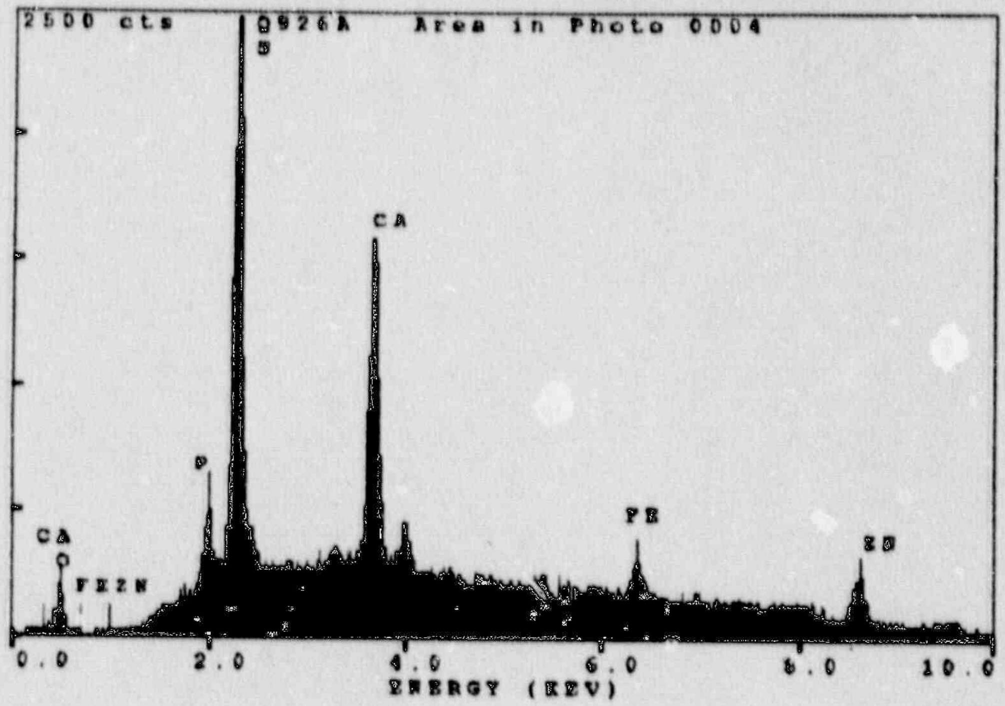
Figure D-3. (a) Debris from behind the No. 2 compression ring. (b) EDS of debris (area "Y") showing presence of P, S, Ca, and Zn.



C

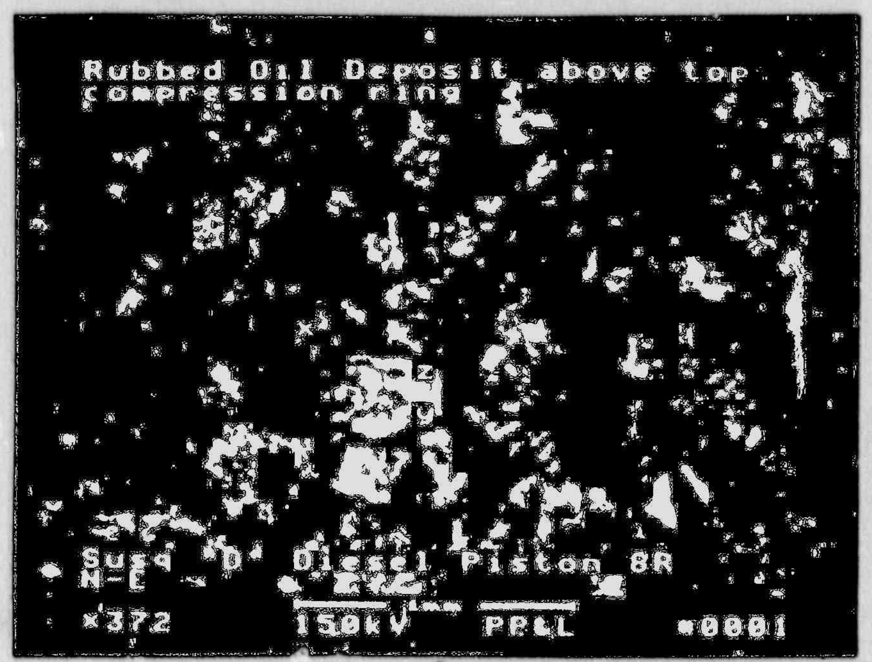
Figure D-3C. EDS of debris (area "X") in Figure D-3a showing the presence of K, Si, Fe, Ti, S, Al and Mg.

