GRAND GULF NUCLEAR STATION

UNIT 1

UPDATE REPORT ON TDI STANDBY

DIESEL GENERATORS

April, 1984

Z3rg1

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ABSTRACT

This report contains a detailed description of the program of preventive maintenance, replacement of components with improved quality products, engine testing, and engineering evaluations which have been undertaken by MP&L, GGNS Unit 1, to enhance reliability and to assure with a reasonable level of confidence that the Transamerica Delaval, Inc. (TDI) diesel engines at Grand Gulf Nuclear Station (GGNS) will perform their required safety function.

This report is an update of the previous report submitted to the NRC in letter AECM-84/0103, dated February 20, 1984. It addresses implementation of vendor recommendations, NRC directives, problems encountered on TDI engines at other locations, potentially significant items identified by the TDI D/G Owners Group, the results of several of the TDI D/G Owners Group technical reports, and updated operating experience information.

1.0 INTRODUCTION

Grand Gulf Nuclear Station, Unit 1 is equipped with three diesel generators, two of which are supplied by Transamerica Delaval, Inc. (TDI). These two diesel generators are sources of emergency power to the GGNS Division I and Division II ESF buses. The third D/G set, dedicated to the HPCS system, is supplied by the Electro-motive Division (EMD) of General Motors.

This report provides a detailed description of a program undertaken by MP&L to enhance the reliability and performance of the two TDI diesel generators at GGNS Unit 1. The report contains a description of activities of preventive maintenance, replacement of various components with improved quality products, and testing of the two TDI diesel generators.

The improvement program includes specific actions which have been or are being taken to correct the problems experienced with TDI diesel generators during the start-up testing phase of GGNS Unit 1. Potential problems identified to MP&L as a result of the experience with TDI diesel generators at other nuclear installations are also addressed.

The main emphasis of this report is to provide the results of an engineering evaluation of the two TDI diesel generators at GGNS Unit 1 for their reliability and performance. This evaluation is intended to provide reasonable assurance to the NRC that the TDI diesel generators will perform their required safety function. This report supplements earlier reports on the Division I and II diesel generators (Reference 12, 13, 14, 19, 20).

Sections 2 thru 9 of the report contain descriptions of repairs or modifications which have been performed. Section 10 concerns TDI's product improvement program. Section 11 focuses on the testing programs, both the testing done in the past and the testing performed after completion of the piston skirt changeout. A summary of the overall engineering evaluation is provided in Section 12. The conclusions reached from these evaluations are provided in Section 13.

Attachment 1 provides details of concerns raised at a meeting of the TDI diesel generator owner's group with the NRC on January 26, 1984, their applicability to Grand Gulf and their resolution.

Attachment 2 provides a summary of various piston skirt designs that have been or are in use in the GGNS TDI diesels.

Table 1-1 provides a list of the principal design specifications for GGNS Unit 1 TDI diesel generators.

Table 1-2 shows the total operating hours, starts, valid tests and valid failures for the GGNS TDI D/Gs. Table 1-2 also shows that the ratio of valid failures (2) to valid starts (137) results in an excellent start reliability in excess of 98 percent.

1.0 (Continued)

Table 1-3 shows the Division I and II approximate run hours under load since the originally furnished piston skirts were modified in November, 1981.

Table 1-3A shows additional testing that has recently been completed on the Unit 1 TDI diesel generators. The 7 day runs, 24 hour runs and 100 hour runs for the TDI D/Gs consist of 11 runs ranging between 60% to 110% load for a total of 695.7 hours or an average of 63.2 hours per run.

Table 1-4 shows the procurement specification estimated electrical loads and the present electrical loadings for the Division I and II diesel generator.

The significant work activities completed on the Division I and II engines are:

- All piston skirts have been replaced with skirts of improved design,
- All 32 cylinder heads were inspected and eight cylinder heads with rejectable indications have been replaced,
- All Division I and Division II connecting and main push rods have been replaced with components of improved design,
- All connecting rod bearings have been replaced,
- Inspection of both crankshafts has been completed, and
- Rework of turbocharger piping and components using ASME welding, procedures and materials has been completed,
- Realignment of the turbocharger has been completed.

These work activities are intended to enhance engine performance and reliability. They have insignificant impact on engine specifications, design criteria, subsystems or performance characteristics.

None of the work activities affect the design considerations listed in Table 1 of IEEE 387-1977. As such, these work activities are considered minor design changes as defined by IEEE 387-1977. Therefore, the post maintenance qualification and availability testing of these diesels was planned according to the guidelines established in IEEE 387-1977 for minor changes. Additional testing was also performed.

TABLE 1-1

DELAVAL ENGINE SPECIFICATIONS

Model	DSRV-16-4
Quantity	2
Engine Serial Number	74033-2624 & 74034-2625
Service	Stationary generator for nuclear service
Fuel Mode	Diesel
Configuration	45° "V" type
No. of Cylinders	16
Bore (in.)	17
Stroke (in.)	21
Cycle Model	4 stroke
Total Displacement (cu. in.)	76,266
Crankshaft Rotation	CW (from Flywheel end)
Firing Order	1L-8R-4L-5R-7L-2R-3L 6R-8L-1R-5L-4R-2L-7R 6L-3R
Continuous Rating (kw)	7000
Overload Rating (kw)	7700
Crankshaft Diameter (in.)	13
Crank Pin Diameter (in.)	13

TABLE 1-2

GGNS D/G OPERATING DATA

Total Run Hours	Division I	Division II
Shop and Pre-Op Run Time (Hrs)	535	252
Since Date of OL Run Time (Hrs)	862	618
Total Run Time (Hrs) (3)	1397	870
TOTAL NO. OF STARTS (3)		
Delaval Shop Runs (1)	310 (2)	5
Pre-Operational Runs	60	60
Since Date of OL Runs	170	120
Total Starts (4)	540	185

NOTES:

1. Source of Information - Delaval Technical Manuals.
2. Division I engine had 300 prototype runs for reliability testing.
3. Data as of April 4, 1984
4. Valid Tests: Div I - 84

   Div II - 53

   137

   Valid failures: 2 (1-Div I Control System Electrical Component) (1-Div.I - Unknown)

   Reliability: 98.5%

   Valid tests and failures are as defined in Regulatory Guide 1.108.

TABLE 1-3

DIVISION I AND II APPROXIMATE
RUN HOURS UNDER LOAD SINCE ORIGINALLY FURNISHED
PISTON SKIRTS WERE MODIFIED IN NOVEMBER, 1981 TO APRIL 4, 1984

Load, + 5%	Division I Hours	Division II Hours
< 50	14	12
50 - 60	450	316
60 - 99	75	13
100	301	251
110	14	10

TABLE 1-3A

RECENT SPECIAL TESTING

PURPOSE	Division I	Division II
7 Day Run	168 hr @ 60%	32 hr @ 60%
		37 hr @ 60%
		70 hr @ 60%
		46 hr @ 60%
	After Piston Skirt Changeout	
Breakin Runs	1 hr @ 20%	1 hr @ 20%
	1 hr @ 20%	1 hr @ 25%
	2 hr @ 50%	2 hr @ 50%
	2 hr @ 75%	1 hr @ 75%
	2 hr @ 100%	2.8 hr @ 100%
	2 hr @ 100%	1 hr @ 20%
	4 hr @ 100%	.5 hr @ 100%
	.8 hr @ 50%	.4 hr @ 100%
	2 hr @ 110%	.2 hr @ 110%
	23 hr @ 100%	1 hr @ 50%
		2 hr @ 110%
		22.1 hr @ 100%
Surveillances:		
4 Hour Runs	8 hr @ 100% (2 runs)	4 hr @ 100%
Reliability Runs		
100 Hour Runs	21 hr @ 100%	100.3 hr @ 100%
	72.3 hr @ 100%	
	100 hr - 32 hr @ 100%, 68 hr @ 75%	

TABLE 1-4

D/G LOADINGS

	Division I	Division II
Procurement Specification	5730 KW	6100 KW
Design DG Rating	7000 KW	7000 KW
Loss of Offsite Power Loads	3627 KW (51.8%)	4745 KW (67.8%)
Post LOCA Loads	4711 KW (67.3%)	3914 KW (55.9%)
Total Connected ESF Bus Load	5963 KW (85.2%)	6397 KW (91.4%)

2.0 PISTONS

2.1 DESCRIPTION

MP&L received information from TDI that during a recent reassembly of the three TDI diesels at the Shoreham station, an inspection of the piston skirts revealed linear indications exceeding 1/16 inch in length in 23 of 24 modified "AF" piston skirts.

As a result of this finding, TDI generated a 10CFR21 report recommending that GGNS and San Onofre inspect 25% of the modified "AF" piston skirts in each engine for linear indications. MP&L subsequently found rejectable indications in three of four modified "AF" piston skirts during the 25% inspection on the GGNS Division II engine. All piston skirts on the Division II D/G were then inspected. The results of these inspections are shown in Table 2-1 and 2-2. The inspection criteria used for the inspection is described in Step 2.3 of this section.

2.2 ENGINEERING EVALUATION

Failure Analysis Associates (FaAA) performed an inspection and analysis of the modified type "AF" piston skirts which were removed from the Shoreham diesels. After comparing the GGNS Division II piston skirt inspection results with the Shoreham evaluation results (Reference 1), FaAA concluded that the GGNS Division I piston skirts could contain fatigue cracks of the same approximate depth as the Shoreham engines.

