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ENCLOSURE 3

FaAA83-10-16
PA07395

EMERGENCY DIESEL GENERATOR CONNECTING ROD BEARING FAILURE INVESTIGATION SHOREHAM NUCLEAR POWER STATION

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1.0 SUMMARY AND CONCLUSIONS

Following the fracture of the crankshaft in one of the Shoreham Nuclear Power Station (SNPS) emergency diesel generator and the observation of cracks in the other two crankshafts, the three Transamerica Delaval Inc. (TDI) R-48 Enterprise engines were disassembled, and all twenty-four connecting rod bearings were inspected. Four upper bearing shells were found to be cracked through the thickness. These bearings had been operated between 600 and 800 hours, compared to an expected life of 20,000 hours for diesel engines in this category.

The factors which contributed to or caused the bearing cracking have been identified. They are unsupported, overhung bearing ends, excessive crankpin journal yawing, and the presence of voids or pores in the size range of 0.5 mm to 0.7 mm in the aluminum alloy bearings. Scanning electron microscopy of the fracture surface of one of the cracked bearings identified these voids as the apparent crack initiation sites.

Mechanical testing of ten specimens from the cracked bearing demonstrated that this bearing material did not meet the TDI material specifications apparently in effect at the time Shoreham's DG's were designed and fabricated. TDI allegedly lowered these specification requirements subsequent to the delivery of the SNPS DG's and the test results meet this reduced specification. The specification requirements did not (nor do they now) include a porosity requirement.

The replacement connecting rod bearings being installed with the new 12 inch journal crankshafts are represented to be qualified to the lower specification, and are therefore equivalent in material quality to the earlier bearings.

FaAA's calculated design loads on the bearings and the estimated actual service applied loads due to specific design features of these engines with 13 × 11 crankshafts are higher than those recommended by Imperial Clevite Inc., a major independent manufacturer of engine bearings, for this bearing material. Therefore, there is little or no margin left to accomodate material that contains voids of up to 0.5 mm in diameter as did the subject bearings.

The bearing shell is loaded by the hydrodynamic pressure generated in the oil film, which causes stresses in the wall of the shell. The calculated peak oil film pressure for the cracked bearing is 30,000 psi, compared to a recommended maximum of 26,000 psi. The actual bearing stress is further magnified by torque-induced journal yawing and by unsupported bearing ends.

Design improvements have been made by changing crankshafts, connecting rods, and journals, in particular, the pin diameter has been increased from 11 inches to 12 inches and the mechanical configuration of the connecting rods has been improved. These changes will result in a longer minimum life for the new bearings installed with 13 × 12 crankshafts. Quantitative calculations and measurments are being performed to determine a conservative estimate of the bearing life in the new configuration. In order to address this limited bearing life potential, a scheduled program of bearing replacement and NDE for void detection may be required.

2.0 INTRODUCTION AND BACKGROUND

The Transamerica Delaval Incorporated (TDI) Enterprise diesel engines at Shoreham Nuclear Power Station (SNPS) are equipped with hydrodynamically-lubricated connecting rod bearing shells made from solid aluminum - 6% tin (Alcoa alloy B850), with an inner surface layer of electroplated lead-base babbitt. Figure 1 shows a schematic representation of a connecting rod bearing half-shell and indicates the nomenclature used to describe its features.

During disassembly of the Enterprise diesels to replace the crankshafts, four out of twenty-four connecting rod bearings were observed to be damaged.

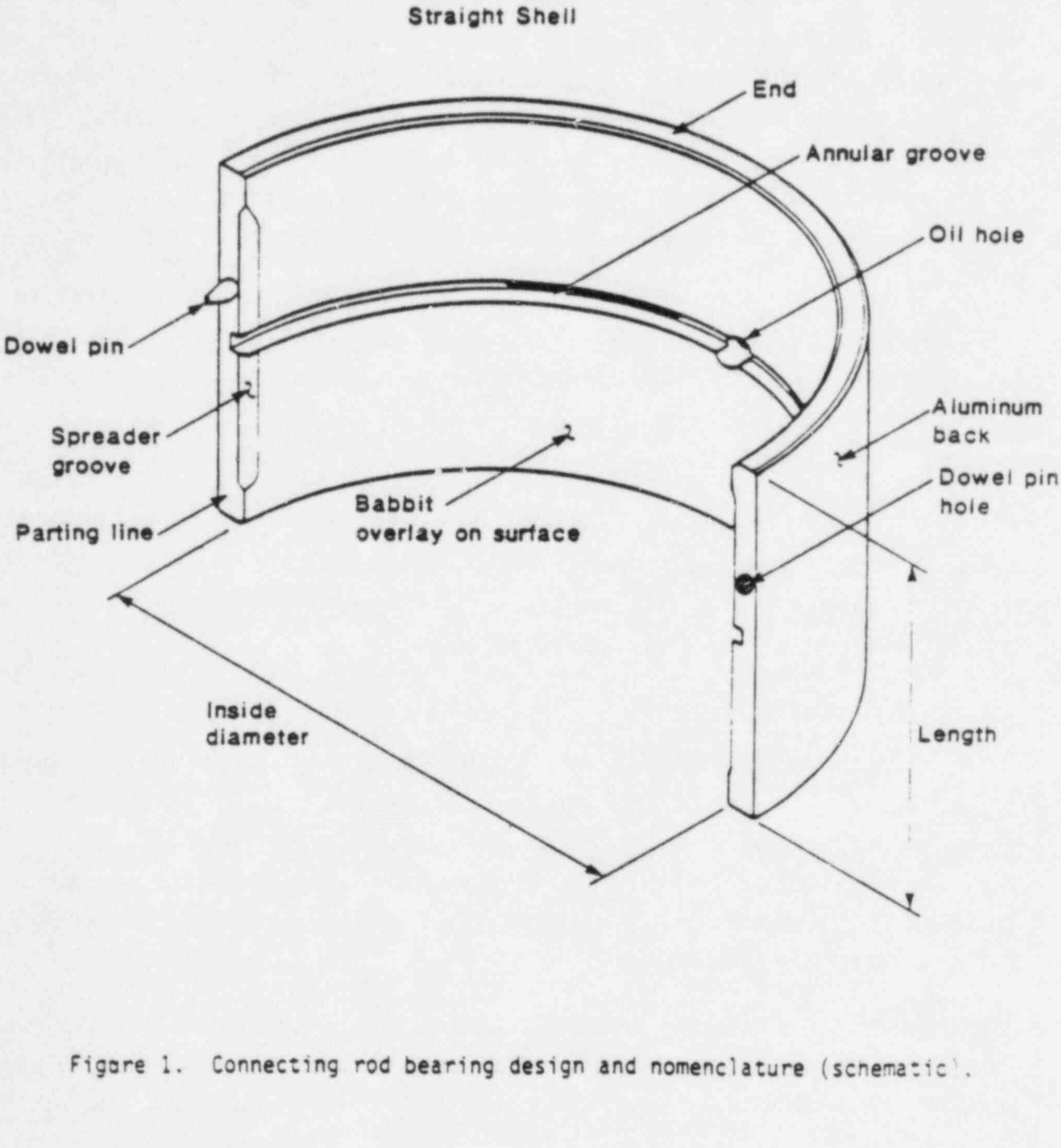


Figure 1. Connecting rod bearing design and nomenclature (schematic).

The most severe damage was suffered by the No. 5 upper bearing (associated with the No. 5 cylinder) of diesel generator set DG103. This bearing is shown in Figure 2. This half-shell was fractured into two separate pieces (the smaller of which is not shown in Figure 2) near one end of the bearing. Three other upper bearings contained cracks through their thicknesses. These cracks were in the same relative location as the fracture surface on the broken bearing, but had not intercepted the bearing end to cause complete fracture. These cracked bearings were in No. 3 and No. 4 in DG103 and No. 4 in DG102.

An investigation was undertaken to determine the cause of the cracking, which had occurred after 600 to 820 hours of engine operation. In diesel engines of this size, the expected connecting rod bearing life would be expected to exceed 20,000 hours.

The investigation included physical and metallurgical examination of one cracked bearing, computation of the design loads imposed on the bearing, and an analysis of design features of the bearing system by direct observation of the components of the disassembled engines.

3.0 METALLURGICAL EXAMINATION NO. 4 UPPER CONNECTING ROD BEARING FROM DG103

As a preliminary step to the detailed metallurgical investigation, all of the connecting rod bearings from DG102 and DG103 were examined visually for overall condition and for possible evidence of unusual operating conditions. The DG103 No. 4 upper connecting rod bearing was removed to FaAA's laboratory for more detailed examination.

Most of the bearings appeared to be in serviceable condition, with the expected polishing of the babbitt overlay occurring in the most highly-loaded areas of the bearing.

The amount of scoring of the bearing surface from circulating solid particles in the lubricant was minimal, indicating that the engines were kept clean internally. There was no evidence of any chemical attack of the babbitt

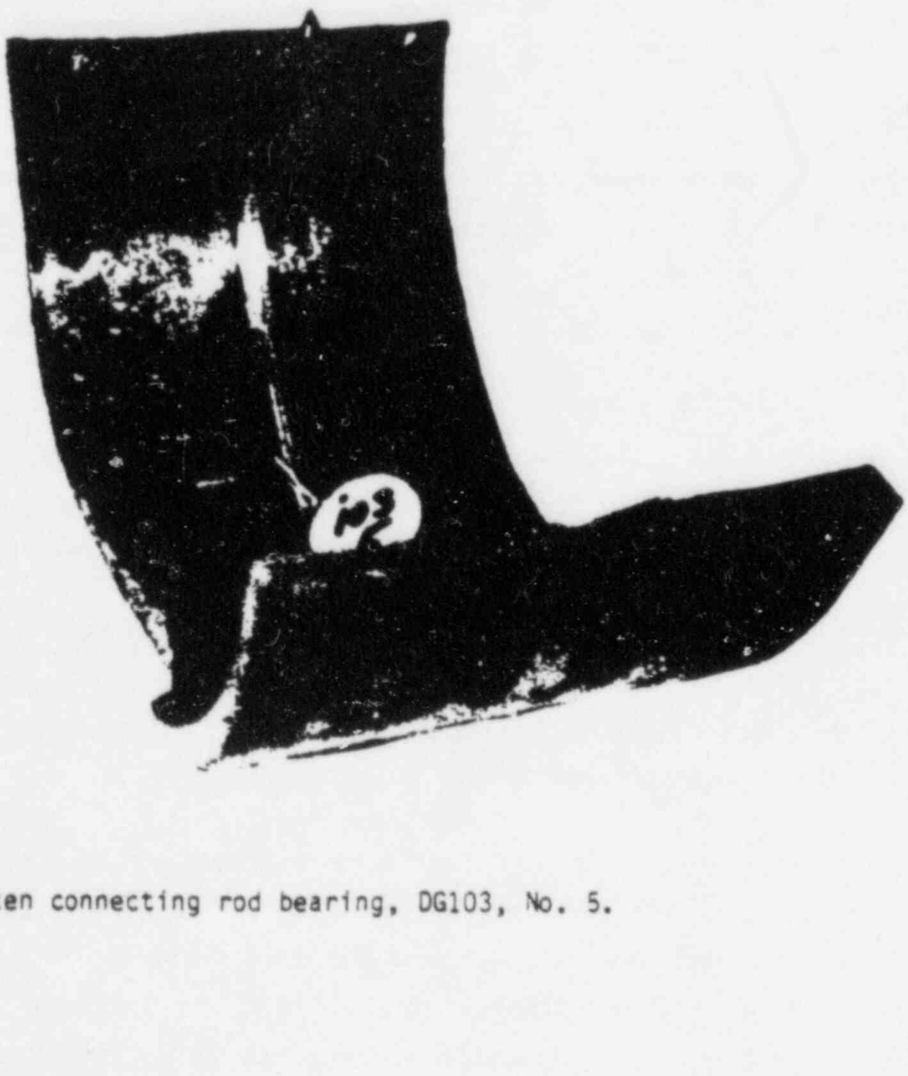


Figure 2. Broken connecting rod bearing, DG103, No. 5.

overlay, indicating that the lubricating oil had remained non-acidic and was essentially uncontaminated by acidic combustion products or by coolant leaks into the oil system.