a

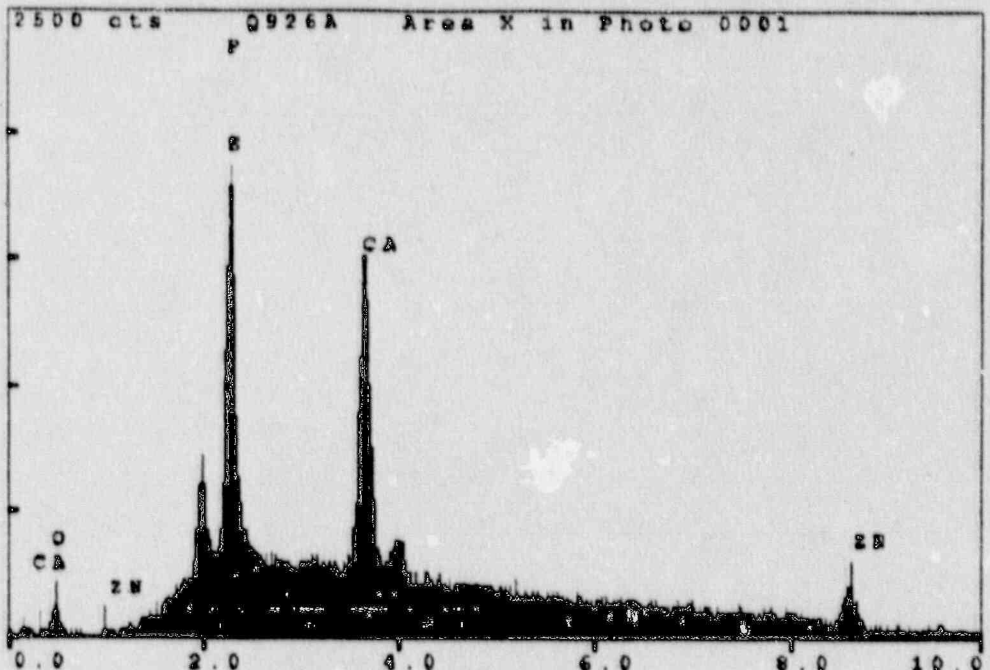


b

Figure D-4. (a) Debris from the lower oil ring of the "D" diesel 8R cylinder. (b) EDS analysis showing the elemental content of the debris in (a).