As a result of these early evaluations MP&L replaced all piston skirts in the two Unit 1 TDI engines with the improved "AE" style skirt provided by TDI. (See Attachment 2 for Details of Piston Designs). MP&L worked with TDI in the final phases of production and inspection of these piston skirts to assure that they are free of rejectable indications (as evidenced by fluorescent magnetic particle examination).

Results of an evaluation by FaAA (Reference 21) indicates that the "AE" type piston will exhibit substantially lower stresses than the replaced modified "AF" type under similar loadings. FaAA has also indicated that these AE skirts have now been operated for over 300 hours in one of the Shoreham engines including 100 hours of full power operation. Other AE skirts have accumulated over 6000 hours in a stationery generating plant, and over 600 hours in an advanced development engine. Inspection of these skirts (one after 6000 hours, two after 600 hours, and four after 300 hours) with a high-resolution eddy current procedure disclosed no cracking.

Experimental stress analysis of a type AE skirt was conducted under hydrostatic loading of the piston crown. The maximum stress measured was below the yield strength of the material and corresponds to cylic loading below the value required to produce a crack.

It was concluded that, based on both analysis and test results, the "AE" type piston skirt attachment would not fail in fatique.

2.3 PISTON INSPECTIONS

(1) Division II modified "AF" piston skirts were nondestructively examined with liquid fluorescent dye penetrant and/or wet fluorescent magnetic particle processes. The specific area of concern was the filleted transition area between the skirt/crown stud hole bore and the skirt wall. All critical filleted areas of each modified "AF" piston skirts were initially inspected with the liquid fluorescent dye penetrant process. The results of this initial inspection are summarized in Table 2-1. The following criteria were used in recording possible indications:

1. all indications were to be evaluated, and

2. a linear indication is defined as an indication in which its length is greater than three times its width.

Numerous indications were found, ranging from 1/32 to 9/16 inches in length. The following additional inspections were performed to determine if the linear indications were superficial in nature. Each linear indication was ground and/or sanded to a depth of approximately 0.062 inches. These indications were then re-inspected using the liquid fluorescent dye penetrant or wet fluorescent magnetic particle process. Linear indications were found ranging from 1/32 to 1/2 inch in length. The results of these additional inspections are summarized in Table 2-2.

To characterize these indications, a confirmatory metallurgical analysis will be performed. The analysis will attempt to determine the mode of cracking, characterize the crack propagation rate, and estimate the depth.

(2) All replacement "AE" piston skirts were nondestructively examined by TDI using the wet fluorescent magnetic particle process prior to installation in the engines. All TDI nondestructive examination procedures were reviewed and approved by MP&L. The following criteria were established as levels of unacceptability:

1. any linear indication greater than 3/16 inch long,

2. rounded indications with dimensions greater than 3/16 of an inch,

3. four or more rounded indications in a line separated by 1/16 of an inch or less, edge to edge, and

4. cracks and hot tears.

These acceptance criteria were derived from ASTM Standard E 125-63, reapproved 1980.

2.3 (Continued)

All piston skirt castings were accepted to the above criteria. It should be noted that all acceptable indications that were found were documented by appropriate records.

2.4 MANUFACTURING DETAILS

The manufacturing details for the "AF", modified"AF" and "AE" piston designs have been provided to MP&L by TDI. The evolution of TDI's piston design is relevant to this report. As such, the details of manufacture for each of these piston designs is presented in Attachment 2.

It is important to note that the "AE" design utilizes a reinforced (lower stressed) casting and a half-stack Belleville washer arrangement. Also, "AE" skirts are heat treated to produce stress relieved nominal 100,000 psi tensile strength nodular iron. The "AE" style skirt is interchangeable with existing R-4 piston crown and requires only minor hardware changes.

2.5 CONCLUSIONS

As a result of the Division II modified "AF" piston skirt inspection, the piston skirts in the Division I and II D/Gs have been replaced with the type "AE" piston skirts.

Based on results of the analytical work completed by FaAA and the operating experience and subsequent inspection of the piston skirts on the TDI Kodiak engine, R-5 test engine, and the Shoreham engine, MP&L has concluded that the "AE" design is capable of performing the required function at all running loads, and the "AE" piston skirt attachment would not fail in fatique.

TABLE 2-1

RESULTS OF INITIAL INSPECTIONS OF GGNS MODIFIED AF PISTONS
IN THE DIVISION II D/G

Piston Identification	Indication Length (Inches)(1) Stud Hole Bore Area (2) #1	#2	#3	#4
#1RB	None	1/8	3/32	1/4,1/32,1/16
#1LB	None	1/32	None	1/16
#2RB	None	1/32	None	1/4,1/16
#2LB	5/64,1/16	None	3/16	1/8,1/32
#3RB	1/4	1/32,3/16	None	1/2
#3LB	1/32	1/32	None	3/16
#4RB	None	None	1/4	None
#4LB	1/16,1/8	None	3/32,1/8	1/16
#5RB	1/32	1/32	1/4	1/4
#5LB	3/32,3/32	1/8	9/16	3/8
#6RB	1/4	3/16	None	1/4
#6LB	1/32	None	1/16	1/32
#7RB	None	None	None	3/32
#7LB	None	None	None	1/32
#8RB	3/32	None	None	None
#8LB	1/16,1/8	None	1/16	1/4

General Notes:

(1) All inspection performed using Liquid Fluorescent Penetrant Process.

(2) See Figure 2-1 for location of the stud bore area within piston skirt.

TABLE 2-2

RESULTS OF ADDITIONAL INSPECTION OF GGNS MODIFIED AF PISTONS IN THE DIVISION II D/G

Piston Identification	Indication Length (Inches)(1) Stud Hole Bore Area (2) #1	#2	#3	#4
#1RB	--	1/8 (MT)	NAD (MT)	NAD (MT)
#1LB	--	NAD (MT)	--	NAD (MT)
#2RB	--	NAD (PT)	--	NAD (PT)
#2LB	NAD (PT)	--	NAD (PT)	NAD (PT)
#3RB	1/4 (MT)	3/16 (MT)	--	NAD (MT)
#3LB	1/8 (MT)	1/8 (MT)	--	3/16 (MT)
#4RB	--	--	1/4 (MT)	--
#4LB	1/8 (MT)	--	1/8 (MT)	1/16 (MT)
#5RB	NAD (MT)	NAD (MT)	1/4 (MT)	NAD (MT)
#5LB	NAD (MT)	NAD (MT)	1/2 (MT)	1/4 (MT)
#6RB	NAD (MT)	NAD (MT)	--	1/4 (MT)
#6LB	NAD (MT)	--	NAD (MT)	NAD (MT)
#7RB	--	--	--	1/32 (MT)
#7LB	--	--	--	NAD (MT)
#8RB	NAD (MT)	--	--	--
#8LB	1/8 (MT)	--	NAD (MT)	1/4 (MT)

General Notes:

(1) PT indicates Liquid Fluorescent Dye Penetrant Inspection.
(2) MT indicates Fluorescent Magnetic Particle Inspection.
(3) See Figure 2-1 for location of the stud bore area within piston skirt.
(4) -- Not performed. No discontinuities present during initial inspection.
(5) NAD No apparent defect.