One feature of note on the majority of the bearings was the shape of the polished region on the babbitt overlay; it was wider at both ends of the bearings, covering almost 90° of arc, than in the middle, where it covered about 45° of arc. This pattern is the result of edge loading, which results when the journal axis is not perfectly parallel with the bearing surface, thereby causing the journal to approach the bearing more closely at the bearing ends.

The contact patterns on the backs of the bearings showed that the ends of the bearings were not supported by the bores of the connecting rods, as a consequence of the large, 1/4 inch chamfers on the bores. Figure 3-A shows a cross-sectional representation of the lack of support of the connecting rod bearing ends.

The DG103 No. 4 upper connecting rod bearing contained a crack approximately four inches long at one end of the bearing. The crack appeared to extend completely through the thickness of the bearing, being visible on both the inner surface and the bearing back.

Two axial cuts through the fracture surface were made from the end of the bearing containing the crack in order to free the major portion of the fracture surface for separation and examination.

The small portion of the bearing separated by the above cuts was examined in a scanning electron microscope (SEM). The SEM examination revealed significant near-surface pores that are the probable initiation sites for cracking. These pores are approximately 0.5 mm to 0.7 mm in diameter. Examples of these pores on the fracture surface are illustrated in Figure 4.

A sample of the subject bearing was submitted to Metallurgical Testing Corporation for chemical analysis. The results of this analysis, along with the chemical specification for alloy B850, are given in Table 1. The results indicated acceptable chemical properties.

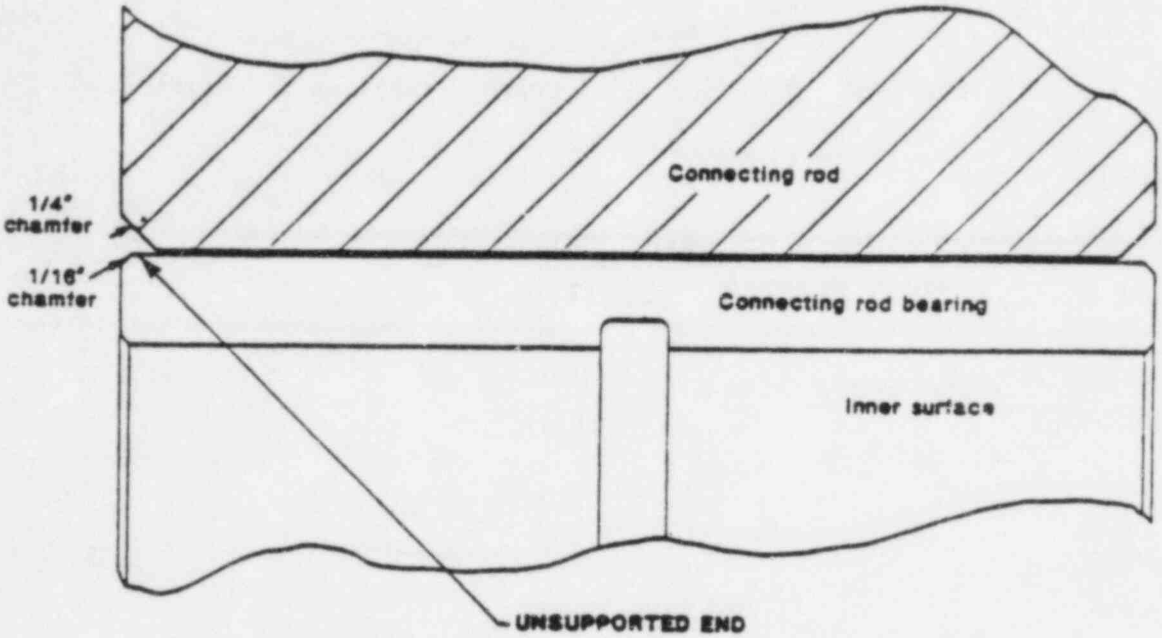
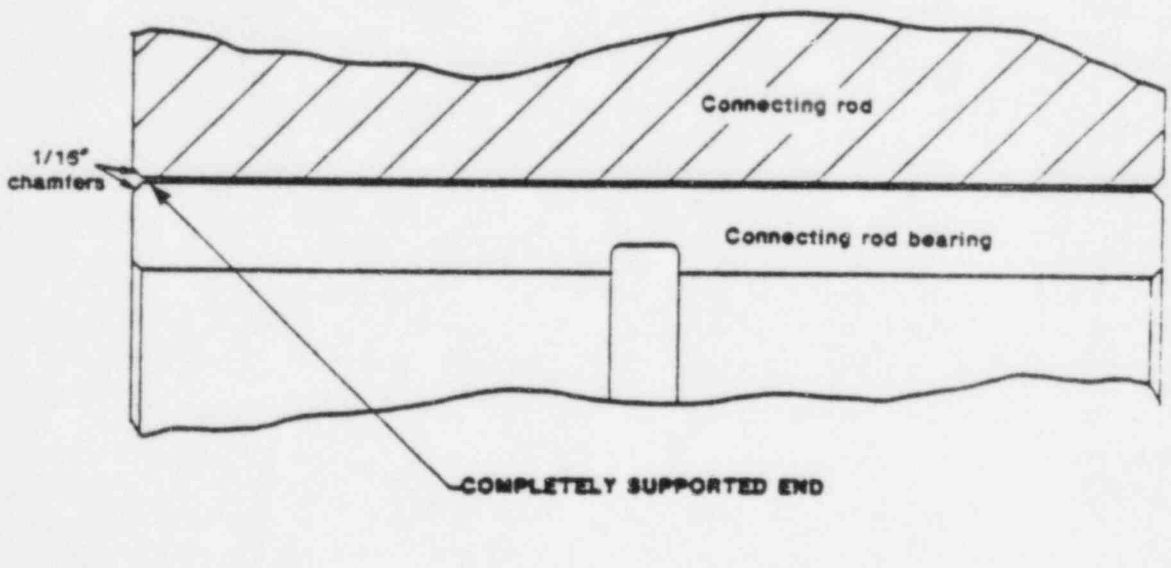


Figure 3a. Bearing: Connecting rod configuration with original 11" journals.



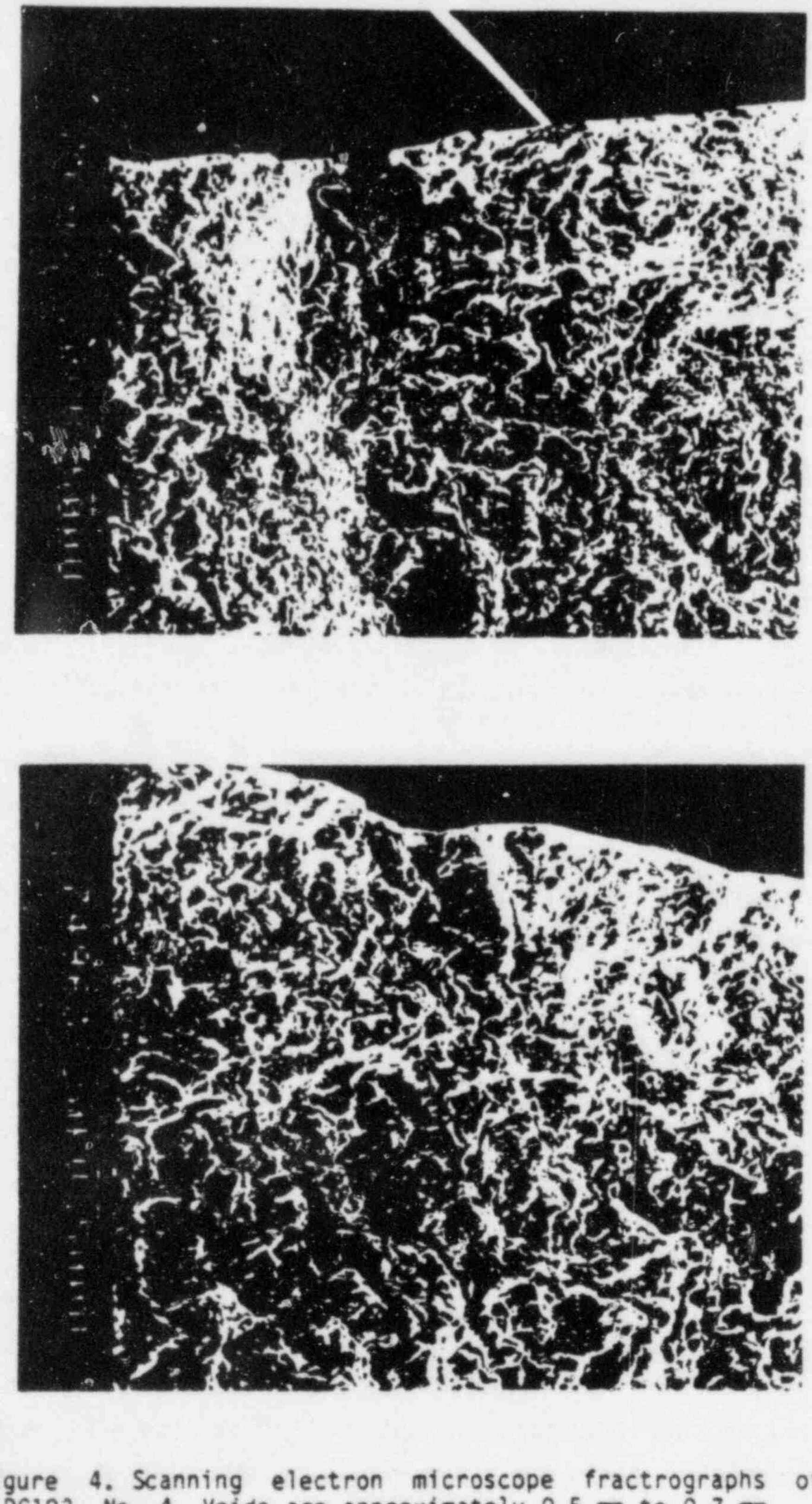
NOTE: Drawing at .74 of original size.

Figure 3b. Bearing: Connecting rod configuration with replacement 12" journals.