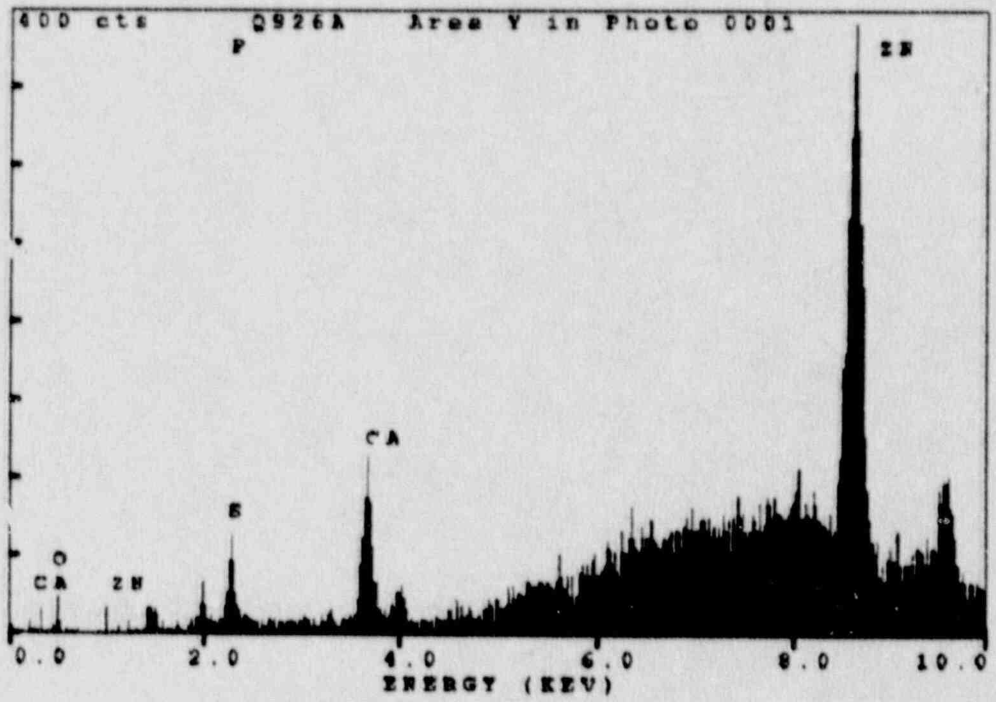


a

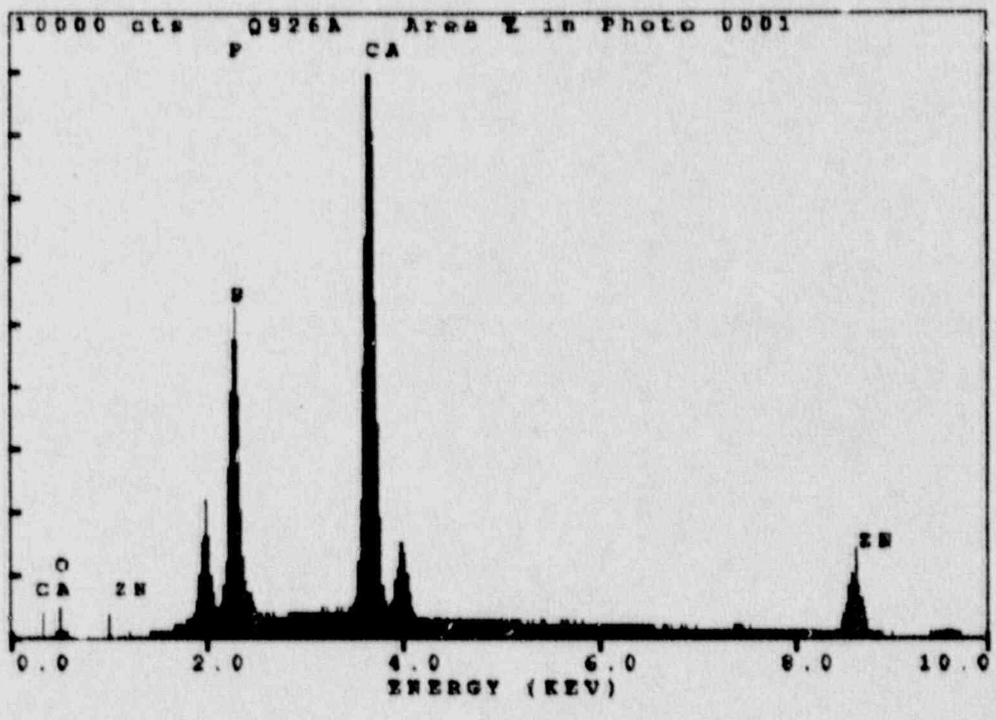


b

Figure D-5. (a) Oil deposits from the top compression ring of the "D" diesel 8R piston. (b) EDS analysis of particle "X" in picture (a) showing the presence of S, Ca, P and Zn.



c



d

Figure D-5. (c) EDS elemental content of area "Y" in (a) showing the presence of Zn, Ca and S. (d) EDS elemental content of area "Z" in (a) showing the presence of Ca, S, P and Zn.