FIGURE 2-1: LOCATION OF STUD BORE AREAS WITHIN PISTON

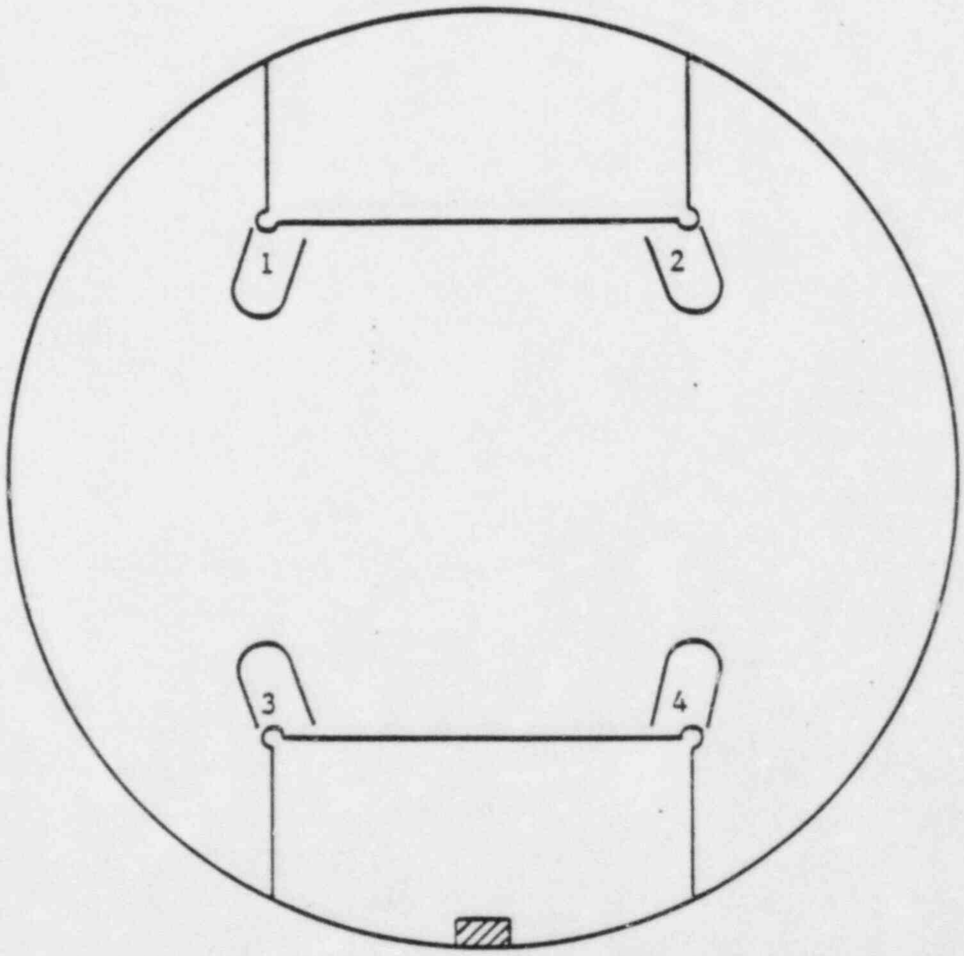


FIG. 2-1

3.0 CYLINDER HEADS

3.1 DESCRIPTION

During disassembly of the Division II engine for piston inspections, red rust was reported in the area of the exhaust valve seats on the #5 right bank head. Subsequent color contrast dye penetrant inspections showed cracks in the stellite exhaust valve seat overlays. Because of the rusting, it is postulated that one of the cracks may extend into the water jacket.

3.2 INSPECTION AND ENGINEERING EVALUATION

As a result of these cracks, an investigation has been initiated to determine the extent of the cracking and required corrective action. The investigation has been divided into two parts; short and long term. The short term investigation was initiated to determine if the extent of cracking is generic to all heads at GGNS. All cylinder heads on both Division I & II D/Gs were removed and were nondestructively examined in the area of the stellite seats with a color contrast solvent removable dye penetrant process. The following criteria were established as levels of unacceptability:

1. Any cracks or linear indications

2. Four or more rounded indications in a line separated by 1/16 inch or less, edge to edge

3. Any rounded indication with dimensions greater than 1/16 inch

4. Linear indications are those indications in which the length is more than three times the width

Two of the 16 heads on the Division II D/G and six of the 16 heads on the Division I D/G were determined to have rejectable indications. Of these, only the Division II D/G #5 right bank cylinder head had an apparent through wall crack. No other visual evidence of cracking was found in the cylinder heads. A description of the indications found during these inspections are detailed for Division I in Table 3-1 and Division II in Table 3-2.

The heads which were rejected on the Unit 1 TDI engines were rejected for minor indications which were revealed during a liquid penetrant examination of the valve seat areas. It is not known if any of these indications would have propagated, but the decision was made to install only clean heads with no indications. To address the long term concern, a failure investigation has been initiated. A metallurgical evaluation will be performed to determine the cause of crack initiation, and the crack propagation mode.

3.3 CORRECTIVE ACTION

Based on the short term investigations, MP&L replaced the two heads on the Division II D/G and the six heads on the Division I D/G with heads that were examined and determined to have no rejectable valve seat indications. No further action is planned, pending the results of the long term investigation.

3.4 CONCLUSIONS

Two heads on Division II and six heads on Division I were determined to have rejectable indications. However, as demonstrated by the operability of the D/Gs prior to the replacement of these heads, the ability of the Unit 1 D/Gs to perform their safety function was not impaired. The replacement of these eight heads with heads free of rejectable valve seat indications provides additional assurance that the potential for head cracking from this source is minimized.

To provide further assurance that any significant cracks in the heads will be detected, additional surveillance will be performed following D/G operation to detect the presence of water in the cylinders. These surveillances will be in addition to the current surveillances which are designed to check for the presence of water prior to manually initiated D/G starts.

TABLE 3-1

INSPECTION RESULTS OF DIVISION 1 CYLINDER HEADS

Head Identification Number	Inspection Results
1LB	No Apparent Defects
1RB	No Apparent Defects
2LB	Linear Indications on Fusion Zone Between Casting and Stellite Valve Seat
2RB	No Apparent Defects
3LB	No Apparent Defects
3RB	No Apparent Defects
4LB	No Apparent Defects
4RB	Linear Indication in Stellite Valve Seat
5LB	Linear Indications in Stellite Valve Seat
5RB	No Apparent Defects
6LB	Linear Indication in Stellite Valve Seat
6RB	Linear Indications in Stellite Valve Seat
7LB	No Apparent Defects
7RB	No Apparent Defects
8LB	No Apparent Defects
8RB	Linear Indication in Stellite Valve Seat

TABLE 3-2

INSPECTION RESULTS OF DIVISION II CYLINDER HEADS

Head Identification Number	Inspection Results
1LB	Incomplete fusion 5/16 inch long on intake valve seat
1RB	No Apparent Defects
2LB	No Apparent Defects
2RB	No Apparent Defects
3LB	No Apparent Defects
3RB	No Apparent Defects
4LB	No Apparent Defects
4RB	No Apparent Defects
5LB	No Apparent Defects
5RB	Twelve linear indications ranging from 3/16 to 3/4 inch. All indications transverse to stellite overlay on two exhaust valve seats. All cracks are contained within the valve seat except for one, which extends from stellite into cast head material.
6LB	No Apparent Defects
6RB	No Apparent Defects
7LB	No Apparent Defects
7RB	No Apparent Defects
8LB	No Apparent Defects
8RB	No Apparent Defects

4.0 CONNECTING ROD BEARINGS

4.1 DESCRIPTION

Shoreham has experienced cracks in connecting rod bearings. These cracks were discovered (See Reference 2), when LILCO disassembled the three diesel engines at Shoreham (TDI Model DSR-48) to investigate a crankshaft failure (See Section 6.0). A complete inspection found that four of the forty-eight connecting rod bearing shells contained cracks. Even though the Grand Gulf D/G design is significantly different (i.e., GGNS has articulated connecting rod design and reduced connecting rod bearing loads) an inspection and evaluation was performed to determine if this concern exists at Grand Gulf.

4.2 ENGINEERING EVALUATION, SHOREHAM BEARINGS

A schematic of a cracked Shoreham bearing is shown in Figure 4-1. FaAA performed an analysis (Reference 18) on one of the cracked Shoreham bearings. The cracked bearing was checked for its chemical and physical properties. A scanning electron microscopy (SEM) analysis of the fracture was also performed and dimensional checks were made for wear. The chemical and physical properties met the current design specifications except for elongation. The elongation was found to be below specification, however, the test specimen was not standard, and led to results that were inconclusive. Reference chemical properties for B850.0-T5 are shown in Table 4-1. A Shoreham SEM examination of the fracture face indicated that voids in the bearing shell may have been crack initiation locations. In compression, voids in the "overhang" area would not pose a problem. However, the bearing/rod arrangement on the Shoreham diesels did not support the end part of the bearings (Figure 4-2). This unsupported end combined with the yawing of the crankshaft would put the internal diameter surface into tension. The surface porosity acting as a stress intensifier may have contributed to crack initiation in the unsupported end ("overhang" area). Also, the subsequent shell thickness measurements showed the bearing to be within the manufacturing tolerances, i.e., no appreciable wear.

FaAA has indicated (Reference 22) that two analyses were performed to determine the effect of the stress reduction on the fatique resistance of the new 12-inch bearing shells. A stress vs. number of cycles equation predicted that, based on the observed life of the 11-inch diameter bearing shells, the 12- and 13-inch shell fatigue life should be approximately 38,000 hours at full load, which is over ten times the usage expected over the 40-year service life of the nuclear standby diesel generators. Expected full load hours at GGNS are 75 full load hrs/year. An alternative analysis demonstrated that the decrease in the stress range is sufficient to prevent fatigue cracks, which indicates an infinite fatigue life for the bearing shells.

Based on fracture mechanics analysis, an acceptance criterion for discontinuities in the aluminum was established. Voids up to 0.050 inch in diameter will not compromise the fatigue performance of the 12-inch and 13-inch connecting rod bearing shells in DSR-48 or DSRV-16-4 standby diesel generators. The maximum load required to be carried by the GGNS TDI D/Gs during an emergency event is only 68% of rating (Section 1-Table 1-4). Further analysis is being performed by FaAA to determine fatigue life at lower loads assuming larger bearing voids in order to envelope less conservative conditions.

4.3 GGNS D/G INSPECTION

The inspections delineated below were performed on Division II D/G components. New bearings were installed in both divisions to expedite the return to service of the diesel and to extend the replacement period of the bearings. The integrity of these replacement bearings was based on an exact part number exchange, visual inspection of the bearings before installation, and favorable results from the inspection/evaluation performed on the original bearings.

1. All of the connecting rod bearings were dimensionally checked for wear and signs of unusual or abnormal wear patterns.

2. Two (25% of total) of the connecting rod bearings were inspected by liquid penetrant (PT) and radiography. The radiography technique utilized an x-ray tube radiation source and obtained a 2-2T film sensitivity yielding at least of 0.015 to 0.020 inches resolution. The PT inspections used a liquid fluorescent dye penetrant process and met the requirements of ASTM Standard E165. No rejectable indications were found.

   Tests to check the chemical and physical properties of two (25% of total) original connecting rod bearings are planned. These tests will be performed in accordance with applicable ASTM Standards. These tests are considered confirmatory in nature.