Table 1

Chemistry of DG103, No. 4 Upper Connecting Rod Bearing

B850	Nominal Composition (%)	Results of Analyses (%)
Al	90.0	balance
Sn	6.0	5.26
Cu	1.0	1.86
Ni	2.0	1.38
Mg	1.0	.77
Fe	--	.36
Si	--	.25
Ti	--	.12



26X

36X

Figure 4. Scanning electron microscope fractrographs of DG103, No. 4 Voids are approximately 0.5 mm to 0.7 mm.

Tension test specimens were cut from the end of the subject bearing containing the crack between each parting line and the fracture surface. The specimens were $^1/_4$ inch gage diameter, 1 inch gage length per ASTM B-557-81 [4], the largest that could be obtained from the finished bearing, and they were oriented parallel to the axis of the bearing and perpendicular to the plane of the fracture.

Ten specimens were prepared and tested according to ASTM standards. The results are listed in Table 2. Ultimate tensile strength ranged from 23.7 ksi to 28.1 ksi with elongations ranging from 0.40% to 0.88%. Only one of the ten test specimens met the apparent original design requirement [3] for tensile strength and none met the elongation requirement. [3] When compared with TDI's alleged current specification requirements [7], all ten samples met the tensile strength criterion, but again, none met the elongation requirement.

The samples were the largest that could be taken from finished bearings, but were one-half the size of samples that would be taken from unfinished castings for Quality Assurance. ASTM Standard B-557-81 states that elongation values obtained from smaller specimens may not equal those obtained from larger specimens.

The microstructure of the subject bearing was examined for the presence of anomalies. As is normal for this material, the tin was concentrated between grains of aluminum. As noted, porosity in the range of 0.5 mm to 0.7 mm was visible, but TDI apparently has no microstructure standard [7] for the acceptable degree of porosity.

Examinations of the three other cracked bearings similar to the metallurgical analysis of the DG103 No. 4 upper connecting rod bearing is underway by performing similar mechanical properties and scanning electron microscopy. Additionally, bearings that did not fail in service will be examined to assist in identifying the quantitative effect of materials properties.

Table 2

Tension Test Results for DG103, No. 4 Upper Connecting Rod Bearing Shell

Test No.	U.T.S. (ksi)	Elongation (percent)
1	25.7	0.80
2	23.7	0.40
3	25.2	0.70
4	25.7	0.76
5	26.5	0.76
6	26.1	0.56
7	26.7	0.72
8	26.9	0.54
9	28.1	0.88
10	26.1	0.68
Specification (1976) [3]	27.0	2.00
Specification (1983) [7]	23.0	2.00

Note: Results are from $^{1}/_{4}$ inch diameter test specimens. Specifications are for $^{1}/_{2}$ diameter test specimens. The smaller test specimens could result in somewhat conservative elongation results, but the tensile strength results are unaffected by this difference in size.

4.0 CONNECTING ROD BEARING DESIGN LOAD CALCULATIONS

A major, independent manufacturer of sleeve bearings (Imperial Clevite Inc.) was engaged to compute the loading on the connecting rod bearings. Journal orbit analysis [1] was employed to determine the thickness and pressure of the hydrodynamic oil film. Data supplied to Imperial Clevite for the computation included the relevant engine and bearing dimensions and design features; operating parameters such as engine speed, power output, mechanical efficiency, lube oil temperature, pressure and viscosity, and peak cylinder pressure.

Imperial Clevite reported that the peak oil film pressure in the 11 inch connecting rod bearings was predicted to be 30,000 psi. A summary of results of the journal orbit analysis for both 11 inch and 12 inch bearings is presented in Table 3. The significance of the journal orbit analysis results is addressed in Section 7.0.

Analysis of the crankshafts for the TDI Enterprise diesel engines by FaAA has predicted a dynamic yawing or pitching of the crank pin journals resulting from the transmission of torque across the journals. For the 11 inch crank pin, the total range of crank pin deflection, end-to-end, is 0.0064 inch at the No. 6 journal. The 12 inch crank pins will exhibit less deflection as a consequence of lower peak torques and higher crankshaft stiffness. The total range of crank pin deflection for the 12 inch diameter No. 6 journal is 0.0039 inch.

The magnitude of the journal deflection contributes to the local stress on the connecting rod bearing. As deflection increases, the total bearing load is transferred toward one end of the bearing, increasing the localized stresses at that end. The quantitative influence of journal deflection on bearing stresses is being determined via Finite Element Analysis.

Table 3

Journal Orbit Analysis of Connecting Rod Bearings [5]

Input Data - TDI Enterprise Diesel Engine

Brake Horsepower	4,880 b.h.p.
Cylinders	8
Bore	17.000 inches
Stroke	21.000 inches
Compression Ratio	11.57:1
Connecting Rod C/L-C/L Length	46.125 inches
Reciprocating Weight	799.4 pounds
Rotating Weight	432.3 pounds
Shaft Diameter	11.000 inches/12.000 inches
Radial Clearance	.0045 inch
Effective Length	3.1885 (x2) inches
Grooving	360°
Oil Viscosity	3.2313 mreyns
Oil Pressure	55 psig
Oil Temperature	165°F
Peak Pressure	1,680 psig
Mechanical Efficiency	88%
Operating Cycle	4-stroke

Output - 11-inch Journal

Maximum Oil Film Pressure	29,745 psi
At Bearing Angle	2 degrees

Output - 12-inch Journal

Maximum Oil Film Pressure	26,780 psi
At Bearing Angle	2 degrees

Imperial Clevite Inc. Recommended Bearing Loads, Solid (Wrought) Al-6% Sn [6]

Maximum Oil Film Pressure, Stationary Diesel Engines, Intermittant Service	26000 psi

5.0 CHANGES IN CONNECTING ROD BEARINGS ASSOCIATED WITH REPLACEMENT CRANKSHAFTS

Replacement of the original 11 inch crank pin journal crankshafts with new 12 inch crank pin journal crankshafts requires the installation of new connecting rods and rod bearings sized to accommodate the 12 inch crank pin. The expected effects of these changes on connecting rod bearing performance are as follows:

1. The new, stiffer crankshafts will result in less crank pin journal yawing deflection. As mentioned in Section 4, the expected reduction in yawing is about 35 percent. This will reduce the concentration of the bearing load on the bearing ends (reduced edge-loading) and therefore, reduce stresses in the connecting rod bearings.

2. The larger-diameter journal will reduce calculated maximum oil film pressure. Table 3 has the summary of journal orbit analysis results, showing the maximum oil film pressure for the new configuration to be 25,800 psi, a reduction of 10% compared to the original configuration. This will result in a direct 10 percent reduction in bearing stress.

3. The new connecting rods have a small, $^{1}/_{16}$ inch bore chamfer rather than the original $^{1}/_{4}$ inch bore chamfer. As Figure 3-B demonstrates by comparion with Figure 3-A, the cantilevered or overhung configuration of the original bearing is eliminated in the newer design.

6.0 QUANTIFICATION OF EFFECTS

Work is ongoing to quantify the effect of overhung bearings, journal yawing, and 0.5-0.7 mm porosity on the life of the bearings. The purpose for using Finite Element Analysis and Fracture Mechanics is to be able to calculate reasonable, conservative recommendations for inspection and/or replacement of connecting rod bearings on a scheduled basis. The cracking observed in the connecting rod bearings is due to the three causes described; the quantification of effects will just establish their relative importance. As detailed in Section 5.0, the changes in design associated with the installation of new crankshafts, connecting rods and connecting rod bearings directly address two of the causes of cracking. Computation of threshold void sizes for fatigue and N.D.E. to select bearings which do not have voids of threshold size will eliminate the third cause of cracking in the reassembled engines.

7.0 DISCUSSION

In FaAA's opinion, the primary cause of cracking in the DG102 and DG103 connecting rod bearings is loading imposed on the bearings in excess of the design capabilities of the original bearings. This, combined with the presence of approximately 0.5 mm to 0.7 mm voids in the cast aluminum bearing material and a mechanical design feature of the connecting rods resulting in unsupported bearing ends, caused the observed cracking.

The high design loads arise from a combination of factors. The calculated peak oil film pressures are above levels recommended by Imperial Clevite, [6] a major independent supplier of sleeve bearings for diesel engines; these loads are locally increased by edge-loading due to crank pin journal yawing, and by cantilevered loading due to unsupported bearing ends.

The calculated peak oil film pressure for the 11 inch journal, 29,745 psi, exceeds Imperial Clevite's recommendation for the allowable peak-oil-film-pressure (26,000 psi) for this type of bearing. [6] Imperial Clevite arrived at this recommendation based upon comparing calculated peak oil film pressures for engines with their actual operating experience. Therefore, FaAA concludes that the original 11 inch connecting rod bearings were not adeqate for use in these engines. SAE Paper 830062 [2] explains the derivation of the recommendations in detail.

Clevite's recommended peak oil pressure applies to solid aluminum - 6% tin wrought bearings or cast bearings without the voids that initiated cracking in the subject bearing.

The material in the cracked bearing exceeds the tensile strength minimum allegedly currently specified by TDI and Alcoa [7], although it does not meet the TDI/ALCOA specification allegedly in effect [3] at the time the SNPS DG's were procured. The results of FaAA testing for ductility did not meet the alleged specified minimum, but this may be due to the use of sub-size ($^1/_4$ inch diameter) tensile specimens. Alcoa specifies that $^1/_2$ inch diameter tensile specimens should be used to determine ductility, and ASTM Method B-557-81 requires that ductility be determined on full-size specimens. Given the configuration of the bearings, full-size specimens could not be obtained.

Scanning electron microscopy of the fracture surface revealed at least four voids in the cast material in the fracture plane that were in the size range 0.5 mm to 0.7 mm. One or more of these voids appears to have been an intiation site for the crack.

The results of the testing and the observed performance of this one bearing demonstrate that meeting the apparent TDI/ALCOA established mechanical properties minimums may not be sufficient to ensure the acceptability of connecting rod bearings in this case. The presence of 0.5 mm to 0.7 mm pores appears to be sufficient to initiate cracks that grow to sizes of concern in 600 hours in the original configuration.

Since all of the changes to the new 13 × 12 connecting rod bearings represent improvements to the bearing stress situation, and since the old 13 × 11 connecting rod bearings did perform adequately for 600 + hours (although exhibiting local damage at that point), it is reasonable to conclude that the new bearings will perform reliably in their required service if our recommendations for periodic replacement are adhered to or, alternatively, if detailed finite element calculations confirm that the new 13 × 12 bearings can be predicted to be adequate even with the void sizes experienced. These recommendations will be determined by detailed stress analysis of the bearings in both the original and the replacement configuration, and by a fracture mechanics and N.D.E. approach to identifying and controlling the maximum acceptable void size in the material.

References

1. Ross, J.M. and R. R. Slaymaker, "Journal Center Orbits in Piston Engine Bearings," SAE Paper 690114, Society of Automotive Engineers, Warrendale, Pennsylvania 1969.

2. Hollander, M. and K. A. Bryda, "Interpretation of Engine Bearing Performance by Journal Orbit Analysis," SAE Paper 830062, Society of Automotive Engineers, Warrendale, Pennsylvania 1983.

3. Aluminum Company of America, Alcoa Aluminum Design Data, Pittsburgh, Pennsylvania 1977.

4. ASTM Standard B-557, "Tension Testing Wrought and Cast Aluminum - and Magnesium Alloy Products," ASTM.

5. Journal Orbit Analyses of TDI Enterprise R-48 Diesel Engine performed by Imperial Clevite Inc. for FaAA, October 6, 1983.