4.5 Discussion of Finding From "A" Diesel Inspection

4.5.1 The "A" D/G was inspected during the middle of November 1989. Three cylinders were found to contain unacceptable levels of smeared tin on the liner. These were 1R, 2R, and 7R cylinders. Also, another piston, 3R, was pulled from a cylinder whose liner showed no signs of tin smearing.

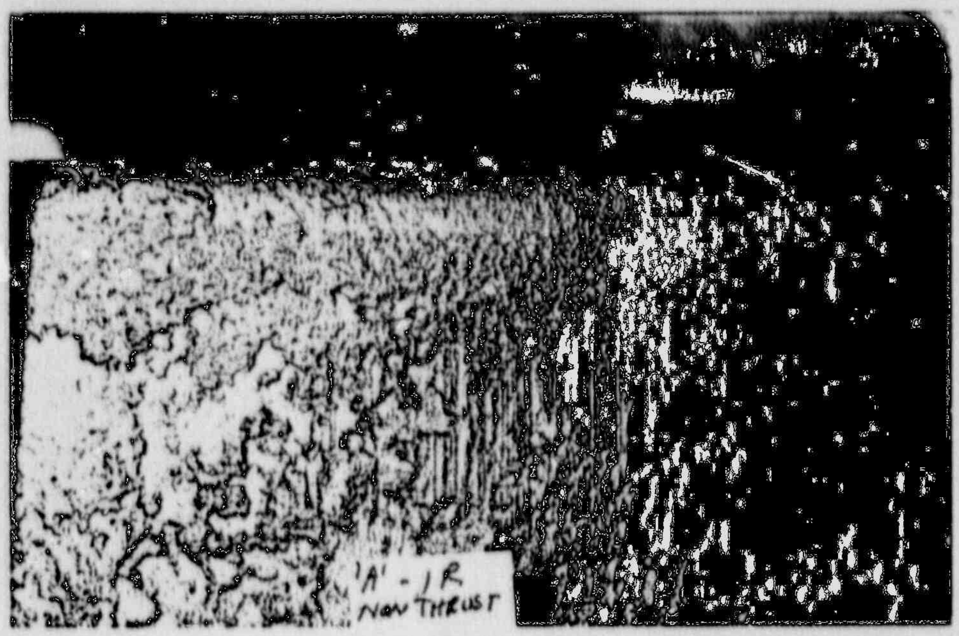
The three pistons from the "tinned" liners showed areas of high buildup of debris on the non-thrust side. The entire taper on all the compression rings for the three pistons was worn completely. Figures A-1a, b and c show the debris buildup on the non-thrust side of the three pistons.

The 3R piston, although from a cylinder with no indications of "tinning," showed almost the same conditions as the other three pistons. The top two compressions ring tapers were extensively worn, 100 percent and 60 percent, respectively. The non-thrust side of the top of the piston contained large amounts of the debris seen on the other pistons. The contents of this debris is the same as those analyzed for the "D" D/G 8R piston - unburned lube oil deposits. In addition, the second (middle) oil ring had a piece broken off. It was assumed that the piece became lodged between the piston and the liner. The liner showed no indication of any scratching or scoring. The tin on the non-thrust side of the piston contained long scratches between the second and bottom oil rings. Refer to Figure A-2 for the photo of the non-thrust side of the piston. Figure A-3 is a photo showing the deposits and scrape marks on those deposits on the non-thrust side of the 3R piston.

The pins from the 1R and 7R pistons were found to have areas of "bluing," indicating overheating. The pins were tested for hardness. The areas of overheating were softer, 20-25 Rc, than the remainder of the pin, 50 Rc. The 1R pin was bent .002" between the bolt holes and .004" across the length 180° from the bolt holes. The pins were analyzed on a Texas Nuclear Alloy Analyzer with the results of SAE #5046 material. Photos of the discoloration on the 1R piston pin can be seen in Figure A-4.