3. The "overhang arrangement" of two bearings/connecting rod assemblies was dimensionally checked for unsupported bearing material. The chamfers on the connecting rods and rod bearings were dimensionally checked to determine if the "overhang arrangement" exists and to verify that the connecting rod configuration was of the correct design.

4.4 INSPECTION RESULTS

The initial Division II D/G inspections indicated the following:

1. Review of the radiographic film showed that any bearing porosity was less than 0.030 inches, however, some linear type indications were present. All linear type indications were directly traceable to minor gouges and marring located on the bearing surfaces

4.4 (Continued)

which occurred during disassembly. As indicated in Reference 22 porosity of less than 0.050 inches is predicted to be of little consequence to the satisfactory operation of the bearings.

2. The results of the dimensional inspections confirmed that the bearings were within manufacturing tolerances. No signs of unusual or abnormal wear patterns were noted. This indicates that there was no misalignment between the connecting rod assemblies and the crankshaft.

3. The results of the chamfer measurements indicate that there is no "overhang" arrangement on the #7 connecting rod/bearing assembly and that the #2 connecting rod/bearing assembly has an "overhang" of approximately 0.016 inch (i.e., 0.016 inch of unsupported bearing material). This amount of "overhang" is insignificant compared to the 0.25 inch "overhang" that existed on the Shoreham bearings at the time of bearing cracking.

4. The results of the Liquid Fluorescent dye penetrant examination indicated that no cracks were present.

4.5 CONCLUSIONS

The differences in design between Grand Gulf and Shoreham (i.e., articulated connecting rod design and reduced connecting rod bearing loads at Grand Gulf) preclude the types of problems that Shoreham has experienced. However, inspections were performed to verify the adequacy of the bearings.

Inspections of the original Division II D/G connecting rod bearings showed that no appreciable wear or unusual wear patterns were present. This confirms proper alignment of the connecting rod assemblies to the crankshaft.

During the piston skirt changeout on the GGNS Unit 1 D/Gs the connecting rod bearings were inspected for unusual or abnormal wear patterns. No signs of unusual or abnormal wear patterns were noted. Two of the original Division II D/G connecting rod bearings were inspected by radiography. The radiography technique utilized an X-ray tube radiation source and obtained a 2-2T film sensitivity yielding at least 0.015 to 0.020 inches resolution. Review of radiographic film showed that bearing porosity was less than 0.030 inches which is well below the preliminary acceptance criteria of 0.050 inches established by the TDI D/G Owners Group. (Reference 22). Further analysis is being performed by FaAA to determine fatigue life at lower loads assuming larger voids to envelope more realistic service conditions.

TABLE 4-1

CHEMICAL SPECIFICATION LIMITS
FOR ALCOA B850.0-T5 ALUMINUM

Element	Composition %
Si	0.4 Max
Fe	0.7 Max
Cu	1.7 - 2.3
Mn	0.10 Max
Mg	0.6 - 0.9
Ni	0.9 - 1.5
Sn	5.5 - 7.0
Ti	0.20
Other Elements	0.30 Max
Al	Remainder

FIGURE 4-1: SHOREHAM CONNECTING ROD BEARING DESIGN AND NOMENCLATURE (SCHEMATIC)

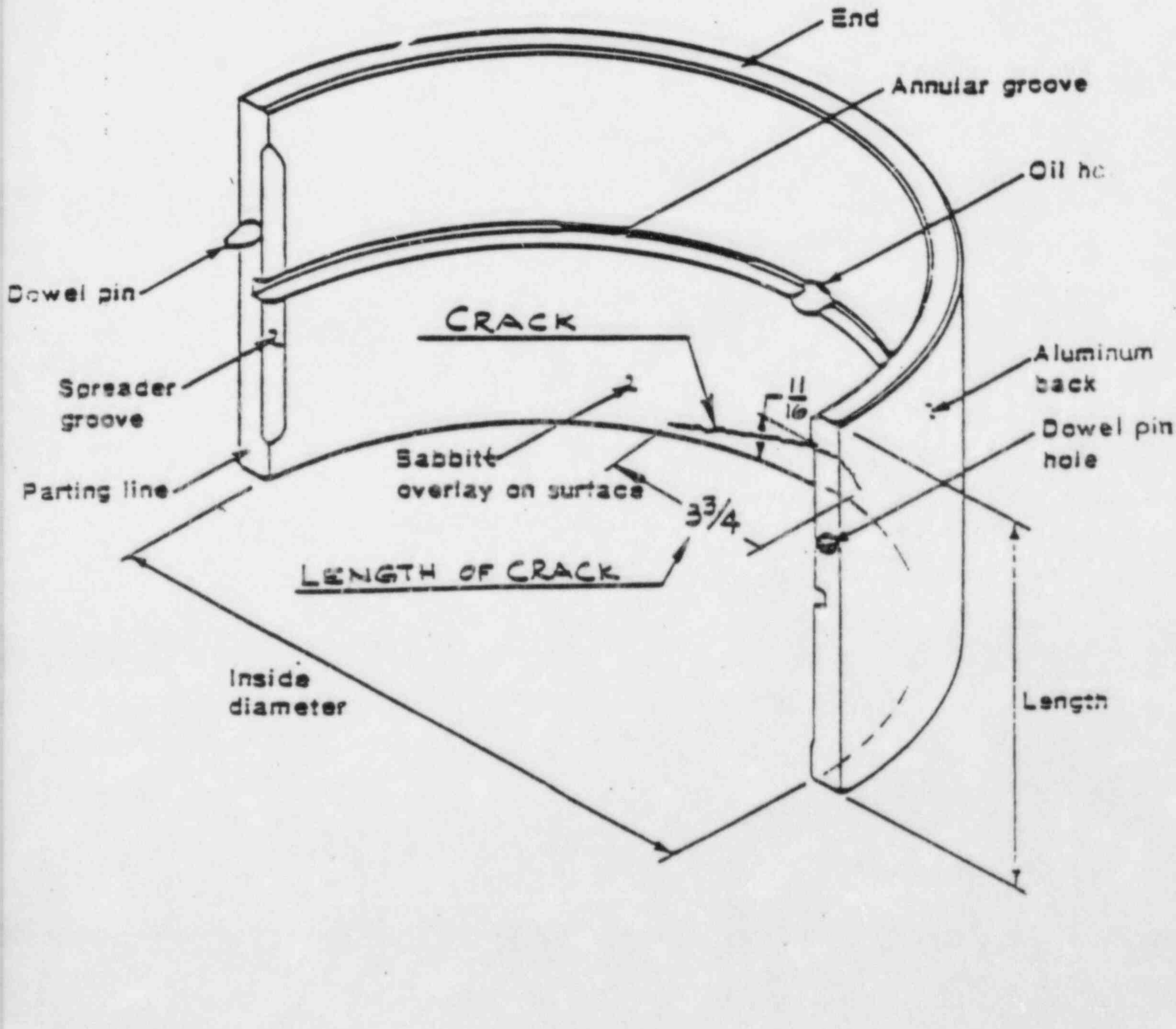
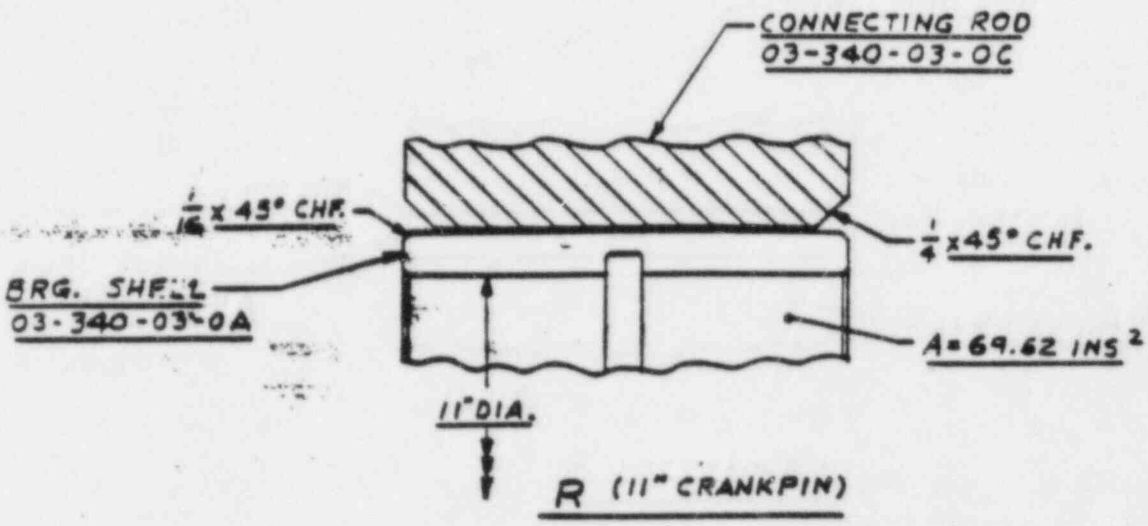
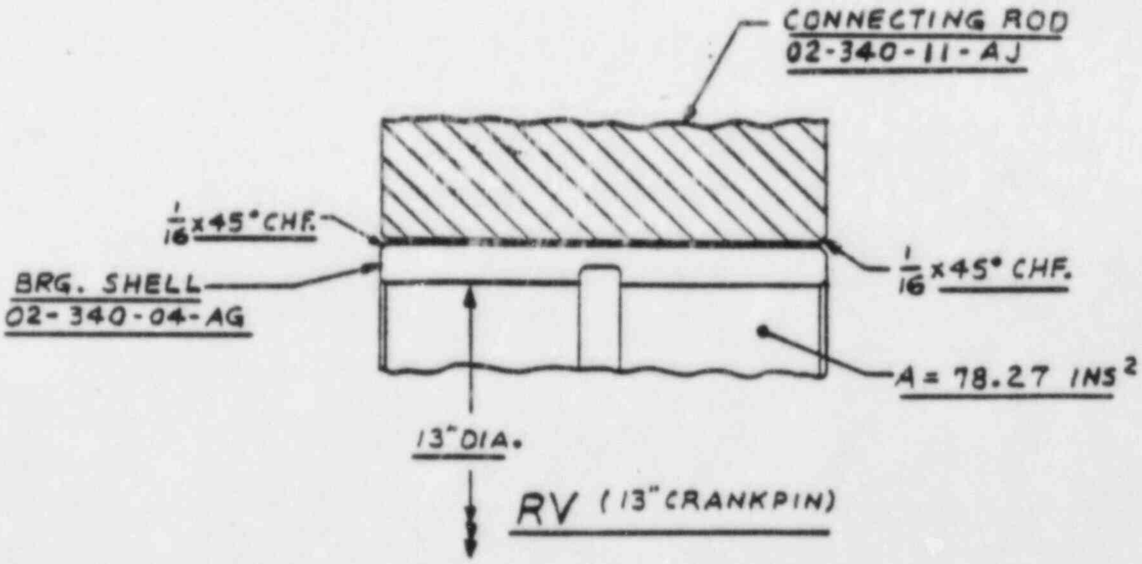