6. W. A. Yahraus (Manager of Product Analysis, Imperial Clevite Inc., Engine Parts Division), private communication with L. A. Swanger (FaAA), October 4, 1983.

7. C. Mathews, G. King (Transamerica Delaval In.., Engine and Compressor Division), private communication with L. A. Swanger (FaAA), October 4, 1983.

8. R. Ewing (Manager of Engineering, Heavy Bearings, Imperial Clevite Inc. Engine Parts Division), Private communication with L. A. Swanger (FaAA), November 2, 1983.

ENCLOSURE 4

STANDBY D/G RELIABILITY

TOPICS FOR DISCUSSION

1. ORIGINAL DESIGN AND PROCUREMENT
2. QA AND INSPECTION ACTIVITIES
3. CODES AND STANDARDS USED
4. OPERATING HISTORY/RESULTS
5. PROBLEMS ENCOUNTERED
6. SRC RECOMMENDATIONS AND MANAGEMENT ACTION
7. TIE - D/G OWNERS GROUP
8. OTHERS CONSIDERATIONS
9. CRANKSHAFT - COMPARISON TO SHOREHAM DESIGN
10. CONCLUSION

EMERGENCY DIESEL GENERAL SPECIFICATION HISTORY

SPECIFICATION REQUIREMENTS DEFINED IN REVISION 2 AND ISSUED FOR PURCHASE IN JULY 1974.

- DESIGN RATING: 7000 KW, 450 RPM, 4160 V

  NOTE: 7000 KW CAPACITY PROVIDES 21% MARGIN ABOVE MAXIMUM BUS LOAD OF 5800 KW.

- NO EXCEPTIONS TO PROTOTYPE TESTING REQUIREMENT OF 300 STARTS WITH ONE FAILURE PER 100 STARTS PERMISSIBLE.

SUBSEQUENT MINOR REVISIONS INCLUDE:

- UPGRADED FUEL OIL SYSTEM TO ASME SECTION III
- EXPANDED NDE SCOPE
- REVISED AND ADDED TECHNICAL DETAILS:

  1) LUBE OIL TRIP LOGIC
  2) TEMP. MONITORING

- CLARIFIED AND DEFINED:

  1) VENDOR SCOPE
  2) INTERFACE WITH MP&L
  3) SEISMIC REQUIREMENTS
  4) TESTING REQUIREMENTS
  5) WELDING REQUIREMENTS

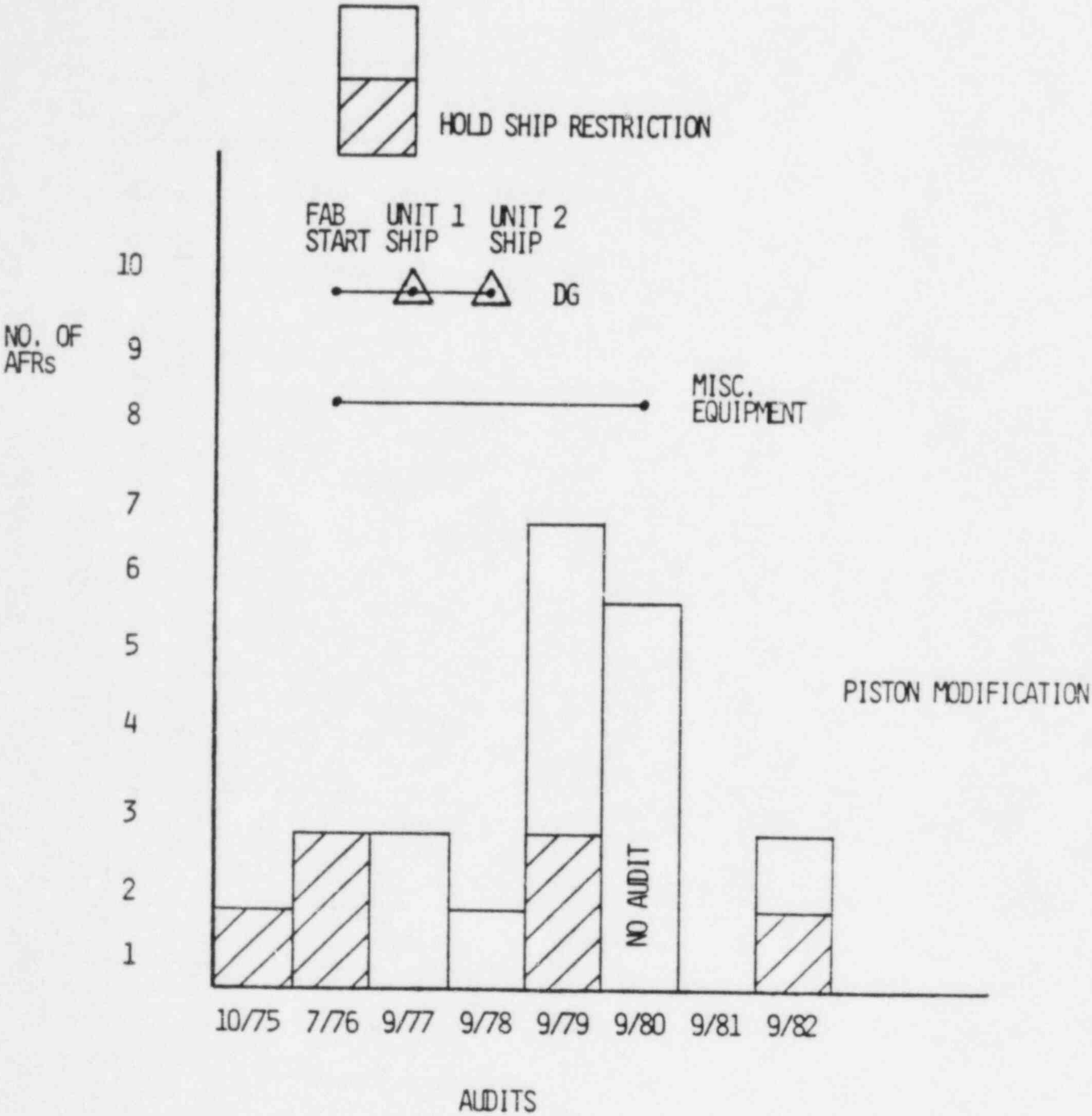
SINCE JULY 1974, <u>NO</u> <u>MAJOR</u> <u>DESIGN</u> <u>CHANGES</u>

QUALITY SURVEILLANCE

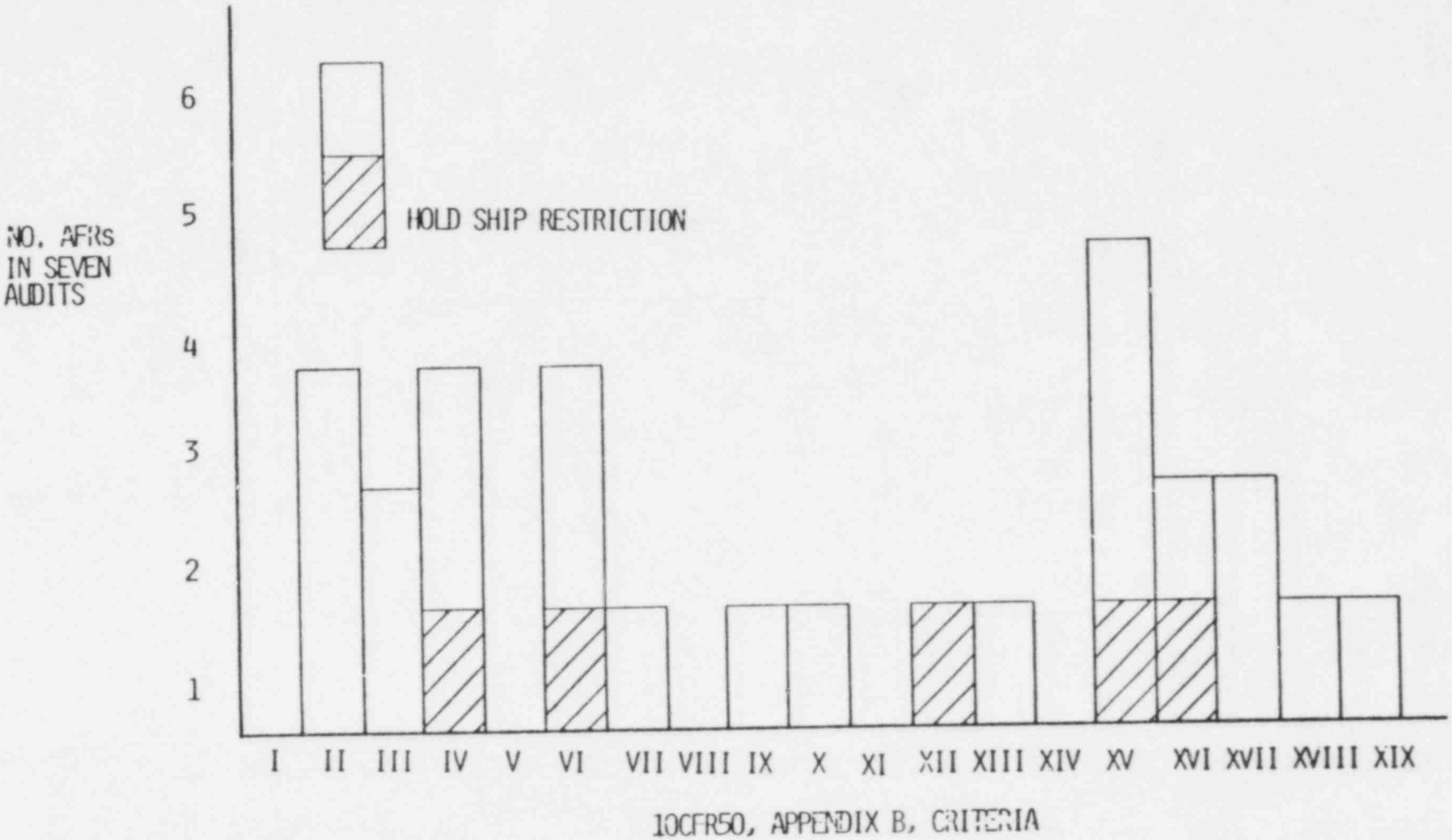
THE SELECTIVE REVIEW, OBSERVATION AND EVALUATION OF: PROCESSES, PROCUREMENTS, MANUFACTURING OPERATIONS, QUALITY CONTROL SYSTEMS, AND PROGRAMS, TO DETERMINE SUPPLIER COMPLIANCE WITH CONTRACTUAL REQUIREMENTS.