The compression rings of the 1R, 2R, 3R, and 7R pistons were dimensionally checked. All were within the proper tolerances considering the amount of wear. Attachment 7 in Chapter VI contains a listing of these dimensions.

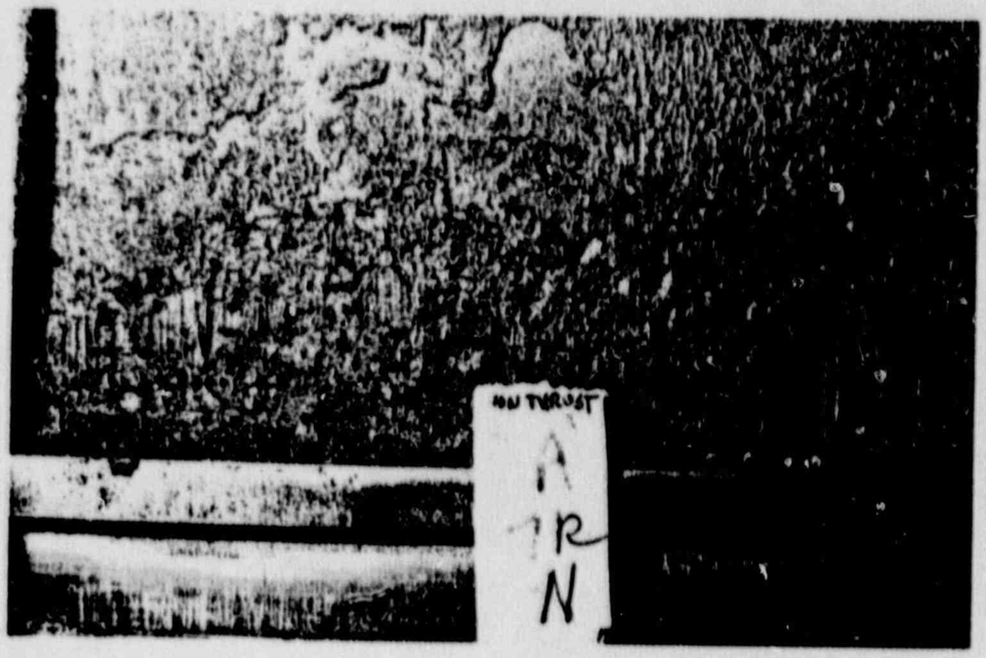
When Maintenance removed the heads from the 8R and 3R cylinders, they discovered damaged gaskets. The gaskets, shown in photos on Figures 5a and b, had disintegrated to a point where just the metal portion of the gasket remained; no binder was left. Cooper has substituted new gaskets that no longer deteriorate.



a

b

Figure A-1. Closeup photos of debris on the non-thrust side of pistons 1R, 2R and 7R (a, b and c, respectively).



c

Figure A-1. Continued.