FIG. 4-1

FIGURE 4-2: COMPARISON OF CONNECTING ROD/BEARING CHAMFER ARRANGEMENTS



SCHEMATIC OF SHOREHAM "OVERHANG ARRANGEMENT" INDICATING UNSUPPORTED BEARING MATERIAL



SCHEMATIC OF GGNS BEARING ARRANGEMENT WITH BEARING MATERIAL SUPPORTED

FIG. 4-2

5.0 PUSH RODS

5.1 DESCRIPTION

On August 11, 1983, during the performance of unrelated maintenance, a rocker arm connector push rod was found to have a cracked weld. The push rod ball disengaged from the shaft as the push rod was removed from the Division I engine. The defective connector push rod was replaced and the Division I engine was tested and returned to service with additional connector push rod inspection criteria specified. During a subsequent inspection of the Division I engine, 14 of 16 connector push rods were discovered with cracked or separated welds. This inspection revealed one of the connector push rod balls was cracked in addition to the weld cracks previously observed. During the inspection of the Division II engine in December, 1983, 13 of 16 connector push rods were also found to have cracked tube-to-ball welds.

5.2 ENGINEERING EVALUATION AND CORRECTIVE ACTION

There were two types of push rod designs used at GGNS. The main push rods had a tubular steel shaft fitted with hardened steel end pieces which were attached to the tube with four plug welds near the ends of the tube. According to TDI, an estimated 2 percent of this design developed cracks in or adjacent to the plug welds on the rods.

The connector push rod consisted of a tubular steel body fillet-welded to carbon steel ball bearings. This design is the type which exhibited defects at Grand Gulf and is shown in Figure 5-1. A 1 1/2-inch high carbon steel ball bearing is fitted to 1 1/4-inch OD tubing with a 1/4-inch wall. The inside edge of the tubing has a 45° chamfer which results in a 7/8-inch circular seating ring for the ball at the end of the tube. The ball is attached to the tube with a continuous 360° fillet weld. The materials of construction are as follows:

Ball Material: AISI 52100

Tube Material: ASTM A519

Weld Material: UNIALOY 850

The first connector push rod found defective was subjected to metallurgical evaluation (Reference 3). The initial weld defect resulted from lack of penetration of the fillet weld with the tubing. Destructive examination of the ball and weld on the opposite end of the defective connector push rod revealed additional cracks in the heat-affected-zone (HAZ) of the ball bearing. The welds exhibited lack of penetration and slag inclusions in the crevice area behind the weld. The metallurgical evaluation concluded that the ball material is difficult to weld. The possibility of finding underbead cracks all around the ball in the HAZ is very high.

5.2 (Continued)

Previous operational experience did not indicate that the cracks would propagate out of the HAZ since the connector push rods are loaded in compression. Furthermore, none of the MP&L defects or other reported defects were associated with underbead cracking. Rather, all previous defects of this design were associated with insufficient weld penetration. Consequently, MP&L concluded that a push rod exhibiting these defects would not result in engine failure.

MP&L proceeded with an interim inspection program, until replacement connector push rods free of defects could be obtained. The discovery of a cracked connector push rod ball in the Division I diesel, however, demonstrated that the underbead cracks could, in fact, propagate through the ball material.

A new replacement push rod design (Figure 5-2) had been developed by TDI. This new design consists of a tubular steel shaft which is friction welded to cylinders of alloy steel on each end. These ends are then machine finished and hardened. The tube material is ASTM A-106 Grade B steel; the ends are AISI 8740/50 steel. During December, 1983, MP&L engineers reviewed all aspects of the push rod fabrication and observed procedural qualification runs at Delaval's push rod fabricator in Los Angeles. Samples of the qualification run were analyzed by MP&L and determined to be acceptable.

5.3 CONCLUSIONS

All intake, exhaust and connector push rods on both Unit 1 engines are the new friction welded push rods.

MP&L plans no further action on push rods; however, copies of metallurgical evaluations of the old and new push rods are being provided to the TDI D/G owners group. FaAA has performed a cyclic wear test to 10^7 cycles on a sample friction welded push rod after which it was examined metallurgically. No signs of abnormal wear or deterioration of the welded joint were observed.

A metallurgical evaluation of a connector push rod with 100 hours of operation at 100% load is also planned.

MP&L considers this problem resolved with no further action required.

FIGURE 5-1: WELDED BALL CONNECTOR

03-390-04-AB

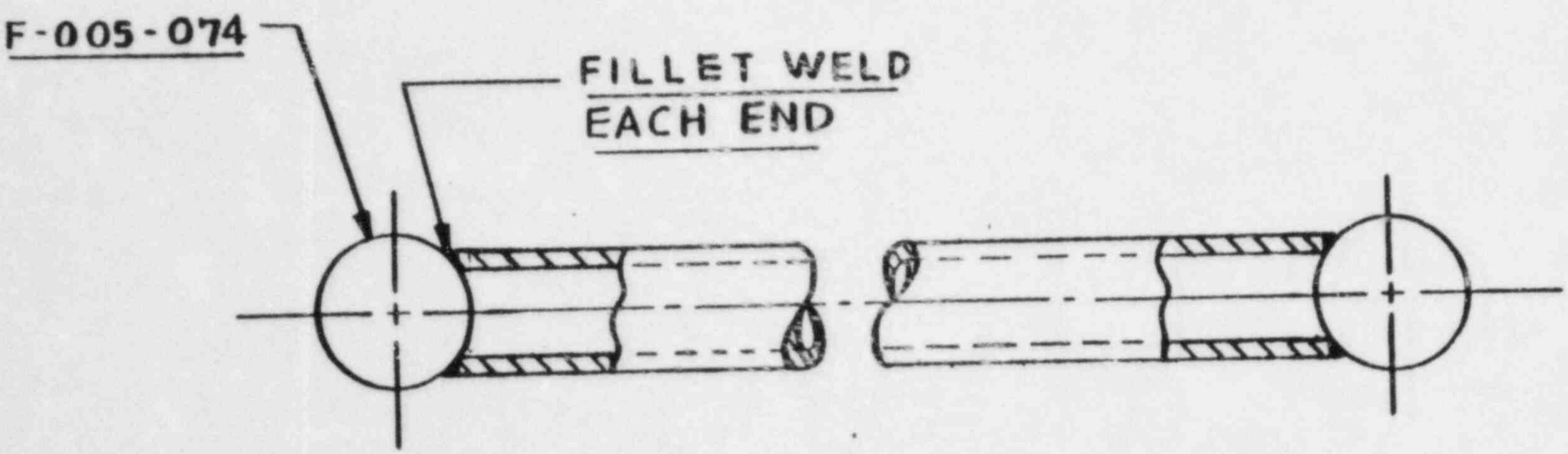


FIG. 5-1

FIGURE 5-2: FRICTION WELDED PUSH ROD

[illegible]0-07-AH

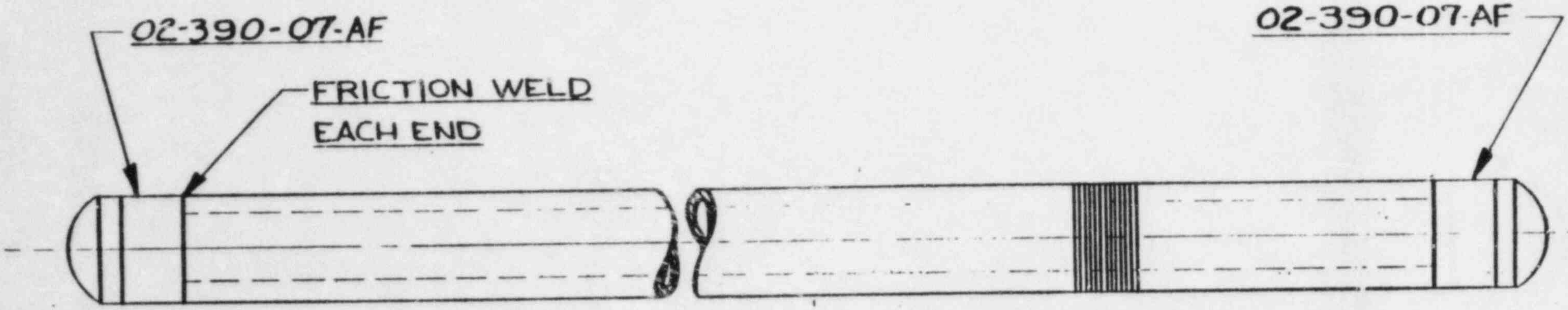


FIG. 5-2

6.0 CRANKSHAFT

6.1 DESCRIPTION

The concern over the design adequacy of the crankshaft was prompted by a crankshaft failure that occurred at Shoreham with their TDI supplied standby diesel generators. Investigation by FaAA revealed that the cause of the crankshaft failure was high cycle fatigue. This led the NRC to issue IE Information Notice No. 83-58 which identified Grand Gulf as having TDI supplied standby diesel generators with possible crankshaft design deficiencies.