DOCUMENTATION REQUIRED FROM TDI

- ENGINEERING DRAWINGS, DIAGRAMS AND DATA SHEETS
- MANUFACTURING/QA/QC PROCEDURES AND REPORTS
- WELDING/NDE PROCEDURES AND REPORTS
- TESTING PROCEDURES AND REPORTS
- INSTALLATION DRAWINGS AND INSTRUCTIONS


HOLD SHIP RESTRICTION
FAB START
UNIT 1 SHIP
UNIT 2 SHIP
DG
MISC. EQUIPMENT
PISTON MODIFICATION
NO. OF AFRs
10
9
8
7
6
5
4
3
2
1
NO AUDIT
10/75
7/76
9/77
9/78
9/79
9/80
9/81
9/82
AUDITS

DISTRIBUTION OF AFRs VS 10CFR50, APPENDIX B



SUPPLIER SURVEILLANCE STATISTICS

	TDI (DIESEL)	PORTEC (GENERATOR)	RTE-DELTA (CONTROL PANELS)	TOTAL
REPORTS	112	8	24	114
DAYS IN SHOP	279	14	19	312
SHIPMENTS	47	3	2	52
DEFICIENCIES	33	1	40	74
WITNESS AND HOLD POINTS	124	4	6	134
IN-PROCESS INSPECTIONS	53	4	3	62

QUALIFICATION TESTING

	TEST[1]	SPECIFIED[2]	PERFORMED	WITNESSED[3]
1)	FUNCTIONAL (COMPONENTS)	X	X	X
2)	OPERATIONAL (SYSTEM)	X	X	X
3)	ELECTRICAL	X	X	X
4)	STARTING AIR COMPRESSOR CAPACITY	TDI[4]	X	X
5)	300 START	X	X	X
6)	SEQUENTIAL LOAD	X	X	X
7)	LOAD REJECTION	X	X	X
8)	MARGIN	X	X	X
9)	ENDURANCE	X	X	X
10)	ACOUSTICAL	TDI[4]	X	NO
11)	CRANKSHAFT TORSION (TORSIOGRAPH)	TDI[4]	X	NO
12)	STARTING AIR BOTTLE CAPACITY	TDI[4]	X	X
13)	LOAD CAPABILITY QUALIFICATION	TDI[4]	X	X
14)	IDLE ENDURANCE	TDI[4]	X	NO

NOTES:
1) TEST NOMENCLATURE TAKEN FROM TDI TEST REPORT.
2) AS REQUIRED BY SPECIFICATION 9645-M-018.0, REV. 22
3) AS WITNESSED BY BECHTEL, ACTING AS MP&L'S REPRESENTATIVE.
4) ADDITIONAL TESTS PERFORMED BY TDI TO AUGMENT THOSE REQUIRED BY MP&L AND DEMA.

EMERGENCY DIESEL GENERATOR DESIGN CODES AND STANDARDS

ESSENTIAL COMPONENTS OF FUEL OIL SYSTEM:

ASME BOILER AND PRESSURE VESSEL CODE (1974 EDITION), SECTION III

NONESSENTIAL COMPONENTS OF FUEL OIL SYSTEM AND ALL COMPONENTS OF AUXILIARY SYSTEMS (LUBE OIL, STARTING AIR AND JACKET WATER SYSTEMS) UP TO ENGINE INTERFACE:

ANSI B31.1 POWER PIPING (1973 EDITION AND SUMMER 74 ADDENDUM)

DIESEL ENGINE AND ON ENGINE MOUNTED PIPING AND COMPONENTS:

DEMA STANDARD PRACTICES
ASTM - A106
ASTM - A53
ANSI - B16.5
ANSI - B16.25
ANSI - B16.10
ANSI - B16.11

PRESSURE VESSELS (LUBE OIL COOLER, JACKET WATER COOLER, AND STARTING AIR TANK):

ASME SECTION VIII, DIVISION I
TEMA CLASS R FOR LUBE OIL AND JACKET WATER COOLERS

PUMPS:

HYDRAULIC INSTITUTE STANDARDS

GENERATORS:

NEMA-MG-1-1972, MOTORS AND GENERATORS

ANSI-C-50.10-1965, GENERAL REQUIREMENTS FOR SYNCHRONOUS MACHINES

ANSI-C-50.12-1965, REQUIREMENTS FOR SALIENT POLE GENERATORS AND CONDENSERS

MOTORS:

NEMA-MG-1-1972, MOTORS AND GENERATORS

DIESEL GENERATORS:

IEEE STD 308 - CRITERIA FOR CLASS IE POWER SYSTEMS FOR NUCLEAR GENERATING STATIONS

IEEE STD 323 - STANDARDS FOR QUALIFYING CLASS IE EQUIPMENT FOR NUCLEAR POWER GENERATING STATIONS

IEEE STD 344 - RECOMMENDED PRACTICES FOR SEISMIC QUALIFICATION OF CLASS IE EQUIPMENT FOR NUCLEAR POWER GENERATING STATIONS

IEEE STD 387 - CRITERIA FOR DIESEL GENERATOR UNITS APPLIED AS STANDBY POWER SUPPLIER FOR NUCLEAR POWER GENERATING STATIONS

DIESEL GENERATOR TESTS:

IEEE STD 115 - 1965 - TEST PROCEDURES FOR SYNCHRONOUS MACHINES

ASME PERFORMANCE TEST CODES, PTC-17-1957 AND PTC-26-1962

TABLE 2

GGNS TDI D/G OPERATING DATA[1]

	Division I	Division II
Shop and Pre-Oper. Run Time (Hrs)	535	252
Since Date of OL Run Time (Hrs)	558	108
Total Run Time (Hrs) [3]	1093	360
Total No. of Starts[3]		
DeLaval Shop Runs	310[2]	5
Pre-Operational Runs	60	60
Since Date of OL Runs	112	60
Total Starts	482	125

NOTES:

1. Source of Information - DeLaval Technical Manual
2. Division I engine had 300 prototype runs for reliability testing
3. Data as of October 11, 1983
4. Valid Starts: Div. I - 46

   Div. II - 37

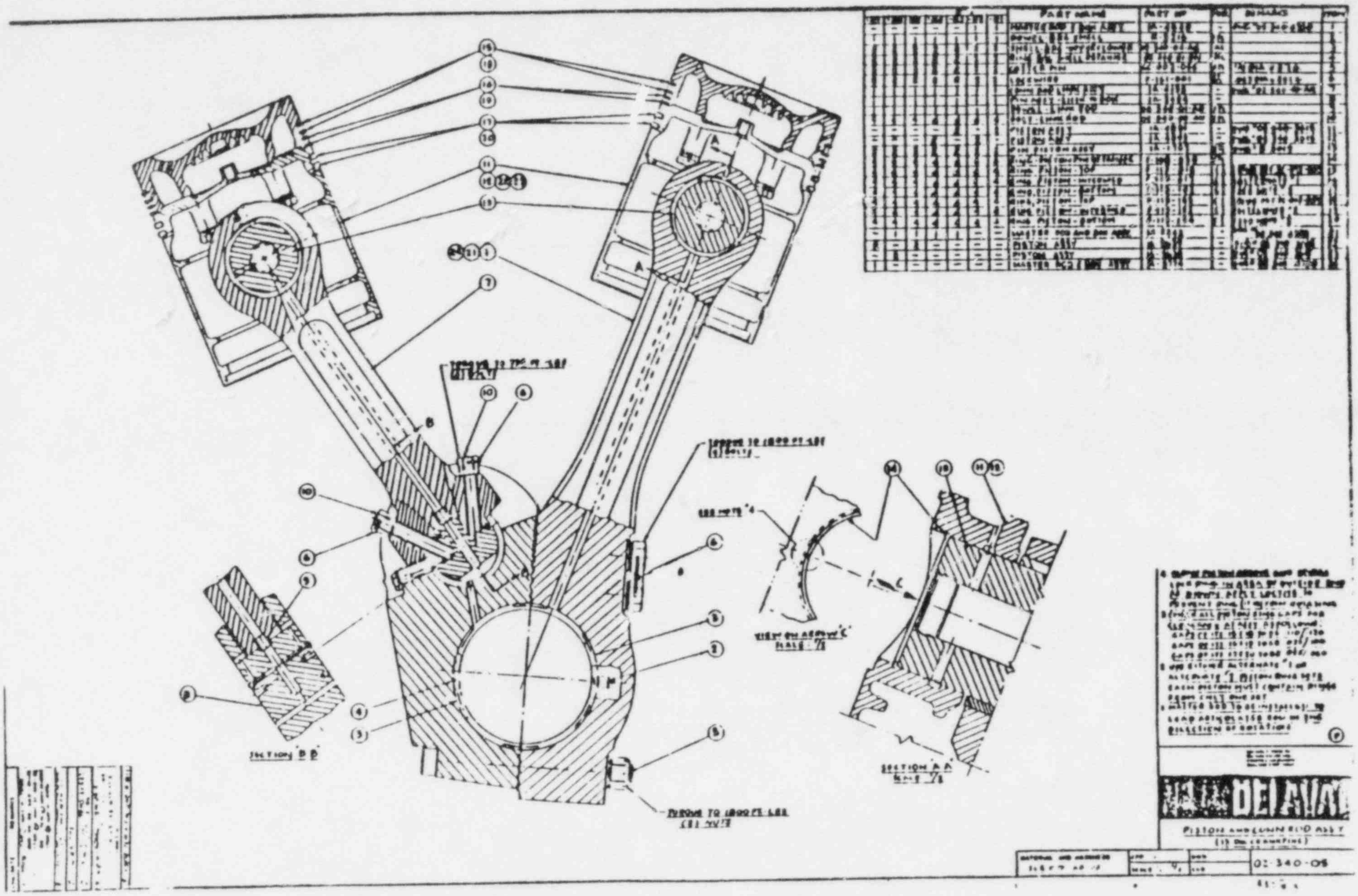
   TOTAL 83

   Valid Failures: 1 (Div. I)