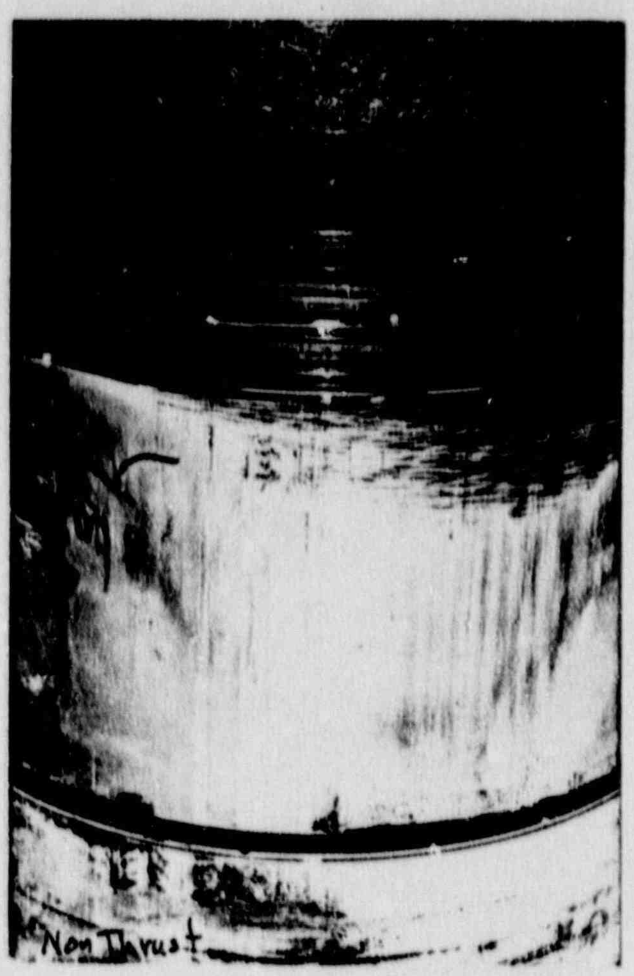


Figure A-2. Photo of the longitudinal scratches on the non-thrust side of the 3R piston - probably caused by the piece broken off the middle oil ring.

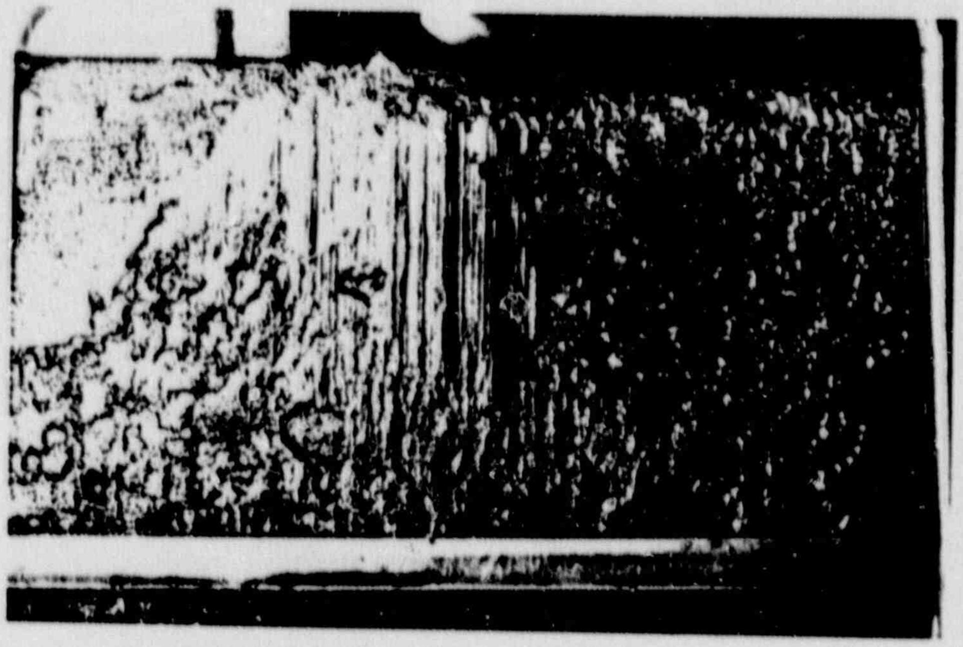


Figure A-3. Closeup photo of debris on the non-thrust side of 3R piston.