6.2 ENGINEERING EVALUATION

Investigations were immediately conducted by MP&L on the applicability of the failures at Shoreham to the Grand Gulf TDI diesel generators. A physical comparison (Reference 19) of the DSR-48 (in-line eight cylinder) series engine crankshaft that failed, with that employed in the DSRV-16-4 (Vee-16 cylinder) series at Grand Gulf revealed some important differences. Among the significant design improvements on the Grand Gulf engine are the larger web size and shape, larger crankpin diameter, larger pin fillet radius and the use of counter-weights. In addition, it was learned that TDI may have used potentially non-conservative (1st generation) design harmonic coefficients in 1974 when the Shoreham stress analysis was performed.

The original GGNS design stress calculations utilized later 1975 (3rd generation) harmonic coefficient values. At the request of MP&L, TDI clarified the use of the GGNS 1975 Vs 1983 post-Shoreham 4th generation coefficients and recalculated the GGNS D/G shaft torsional stresses using the latest coefficients (Reference 4).

The changes made in the newest harmonic coefficients were an analysis refinement that resulted from analytically generated results being compared to actual test results. The changes were minor and did not substantially affect the analysis. The TDI results indicated that the GGNS crankshaft stresses are significantly less than the maximum allowable Diesel Engine Manufacturers Association (DEMA) standard and are also only $\approx$ 60% of the stresses in the failed crankshaft at Shoreham. This confirmed that a substantial design margin existed in the GGNS crankshafts.

To verify the adequacy and results of the TDI crankshaft design analysis, MP&L requested Bechtel to evaluate TDI's analytical methods. Bechtel concluded that the analytical methods used to predict crankshaft stresses by TDI are in accordance with industry standard practice and appear to be properly applied (Reference 5).

The results of the Bechtel analyses are summarized below:

- o The shaft configuration lends itself to a simple dynamic model which adds assurance to the accuracy of the calculation. The calculated first mode natural frequency has been confirmed by results of the torsiograph test, while the predicted single mode shaft stresses are within the DEMA allowables.

6.2 (Continued)

- o The harmonic coefficients, cylinder firing sequence, and engine configuration are such that the response of the major orders of critical speed are minimized. The harmonic coefficients have not been verified but it appears that a significantly detailed effort has been undertaken by TDI to provide accurate values.

- o The TDI analysis did not combine the response from the various harmonics of a given mode and of other modes to calculate total stress. However, because of the expected random phasing, the reduced effects of higher modes, the first mode stress margins compared to DEMA, and the torsiograph results, the total stress remains acceptable.

- o A comparison of the Grand Gulf and Shoreham crankshafts has been provided in Table 6-1. The improvement in the web area, fillet radius, properly applied counterweights, and shot-peened fillet radius surface finish provide for a significant reduction in stress concentrations.

- o The torsiograph results provide verification of front end angle of twist and an indication of shaft stress even though it is not a direct stress measurement. One important piece of information suggested by the torsiograph tests is that the first mode dominates the response of the crankshaft. This would tend to confirm TDI's use of first mode response to predict crankshaft stresses.

- o To address the total stress in the circular portion of the shaft, Bechtel performed an independent dynamic analysis using the normal mode method and applying modal superposition (Reference 16). Five sets of harmonic coefficients were considered in the analysis with the most important being the actual measured gas pressure values obtained from an engine of the same configuration and BMEP. The harmonic coefficients used by TDI are in good agreement with those derived from the measured gas pressure values. The results of the single order and total stress calculations are tabulated in Table 6-2 along with other crankshaft stress results for comparison.

- o It should be noted that TDI's analytical crankshaft stress determination is based on individual harmonics within a given mode. TDI did not determine the stress for a specific harmonic due to the response of all modes, or sum the effects of all harmonics and stresses from experimental shaft deflection measurements to which a theoretical deflection/stress relationship was applied. The theoretical deflection/stress relationship is based solely on the characteristics of the first mode, whereas, the measured deflection includes the response of all modes.

6.2 (Continued)

- The value of overall stress reported by FaAA for the Shoreham crankshaft represents the average stress taken over the peak stress excursion. To provide a meaningful comparison a similar average stress was computed by Bechtel for the GGNS crankshaft. An average stress is a useful value for comparison with DEMA standards since the measure of stress reversal is directly related to fatigue life. The peak stress calculated by Bechtel for GGNS crankshaft is 6034 psi. Both the peak stress and the averaged reversed stress for the GGNS crankshaft are within the limits for allowable stress published by DEMA. More importantly it should be noted that the total GGNS crankshaft stress is lower than the FaAA calculated stress for the new Shoreham crankshaft, even though the rated output of the GGNS diesel is twice that of the Shoreham diesel.

As a further verification of crankshaft adequacy, during December, 1983, and January, 1984, when the Division I and II engines were disassembled for maintenance and replacement of existing piston skirts with improved piston skirts, the Division I and II crankshafts were inspected using accepted NDE methods. No rejectable indications were discovered.

All the rod bearing journals were examined using a liquid fluorescent dye penetrant process. The entire journal surface was inspected with particular attention to the journal fillets. All linear indications were evaluated with respect to integrity. The results of the examination are shown in Table 6-3.

6.3 CONCLUSIONS

The method of analysis used by TDI has been reviewed and is in accordance with industry standard practices. Additionally the total stress values not addressed in TDI analysis have been calculated based upon measured gas pressure input and are shown to be within the DEMA limits. Liquid penetrant examination has shown no defects to exist on the journal fillets.

The total stress analysis results are lower than DEMA recommendations and when combined with acceptable liquid penetrant examinations alleviates the concerns over the design adequacy of the GGNS crankshafts.

TABLE 6-1

SHOREHAM AND GGNS CRANKSHAFT DATA

	Shoreham R Series	Grand Gulf RV Series
Web Width	21 in.	25 in.
Web Thickness	4 1/2 in.	5 1/8 in.
Web Shape	Flat Sided	Round
Crank Pin Dia.	11 in.	13 in.
Fillet Radius	1/2 in.	3/4 in.
Fillet Finish	Not Shot Peened	Shot Peened

TABLE 6-2

CRANKSHAFT STRESSES AS REPORTED BY VARIOUS ANALYSES

	Single Order Stress (psi)			Total Average Stress (psi)		
Crankshaft	Bechtel	FaAA	TDI	Bechtel	FaAA	TDI
Shoreham (11" pin)	-	5790 [15]	4570 [2]	-	8910 [15]	5314 [2]
Shoreham (12" pin)	-	3300 [17]	2990 [17]	-	5640 [17]	4208 [2]
GGNS (13" pin)	2389 [16]	-	1967 [4]	5084 [16]	-	3507 [2,4]

General Comments:

(A) DEMA limit for single order stress is 5000 psi and 7000 psi for the total stress.

(B) References to the source of information are identified in parentheses.

(C) The differences between the TDI and Bechtel calculations are primarily due to Bechtel's summation of stresses from all modes (modal superpositions) and TDI's method of including only harmonics within a single mode. For additional information see discussion in Section 6.2.