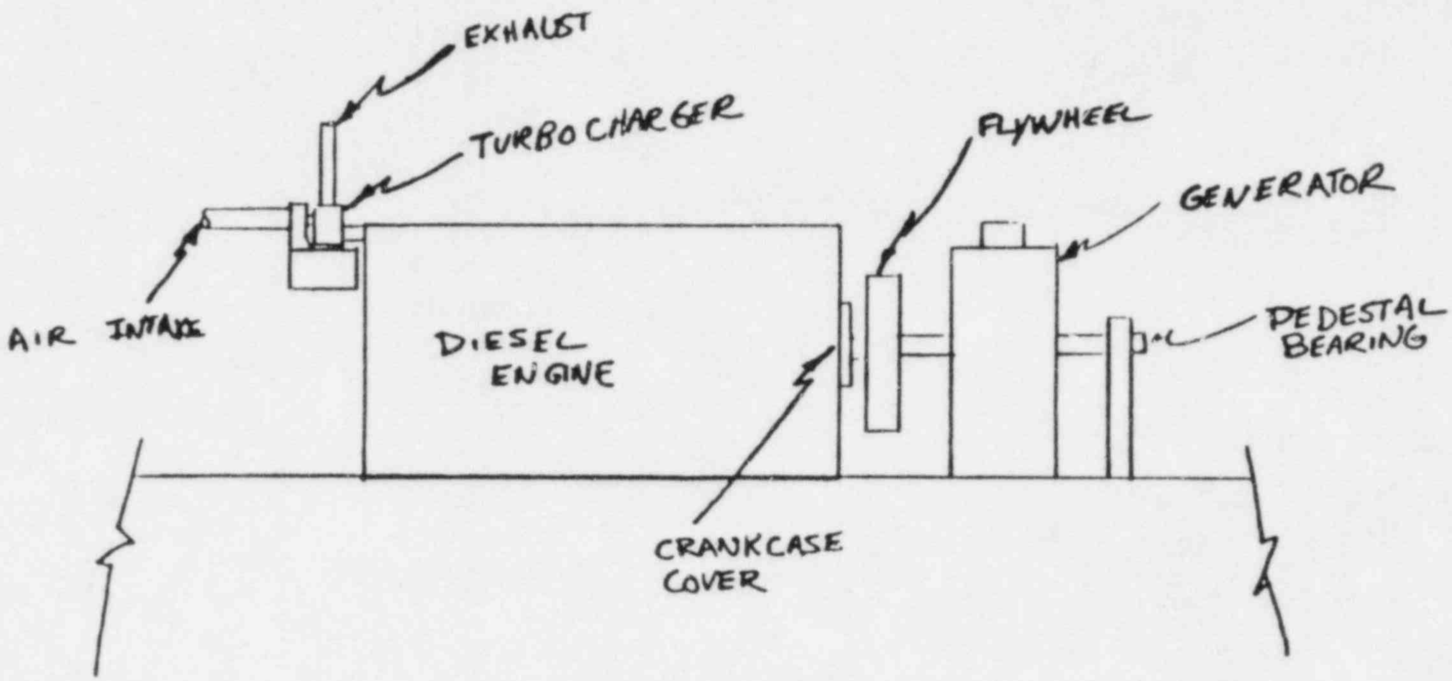
   Start Reliability : .988

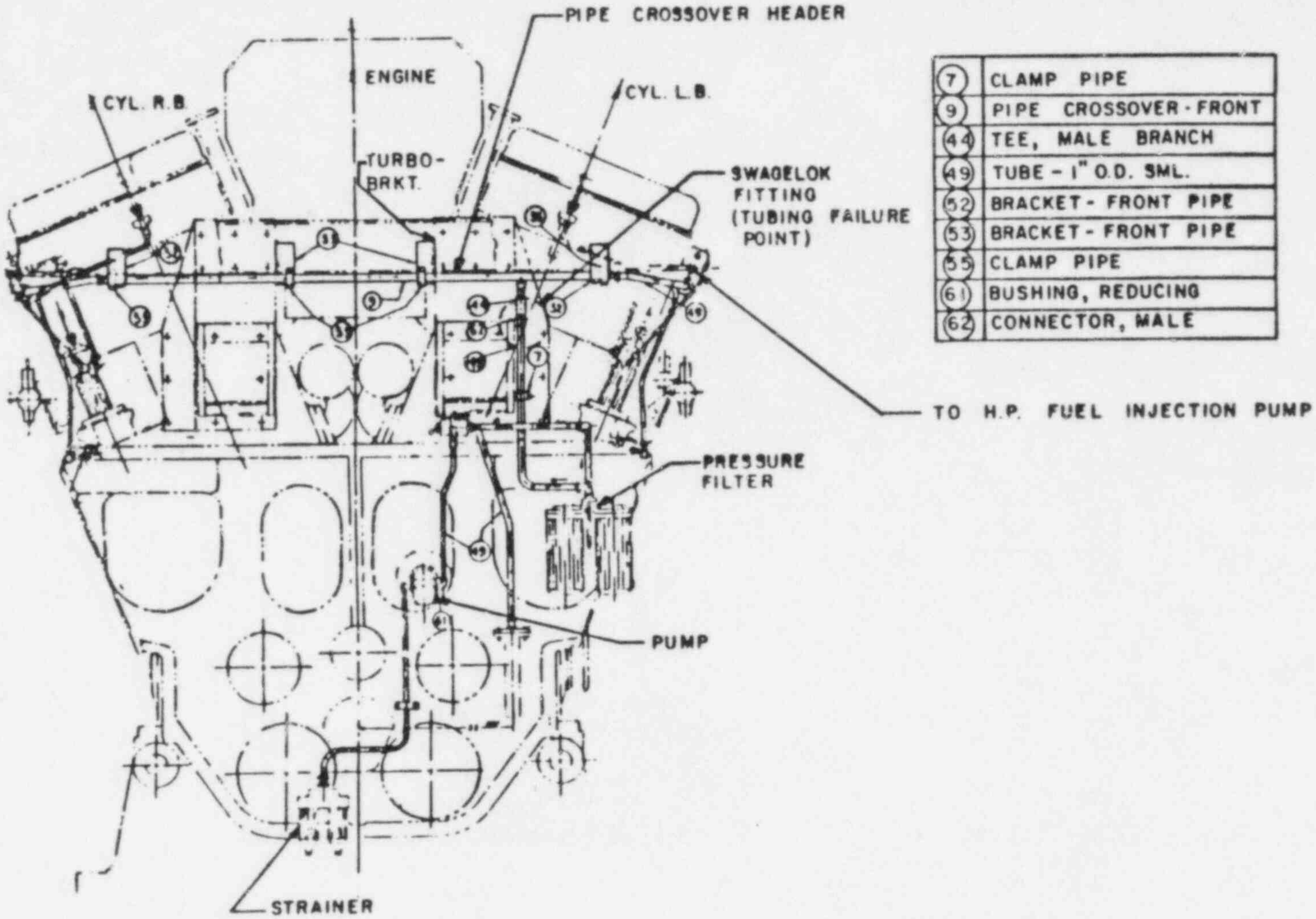
MAJOR PROBLEMS CORRECTIVE ACTIONS

- PISTON CROWNS
- CRANKCASE CAPSCREWS
- AIR START VALVES
- TURBOCHARGER VIBRATION
  - HOLDOWN CAPSCREWS
  - CRACKED WELDS
  - JACKET WATER DISCHARGE
  - LOW PRESSURE FUEL LINE FAILURE
  - FIRE
- HP FUEL INJECTION LINE FAILURE
- CRACKED WELDS ON CONNECTOR PUSHRODS


SECTION 'B-B'
SECTION 'A-A'
DELAVAL

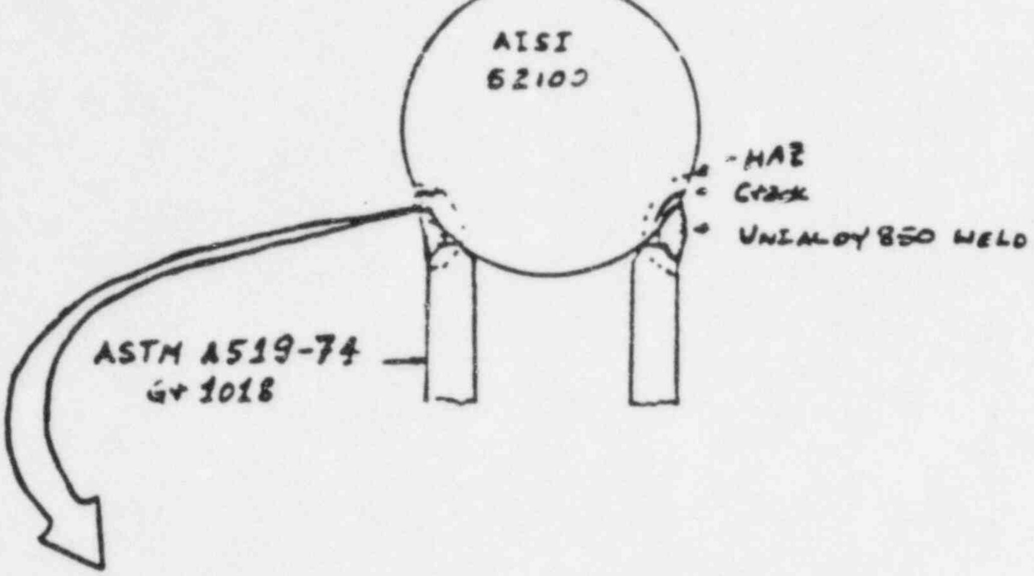
DIESEL GENERATOR

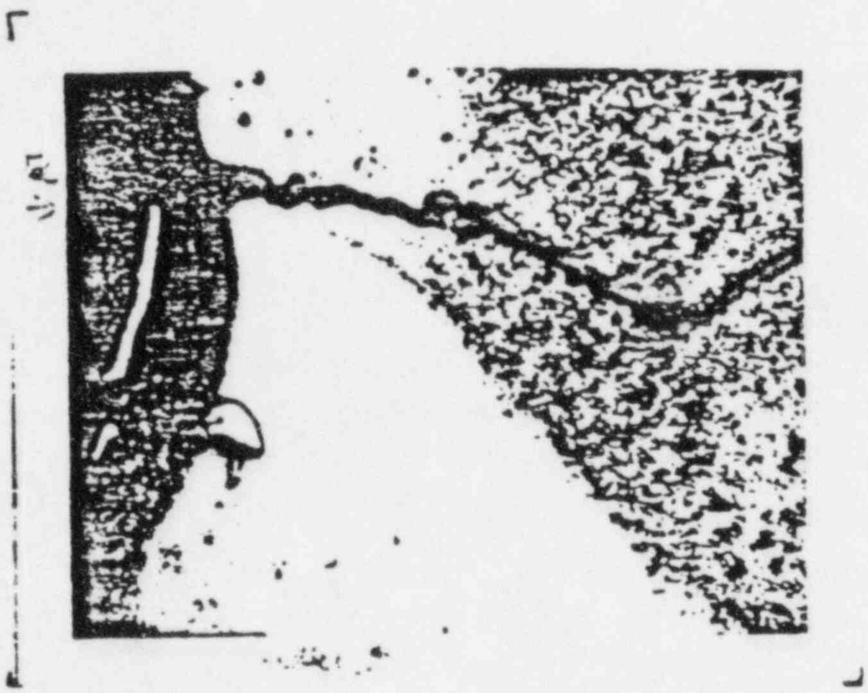




7	CLAMP PIPE
9	PIPE CROSSOVER - FRONT
44	TEE, MALE BRANCH
49	TUBE - 1" O.D. SML.
52	BRACKET - FRONT PIPE
53	BRACKET - FRONT PIPE
55	CLAMP PIPE
61	BUSHING, REDUCING
62	CONNECTOR, MALE

DELAVAL DIV. I FUEL OIL HEADER ASSEMBLY





Etchant: Nital

Magnification: 100x

Figure 8: Micrograph of crack in Ball HAZ - Left Side

Shows austenitic weld and clean fusion line. Crack emanating from ball surface and is located in the acicular martensitic HAZ.