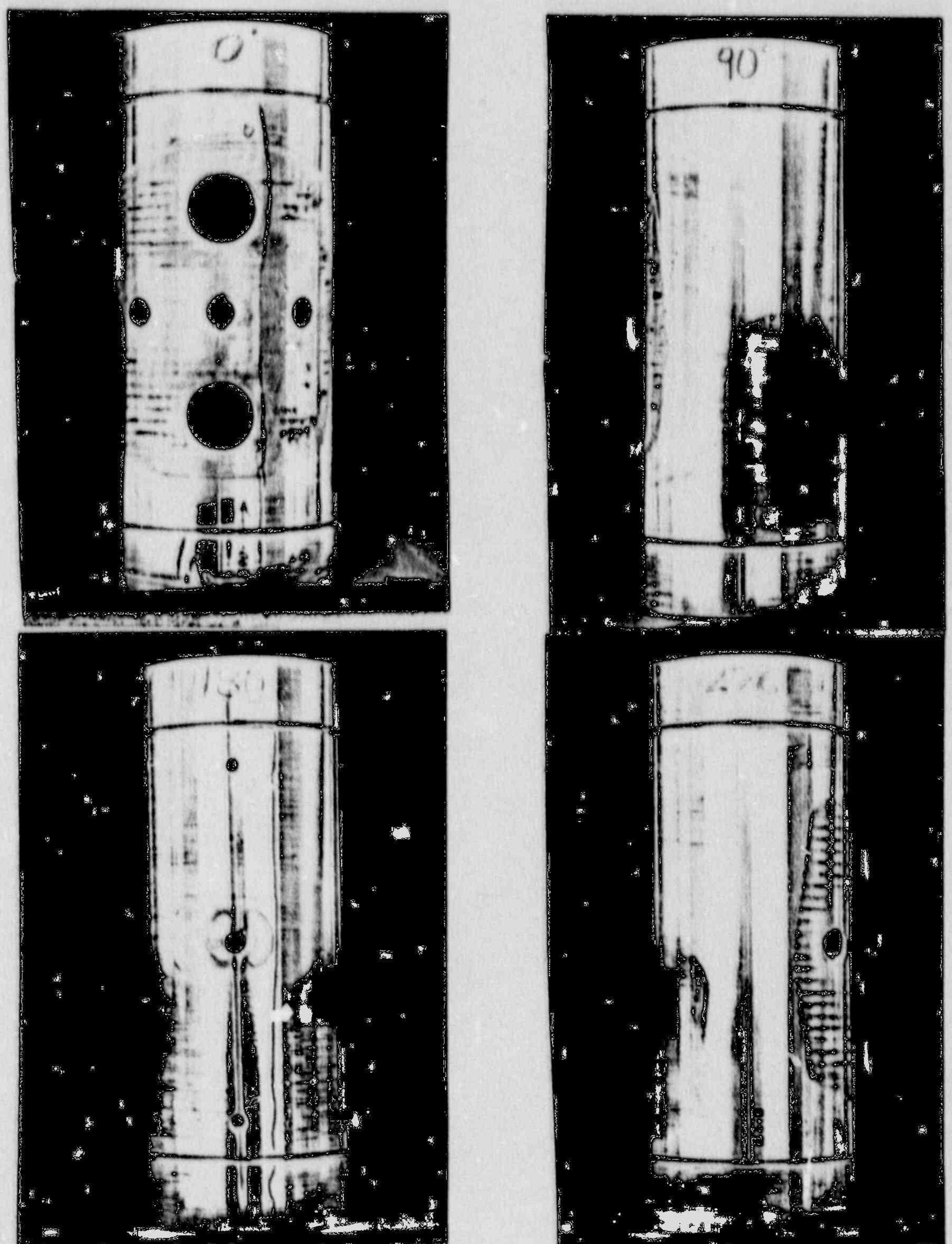


Figure A-4. Piston pin from the 1R piston. The largest amount of discoloration is along each of the axes 90° to either side of the bottom of the pin.

Figure A-5. Photos of deteriorated head gaskets found after removing head on cylinder 3R. All the binder in the gasket had disintegrated leaving the metal windings.

4.6 Deposit Analysis from 8R Piston - "D" Diesel

4.6.1 The attached deposit analysis was obtained from the side of the top of the "B" diesel 7L piston head after the failure. The elemental composition shows that the solids contain Zn, Ca, Mg, Sn, P, S and Fe as the major constituents. From EDS measurements made from various deposits taken from the failed components and displayed in various sections of this report, it is noted that other elements were found occasionally. These include Ti, Si, Al, Pb, Mn, Ni and Cu.