TABLE 6-3

CRANKSHAFT LIQUID PENETRANT INSPECTION RESULTS

Rod Bearing Journal Number	Inspection Results
Div I	
All Journals	Indications were present - evaluated as wear surface marks and marring caused by micrometer measurements. No apparent defects.
Div II	
#1, #2, #3 #4, #5, #6 and #8	Indications were present - evaluated as wear surface marks and marring caused by micrometer measurements. No apparent defects.
Div II	
#7	No indications present - No apparent defects

FIGURE 6-1: Crankshaft Comparisons

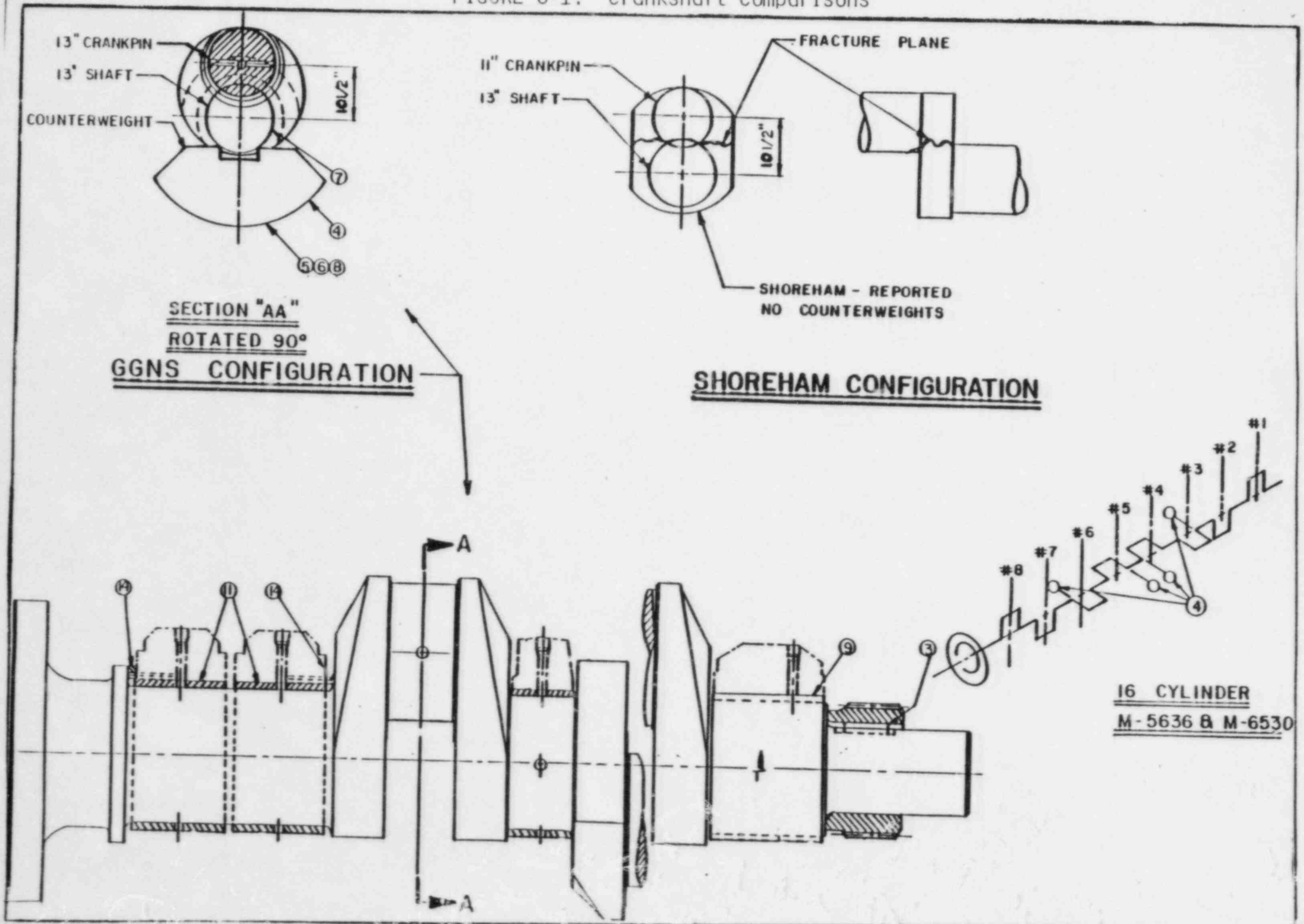


Fig. 6-1

7.0 L.P. FUEL LINE FAILURE

7.1 DESCRIPTION

On September 4, 1983, the Division I D/G was started for maintenance operation. The engine was manually stopped and the outside fresh air fans secured when a fire was reported at the engine. The fire was caused by a break in a 1-inch fuel oil supply header. The break sprayed fuel oil onto the exhaust gas piping to the left bank turbocharger. Closer examination revealed that the tubing cracked circumferentially along a line between the two ferrules of the Swagelok connector which connects the 1-inch tubing to the cross connect pipe between the the right and left bank fuel oil supply lines. The fire required extensive rework and replacement of various components.

7.2 ENGINEERING EVALUATION

Three possible causes of the tubing failure identified in the analysis by Middle South Services (Reference 6) are as follows: (1) an improper tubing material, (2) improper fitup and assembly of the tubing connector, and (3) vibration loading.

(1) The strength of a Swagelok type connection depends on controlled deformation of the tubing between the body of the connector and the front and back ferrules. Consequently, the tubing must be ductile enough to deform significantly without cracking. Swagelok recommends an ASTM A179 material. Metallurgical analysis revealed that the tubing composition, hardness and ductility were all within the specified ranges for ASTM A179 material and that the tubing was acceptable for the application. The 0.049-inch tube wall was the minimum recommended by Swagelok, but more than adequate for the operating pressures. The tubing was replaced with Delaval standard spares.

(2) A Swagelok representative from the Oakland Valve and Fitting Company, inspected the failed tubing and the associated Swagelok connector which had been sectioned for analysis. The Swagelok representative stated that the tubing had been properly deformed and that fitup or assembly problems would have been very unlikely. The Swagelok fitting was replaced with Delaval standard spare material.

(3) Vibration and fatigue were the most likely causes of the failure. The assembly of the Swagelok fitting forms a small ledge which acts as a stress concentrator. There were no supports on this section of tubing, although Delaval drawing 02-450-13 shows a clamp or support as item number 7.

(3) (Continued)

Analysis revealed no evidence of cracking at the other end of the tubing at the fuel oil filter Swagelok connection. The crack in the failed section of tubing initiated at the root of the ledge. The crack was initiated and propagated by high cycle fatigue mechanisms. This particular section of tubing had been subjected to unusual vibration loading by a defective left bank turbocharger.

The root cause of the failure was determined to be the unusual vibration loads imposed by the defective turbocharger combined with the absence of any supports to isolate the Swagelok connector from vibration loading. The defective, out-of-balance turbocharger (combined with a period of operation after the turbocharger mounting bolts were discovered to be loose) is suspected as the initiating source of vibration. The fuel oil header, to which the failed tubing section connects, is mounted to the turbocharger mounting pedestal. This turbocharger, however, was replaced prior to the ultimate failure of the tubing which resulted in the fire.

7.3 CORRECTIVE ACTIONS

MP&L designed and installed a tubing support for this section of tubing on both standby diesels. In addition, following completion of all rework related to the fire, the engine was subjected to a maintenance run to verify that all components were functioning properly. During the maintenance run, the engine was instrumented for vibration analysis. The results of the vibratory analysis revealed that the engine exhibited vibration levels which were well within the limits which could be expected from this type of machinery. These actions were described to the NRC in Reference 12.

MP&L contracted Technology for Energy Corporation (TEC) to perform vibration testing on both Unit 1 engines following the September, 1983 fire rebuild effort. During the course of this testing all piping systems in the area of the turbochargers were inspected along with most other major engine components. No areas of abnormal or suspect vibrations were reported. In conclusion, TEC stated that both engines had normal vibration levels and that no further vibration problems should be anticipated in the operation of these engines. MP&L has further committed to develop and implement a vibration monitoring program to routinely inspect both Unit 1 engines.

7.4 CONCLUSIONS

The root cause of the low pressure fuel line failure was attributed to unusual vibration loading and the absence of any supports to isolate the Swagelok connector from this loading. The corrective actions taken alleviates the loads imposed on these lines. Therefore, further failures of these lines are not expected.

8.0 H.P. FUEL LINE FAILURE

8.1 DESCRIPTION

Shoreham experienced a failure of a fuel injection line during preoperational testing. TDI filed a 10CFR21 notification on July 20, 1983 to alert the NRC to a deficiency involving a possible draw seam on the ID of the high pressure fuel injection lines supplied on TDI diesel generators. The tubing failures at Shoreham were attributed to the draw seam which acted as a stress riser and failed when subjected to repeated operating cycles (about one million cycles).

At approximately the same time of the notification, a high pressure fuel injection line on the GGNS, Unit 1, Division I diesel generator failed. An analysis of the failed tubing attributed the failure to the tubing manufacturing flaw.

8.2 ENGINEERING EVALUATION AND CORRECTIVE ACTION

All of the GGNS D/G fuel lines were original equipment, except one on each division, and had been subjected to more than ten million operating cycles. Therefore, they were considered free of defects of this type. The two lines that were not original equipment had been replaced during startup testing because of leakage around the fittings. One of these two replacement lines subsequently failed, as stated above, at approximately one million cycles.

Based on the results of an analysis performed by Middle South Services, (Reference 7), the failed tubing exhibited a crack which initiated from a manufacturing flaw on the inside surface of the tube. The flaw, which ran the entire length of the failed tubing section, was formed by a defective mandrel during the initial extrusion phase of the forming process. Additional rolling operations lapped over the flaw, which was about 6-8 mils deep. The fuel injection line operating pressure, which cycles between atmospheric pressure and about 5000 psi, provided the fatigue loading which produced cracks along the stress riser provided by the manufacturing defect. The preexisting flaw acting with the fatigue stresses generated by the cyclic operating pressures produced the failure. These evaluations and actions were described to the NRC in Reference 12.