SRC - REVIEW OF D/G RELIABILITY

- SRC CONCERNS ON D/G RELIABILITY LED TO INITIATION OF SUBCOMMITTEE REVIEW OF SUBJECT

- REVIEW COMPLETED 9/15/83

- RECOMMENDATIONS

  - PERFORM COMPLETE 18 MONTH SURVEILLANCE ON DIV I D/G, PRIOR TO RESUMPTION OF CRITICAL OPERATIONS

  - IDENTIFY ROOT CAUSE OF FUEL LINE FAILURE (DIV I D/G)

  - INSPECT ALL D/G PIPING, TUBING, AND CONNECTIONS FOR FLAWS/CONDITIONS SIMILAR TO DIV I D/G FUEL LINE

  - PERFORM 7 DAY RUN @ 50% LOAD ON ALL D/G'S, PRIOR TO 5% POWER

  - DEVELOP VIBRATION MONITORING PROGRAM FOR DIV I/II D/G'S TO IDENTIFY/DETECT VIBRATION RELATED PROBLEMS

  - MONITOR PROBLEMS WITH TDI CLOSELY, PARTICULARLY AT SHOREHAM

    - DETERMINE APPLICABILITY AT GGNS; TAKE APPROPRIATE CORRECTIVE ACTIONS

    - INCORPORATE FINDINGS INTO PREVENTIVE MAINTENANCE PROGRAM

SRC - REVIEW OF D/G RELIABILITY (CONT'D)

- CRANKSHAFT FAILURE
  - FOLLOW SHOREHAM INVESTIGATION
  - CONDUCT COMPARISON OF DESIGN, FABRICATION, OPERATIONAL HISTORIES
  - EVALUATE POTENTIAL FOR SIMILAR FLAWS AT GGNS
  - EVALUATE NEED FOR INSPECTION OF CRANKSHAFTS DURING FIRST OUTAGE
- INSTITUTE PRE-PLANNING SESSIONS FOR MAINTENANCE; REDUCE PERSONNEL ERROR

° RECOMMENDATIONS ACCEPTED AND ENDORSED BY SRC - 10/83

D/G OWNERS GROUP TIE MEETING

- Meeting Sponsored by MP&L

- Held in Atlanta, GA, October 25, 1983

- Purpose of Meeting:

  - Provide a forum for the interchange of technical information by D/G owners

  - Formulate long-term actions to improve D/G reliability

  - Provide feedback information to owners, vendors, A/Es on design, operational and maintenance problems

- Attendees:

  - Utilities represented

  - INPO

  - EPRI

  - 50 representatives in attendance

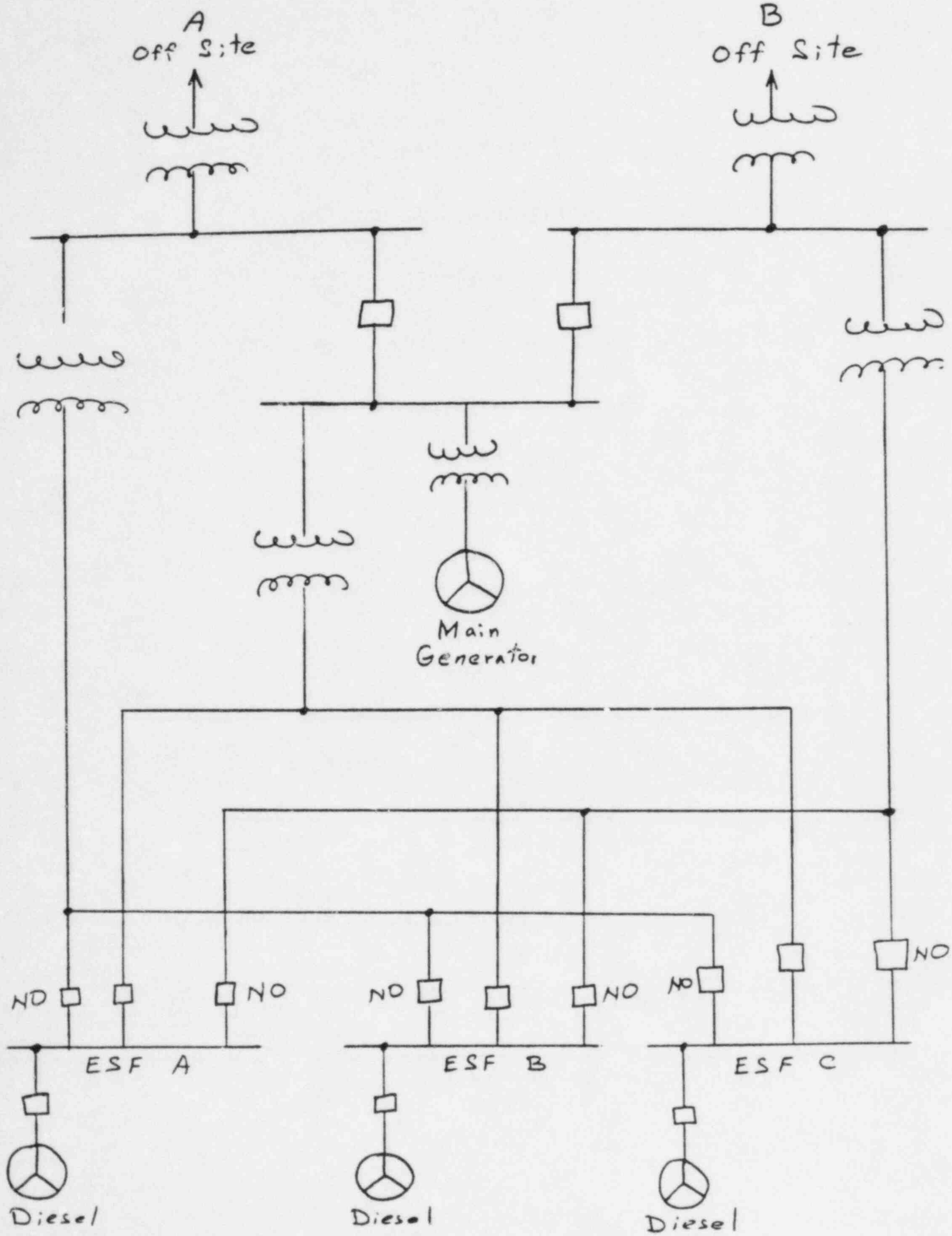
  - LILCO, EPRI, MP&L, TVA, PEC, MSS presented papers

D/G OWNERS GROUP TIE MEETING (CONT'D)

- RESULTS:
  - WELL RECEIVED
  - FORMED D/G OWNERS GROUP STEERING COMMITTEE, CHAIRED BY MP&L
  - NUTAC WILL BE CHARTERED TO ADDRESS D/G RELIABILITY ISSUES

OTHER CONSIDERATIONS

- Loss of Offsite Power Is Rare in USA
- MSU Has Strong, Reliable System
- Plant Has Advanced Distribution Design
- Plant Has RCIC, HPCS Capability


A
Off Site
B
Off Site
Main
Generator
NO
NO
NO
NO
NO
NO
NO
ESF A
ESF B
ESF C
Diesel
Diesel
Diesel