Si and Mn are elements found in cast iron in high enough proportions that these could show up in wear debris anywhere in the engine. Pb is in Pb-Sn solders used occasionally to repair holes in castings per C-B specifications, so that may be the source of this element. Cu is found in the bronze bushing around the pin, so this element may be from wear of this part.

Ni and Al are elements whose origin we cannot rationalize at this time.

DEPOSIT ANALYSIS

SUSQUEHANNA SES DIESEL GENERATOR

LAB NUMBER	80476- 0
DATE RECEIVED	11/17/89
DATE TAKEN	11/14/89
LOCATION	B ~~D~~ D/G DJM 1/4/90
ORIGINATOR ID	89-031 M
SAMPLE DESCRIP	BLACK SOFT CHUNKS, DEGREASED AND DRIED AT 103C PRIOR TO ANALYSIS. THE DEPOSIT WAS SLIGHTLY, ABOUT 10%, MAGNETIC. THE DEPOSIT TURNED TAN IN COLOR AFTER BEING PLACED IN THE FURNACE AT 900C.

LEGEND FOR CD
ND = NOT DETECTED
NR = NOT RUN
< = LESS THAN

ANALYTICAL PROCEDURE	UNITS	CD	RESULT
2327 IGN CHG @ 450C	%		-72.60
2328 IGN CHG 450-900C	%		-10.34
2329 ACID INSOLUBLE	%		0.10
2304 FE2O3	%		0.75
2305 CUO	%		0.03
2306 NIO	%		0.01
2308 ZNO	%		4.76
2307 MNO2	%		0.02
2300 CAO	%		3.40
2301 MGO	%		2.34
2311 AL2O3	%	<	0.02
2309 CR2O3	%		0.01
2302 NA2O	%		0.05
2303 K2O	%		0.03
2315 SNO	%	<	2.0
2310 PBO	%		0.01
2312 V2O5	%	<	0.2
2318 SULFATE AS SO3	%		0.84
2317 PHOSPHATE, P2O5	%		3.99
2743 TOTAL FOUND	%		99.28

DEPOSIT COMMENTS THE DEPOSIT WAS COLLECTED FROM ABOVE THE TOP RING OF THE 'D' DIESEL GENERATOR. THE DEPOSIT APPEARS TO BE INDICATIVE OF UNBURNED OIL AND ITS ADDITIVE PACKAGE.

5.0 Acknowledgements

In addition to the personnel who have signed this report, many PP&L individuals have contributed significantly in the development of this report. When contacted to provide their skills, all of these individuals enthusiastically supported this effort and gave their best, often at the expense of their own established priorities. Their spirit made this effort a pleasure to be involved in. Special thanks to the Hazleton Chemical Laboratory. The whole lab was significantly impacted by this effort, and we appreciate the patience of those whose work was displaced to support our efforts. In addition, no report of this nature could be prepared without knowledgeable input from the mechanics who work on these engines. It is the author's collective opinion that we have an extremely knowledgeable and dedicated maintenance crew working on these engines.

Because of the numbers of people involved, we may have missed acknowledging someone, we apologize.

J. L. Adams, Maintenance
V. F. Beurhing, Hazleton Chemical Lab
C. A. Boschetti, NF&SE Systems Engineering
L. E. Crivellaro, NPE
D. H. Cassel, NPE Support
S. J. Daderko, NS Planning
E. W. Elyard, et al, NQA
D. L. Fetter, Reprographics
D. Flame, Maintenance
D. J. Gandenberger, Licensing
W. J. Gulliver, NQA
B. J. Heacock, NPE-Resident
M. Heidorn, Technical
D. Heim, Maintenance
F. Huff, Hazleton Chemical Lab
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S. Mantz, Maintenance
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F. A. Nederhand, NF&SE
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T. Pensock, Hazleton Chemical Lab
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R. Sadison, et al, Maintenance
A. Sanders, NF&SE Systems Engineering
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R. A. Schmitt, NPE
A. Snyder, Hazleton Chemical Lab
P. Super, Hazleton Chemical Lab
D. J. Walters, Licensing
T. J. Wales, NPE-Nuclear Design