GRAND GULF NUCLEAR STATION

UNIT 1

ONE LINE DIAGRAM

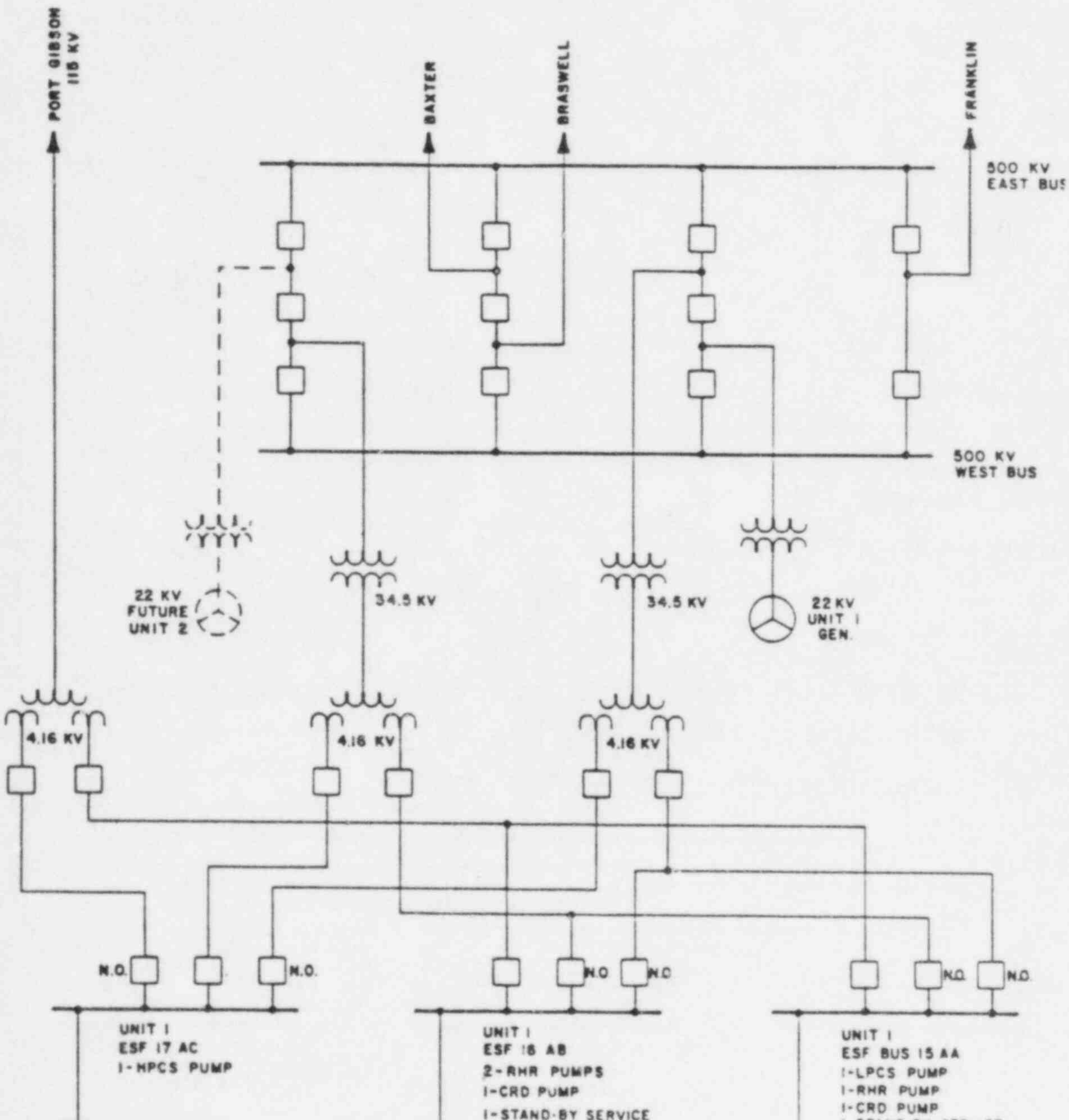


TABLE 5

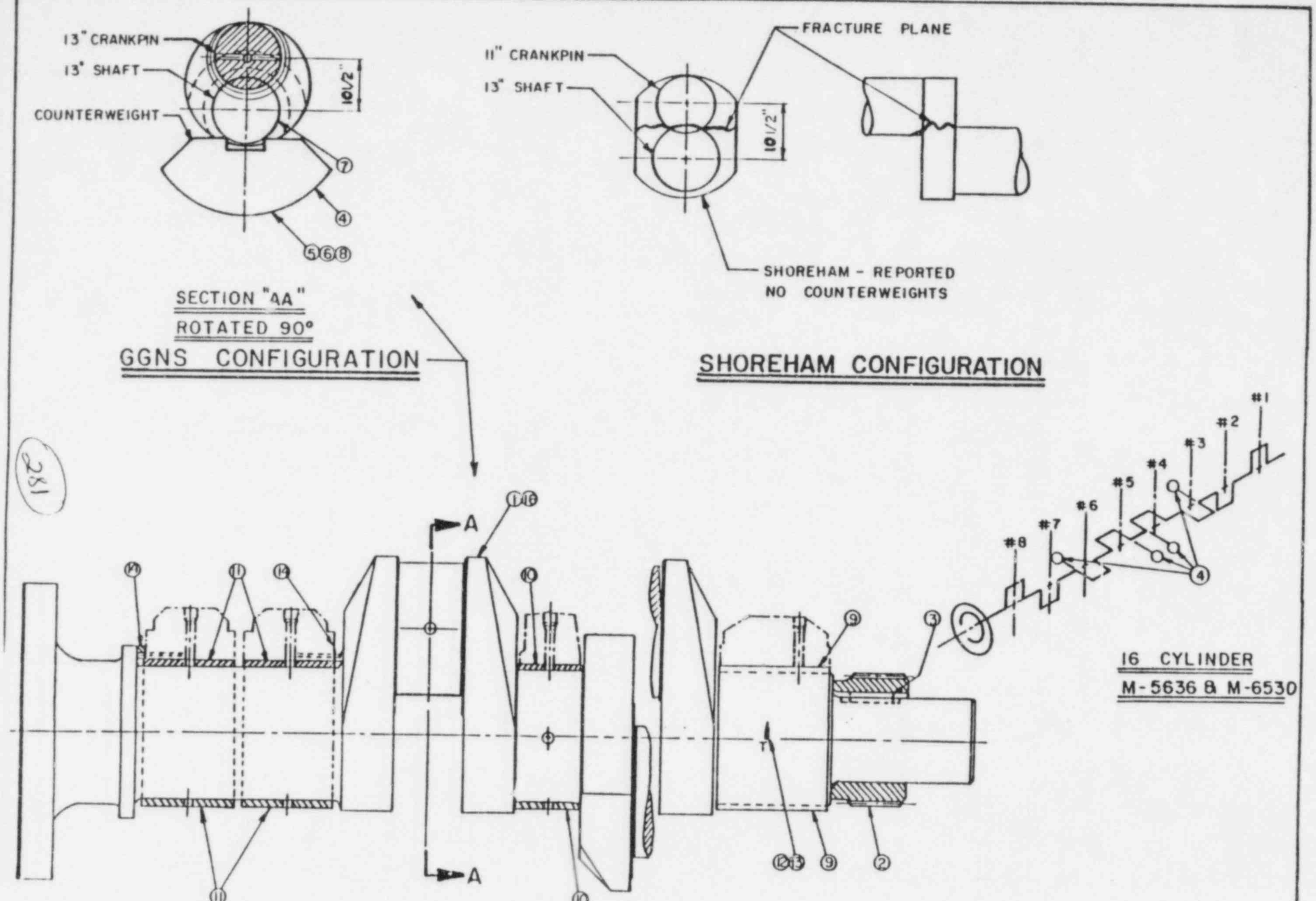
DELAVAL D/G DATA

	SHOREHAM	GRAND GULF
Horse Power	4889	9770
Elec. Output	3500kw	7000kw
Bore, In.	17"	17"
Stroke, In.	21"	21"
Crankshaft Length, Ft.	19-1/2 ft.	20' - 7"
Crank Pin Diameter, In.	11"	13"
Number of Bearings	11 Main	10 Main (Last 2 in one journal)
Crankshaft Diameter, In.	13"	13"
Compression Ratio	12:1	11.6:1
RPM	450	450
Torsional Stresses[2] MAX (PSI)	3000	5100
Synchronous (450 RPM) (PSI)	2500	1800
(500 RPM) (PSI)	6200	4700

NOTES: 1. Data on Shoreham was obtained in telephone communications with LILCO Personnel and Delaval.

2. Torsional stress data at RPM's <450

FIGURE 6



13" CRANKPIN
13" SHAFT
COUNTERWEIGHT
10 1/2"
SECTION "AA"
ROTATED 90°
GGNS CONFIGURATION
11" CRANKPIN
13" SHAFT
FRACTURE PLANE
10 1/2"
SHOREHAM - REPORTED
NO COUNTERWEIGHTS
SHOREHAM CONFIGURATION
A
A
#1
#2
#3
#4
#5
#6
#7
#8
16 CYLINDER
M-5636 & M-6530

281

CONCLUSIONS

1. DESIGNED, PROCURED, MANUFACTURED, INSPECTED, AND TESTED TO INDUSTRY STANDARDS

2. OPERATING HISTORY SHOWS HIGH STARTING RELIABILITY - GREATER THAN 95% (IN ACCORDANCE WITH R.G. 1.108)

3. PROBLEMS ENCOUNTERED HAVE BEEN STUDIED AND CORRECTED

4. MP&L INITIATED AGGRESSIVE PROGRAM FOR FURTHER STUDY

   - SRC RECOMMENDATIONS

   - TIE/NUTAC FOR D/G RELIABILITY

5. MP&L ELECTRICAL SYSTEM - RELIABLE DESIGN AND PERFORMANCE

OVERALL - GGNS D/G PROVIDES RELIABLE SOURCE OF EMERGENCY POWER

UNITED STATES
NUCLEAR REGULATORY COMMISSION
WASHINGTON, D. C. 20555

ENCLOSURE 5

OCT 3 1 1983

Docket No.: 50-416/417

Mr. J. P. McGaughy, Jr.
Vice President - Nuclear Production
Mississippi Power & Light Company
P. O. Box 1640
Jackson, Mississippi 39205

Dear Mr. McGaughy:

Subject: Request for Additional Information - TDI Diesel Generators

On October 21, 1983, we issued Board Notification No. 83-160 concerning Transamerica Delaval (TDI) emergency diesel generators. We have identified quality assurance problems at TDI and have evaluated a number of operational problems reported for the TDI units, as well as, the crankshaft failure observed at Shoreham. Our level of confidence in the reliability of all TDI diesel generators has been reduced.

To evaluate the reliability of the TDI diesel generators at Grand Gulf, we request that you provide the information identified in the enclosure. After you have studied this request, we suggest a meeting between MP&L and our staff to discuss this matter. As our investigation of the problem continues, additional requests for information may be necessary.

We request that this information be provided as soon as possible and prior to full power licensing. Where you are unable to provide the information on that schedule, we ask that you provide a justification for continued operation without this information. If you have any questions concerning this matter, please contact M. Dean Houston, Project Manager (301) 492-8358.

Sincerely,

A. Schwencer

A. Schwencer, Chief
Licensing Branch No. 2
Division of Licensing

Enclosure: As stated

cc: See next page

PDR
Dupe ~~8311140474~~

REQUESTS FOR ADDITIONAL INFORMATION
DELAVAL DIESEL GENERATOR EVALUATION
GRAND GULF UNITS 1 & 2
DOCKET NO.: 50-416/417

430.1 Provide a copy of the procurement specifications to which the standby diesel generators (DG) were ordered.

430.2 Provide the performance specification and inspections performed upon receiving the DGs to show that the procurement specifications were met.

430.3 Identify the materials used in the design of the DGs at your plant (specifically limiting components such as crankshafts, camshafts, rocker arms, bearing materials, cylinder blocks, cylinder heads, pumps, turbochargers, etc.). Discuss how you assured yourself that design materials used in the manufacture of your DGs were as stated and in accordance with materials described in the TDI proposal and your purchase specifications.

430.4 Does TDI have a program where parts/components, etc., are modified (such that design margins are reduced) in order to improve operability and DG reliability. Does this apply to any DG parts at your plant.

430.5 If applicable, provide responses to all NRC open items on standby DGs at your plant.

430.6 Identify each of your DGs by model number and rating (continuous duty and short time overload) as purchased and discuss all tests (including torsional and other design proof tests) performed on the DGs that were observed (also those not observed) by you at the manufacturer's facilities.

430.7 In addition to qualifications tests that were performed in accordance with regulatory guides 1.9 and 1.108, and IEEE Std. 387, describe all other onsite tests performed on your DGs.

430.8 In addition to any deficiency reports already provided to the NRC, summarize and describe problems encountered and resolved during installation and preliminary operation of the DGs. During this period, were any unusual or abnormal operations observed such as excessive vibration, noise, etc., and how were these conditions corrected. Provide a detailed summary of the complete operating histories of your DGs.

430.9 Tabulate, compare and discuss differences in present actual DG loading to estimated loads included in the procurement specifications. Identify the magnitude of the increased load (if any) on the DGs and describe how the increased loading affects the DG capability with regard to reserve margin.

430.10 If DG loading has increased from that specified in the procurement specifications, has it been necessary to upgrade the standby DGs to meet the new load requirements. If DG upgrading has been performed, provide a detailed description of the upgrading accomplished on your DGs. What is the revised manufacturer's rating for each upgraded unit for normal continuous duty and short time overload conditions. Is the DG built-in design margin (after upgrading) still within the recommendations of IEEE Std. 387. What is the reserve load carrying capability (margin) of your upgraded DGs.

430.11 Perform an internal visual inspection of each standby DG with regard to potential crankshaft and/or web cracks as identified at the Shoreham Station and provide a detailed discussion of your findings. In addition to the above, perform any non-destructive testing (NDT) such as dye penetrant testing, etc., as deemed appropriate to assure absence of cracks at these engine parts, or any other location where cracks are observed.

430.12 Should your inspection and NDT show evidence of crankshaft, web or cracks in any other area(s) of the machine, identify their location, size and depth and provide a detailed plan of how you propose to restore the availability and reliability of the standby DGs to acceptable standards. If cracks are observed, you may be required to respond to additional staff requests.

430.13 Should the results of your visual inspection and NDT show no evidence of cracks, justify that the DGs at your plant are sufficiently reliable so as to provide reasonable assurance that the facility [can be operated]* without undue risk to the health and safety of the public.

Your justification should include, but not be limited to the following: (1) quality assurance program conducted by you during procurement, manufacturing and receipt of your DGs, (2) your assessment of the TDI manufacturing process, inspection, and quality assurance program conducted during manufacture of your DGs, (3) your assessment of TDI responsiveness to problems that have occurred with your engines during installation and preliminary operation including assessment of TDI performance, (4) comparison of your DGs with all other TDI emergency

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*[can continue to operate]

DG models now in use or to be used in other nuclear generating stations (and other non-nuclear facilities) to show that the conditions and/or failure modes present at Shoreham will not occur at your plant and at other nuclear plants; provide any supporting information that may be obtained from non-nuclear installations, (5) have you (or others) independently reviewed or verified any TDI design calculations for critical components of your DGs, and if not how have you assured yourself that the DGs are designed to DEMA standards and applicable industry codes and standards, and (6) your overall assessment of the DGs at your plant with regard to TDI system design, operating experience to date, and system dependability, availability and reliability to warrant operation of your plant.

In addition, provide a tabulation of the number of times (including date of occurrence) voltage was lost at the emergency bus(es) requiring operation of the DG(s) including a brief description of each incident. In the above tabulation, also identify the loss of emergency bus voltage due to loss of offsite power.

430.14 Shoreham has recently identified that connecting rod bearing materials are not in accordance with design specifications on their engines. This condition may also exist on all other TDI diesels. Provide assurance that correct bearing materials have been used in your engines. Should you find that improper bearing materials have been used in your diesels, how do you propose to correct this problem, and schedule of accomplishment.

* U.S. GOVERNMENT PRINTING OFFICE: 1983-421-299:190