8.3 CONCLUSIONS

The TDI 10CFR21 notification indicates that the failures occur at approximately one million operating cycles and that fuel lines that have in excess of ten million operating cycles without failure are satisfactory. All of the original lines on the Division I and II diesels were, therefore, considered free of internal flaws of this type because they have in excess of ten million operating cycles and have not failed. One line on the Division I diesel, the one that failed and was replaced, and one line on the Division II diesel were not original lines and were considered suspect. Replacement lines were ordered and installed in place of these two lines.

8.3 (Continued)

TDI has inspected the fuel line material used for the new MP&L lines using a sampling technique where a 1-½ inch long portion is cut from each end of each 17-20 foot long stock tube. These short sections are split axially by saw cut, and the bore surfaces checked for draw deficiencies. The basic assumption of the inspection is that any deficiencies in the tubing would exist throughout the entire length. If there are no imperfections found in the end pieces, then there are none in between, and the tube is considered acceptable. The new MP&L lines successfully passed the TDI inspection.

This problem is, therefore, considered resolved for the GGNS Unit 1, TDI diesels.

9.0 CRANKCASE CAPSCREWS

9.1 DESCRIPTION

During the performance of a 24 hour run test on March 15, 1982, the Division II D/G tripped on a "Generator Differential" which was accompanied by an observed electrical arcing flash inside the generator. In a subsequent inspection of the generator it was found that the stator insulation had been damaged and that a 15/16 inch capscrew head from a 5/8 UNC X 1-3/4 inch long capscrew had imbedded in the stator and damaged the generator. It was determined that the capscrew head was from a capscrew on the diesel's rear crankcase cover that had sheared off and entered the generator through the air gap on the end of the generator. The generator was replaced with a generator from Unit 2 and all rear crankcase cover capscrews on the Unit 1, Division I and II diesels, were replaced with new replacement capscrews.

An independent lab performed an analysis (Reference 9) of the 42 capscrews removed from the Unit 1, Division I and II diesel generators. A review of the analysis produced the conclusion that the failure mode was due to a low-stress fatigue front expanding from an initial small crack. It was also noted that the failed capscrews had a decarburized skin which may have contributed to the failure.

On October 4, 1982, the rear crankcase cover capscrews were checked for the correct tightness (60 ft-lbs). Three of the capscrews on the Division II diesel generator were found to be less than 60 ft-lbs (20, 23 and 35 ft-lbs). Any capscrew not within ± 2 ft-lbs of the 60 ft-lbs was to be torqued to within the acceptable range. When the capscrew that was found at 20 ft-lbs was tightened, it sheared off approximately one inch from the bottom side of the head before reaching 60 ft-lbs.

9.2 INSPECTION AND TESTING

The Division II D/G was instrumented by Nutech in January of 1983 and data was obtained during an operational test run. The test data indicated that the highest vibration amplitude occurred during the startup and shutdown of the diesel, with capscrew stresses at 6000 psi. The vibration amplitude was much less during steady state operation at 450 RPM, with the capscrew stresses at 3000 psi. However, the test results were inconclusive as to the root causes of the vibration source. The present information indicates that the capscrews failed by a combination of metallurgical and transient vibration factors and that the failures are unique to the Division II D/G.

9.3 CORRECTIVE ACTIONS

The main thrust of the corrective action taken was the design and installation of protective screens for the generator air gaps. The failure of the rear crankcase cover capscrew, by itself, would not prevent the diesel from performing its safety function. On the other hand, the entry of foreign material into the generator could cause failure, therefore, the screens were installed to protect against a similar mode of failure. At the same time fatigue resistant, high strength capscrews and tab washers were installed to extend the life of these capscrews. One of these capscrews was pulled from each division and subjected to destructive analysis.

While there was no sign of crack initiation there were signs of fretting on the threads of the capscrew removed from the Division II D/G. The expected life of these capscrews has not been confirmed. After the metallurgical report is evaluated, the schedule for removing another bolt from the Division II D/G for analysis will be determined.

9.4 CONCLUSIONS

Although MP&L is continuing to inspect the crankcase cover capscrews and isolate the source of cyclic loading, the possibility of failure of one of these capscrews no longer poses a threat to diesel generator operability due to the installation of protective screens on the generator air gaps.

10.0 TDI PRODUCT IMPROVEMENTS

TDI has a product improvement program which addresses both changes that are required to ensure diesel generator operability/reliability and changes that are developed to extend component life, allow easier maintenance operations, or use improved manufacturing techniques. The TDI program classifies changes as follows: (1) changes required to correct 10CFR21 deficiencies, (2) changes developed to improve diesel generator performance or reliability (not as a result of a potential defect) and issued to customers under TDI's Service Information Memo (SIM) program, and (3) changes developed by TDI that are determined by TDI to be relatively insignificant to diesel generator operation and therefore do not necessitate immediate customer notification.

The TDI program of product improvement has included applicability reviews for the diesel generators installed at Grand Gulf and the applicable changes have been identified to MP&L (Reference 2). The TDI Nuclear Check List for SIMS identifies those that are applicable to TDI diesels at nuclear stations. The thirty-three SIMs identified by the list were reviewed by MP&L to determine which SIMs could be considered product improvements. Four categories; product improvement, instructions, information and guidelines were utilized for the review. Eight of the thirty-three SIMs reviewed were considered to be product improvements, nine SIMs as recommended instructions, ten SIMs as informational and six SIMs as guidelines. A listing of the eight SIMs considered product improvements is provided in Table 10-1. Review of the vendor manual for the TDI diesels and other documents indicates that the eight product improvement SIMs have been incorporated on the Unit 1, Division I and II diesel generators.

A continuing review will be performed for TDI SIMs as they are received to determine their applicability to the GGNS TDI diesels and appropriate actions taken, as deemed necessary.

TABLE 10-1

TDI PRODUCT IMPROVEMENT SIMS

SIM NO.	SUBJECT
64	1. Increase link rod torque - 735 to 1050 ft/lbs 2. Increase rod bolt torque - 1 1/2 in bolt 1200 to 1700 ft/lbs 1 7/8 in bolt 1800 to 2600 ft/lbs 3. Product improvement designed to increase reliability 4. Deletes SIM 270 5. Use in conjunction with SIM 332 6. Incorporated during "AF" piston skirt modification in November, 1981
307	1. Change in ring end gaps on new piston rings in 4 valve R & RV engines 2. Incorporated
313	1. Information on removing intake manifold supports on 4 valve RV engines to reduce oil leakage at the camshaft covers 2. Incorporated
324	1. Modification of type "AF" piston skirt 2. Incorporated on Unit 1, Unit 2 "AF" piston skirts have not been modified 3. The modified "AF" piston skirts have been replaced with the "AE" style piston skirts on the Unit 1, Division I and II D/Gs
324A	1. Information for reuse of piston crown studs 2. Incorporated
332	1. Newer harder washers on connecting rod bolts RV engines 2. Incorporated
360	1. Information on possible problem of air start valve capscrews being too long 2. Incorporated on Unit 1, Div I and II, tracking document issued for Unit 2
361	1. Information on potential problem with commercial grade cable in certain engines and panels 2. Incorporated on Unit 1, Division I and II engines - Cable replaced with Class 1E qualified cable, tracking document issued for Unit 2

11.0 QUALIFICATION/RELIABILITY DEMONSTRATION TESTING

11.1 HISTORY

All the GGNS Unit 1 diesel generators have been tested and qualified in accordance with the requirements of Regulatory Guides 1.9 and 1.108 and IEEE Std. 387-1977. The Division I and Division II engines were shop tested by TDI, including a 300 prototype test run on the Division I engine as required by IEEE 387-1977. On-site testing was done by Bechtel and MP&L before fuel loading in June, 1982. Since then the engines have been tested in accordance with the plant surveillance test procedures, as described in the plant technical specifications.

Augmented testing such as a 7-day performance run was performed on both of the TDI engines under a directive of MP&L management (Reference 10) before the present maintenance and parts replacement work was started in December, 1983.

To verify the operability and reliability of the Division I D/G following the D/G rework after the fire, the 18 month functional test was repeated for the Division I D/G. This additional 18 month functional test included the following:

1. Starting air receiver capacity test
2. Testing of D/G trips and response to ECCS actuation signals
3. 100% load rejection
4. Simulated loss of offsite power followed by the loss of and restart of the D/G
5. Simulated loss of offsite power in conjunction with ECCS actuation signals
6. 24 hour load test
7. LOP/LOCA test

This additional 18 month functional test was completed satisfactorily.

The following sections outline the recent tests that were performed on the Division I and II diesels following completion of maintenance work, before each of the two engines was returned to service.

11.2 REQUALIFICATION TESTING REQUIREMENTS

Testing requirements for modifications to a previously qualified diesel generator unit are set-forth in IEEE Std. 387-1977. The recent maintenance and parts replacement work on the two TDI diesels had no significant impact on engine specifications and design criteria, related subsystems, or engine performance characteristics. Nor, did these work activities involve changes in plant load characteristics

11.2 (Continued)

for the two TDI engines. No modification of the generator or related electric or instrumentation circuitry was performed. Therefore, none of the design considerations listed in Table-1 of IEEE Std. 387-1977 were modified or altered. As such, the various tasks performed during the current maintenance activities were considered minor design changes as defined by IEEE 387-1977 criteria. Appropriate testing was conducted to verify satisfactory operability of the engines.

11.3 REQUALIFICATION/DEMONSTRATION TESTING FOLLOWING PISTON SKIRT REPLACEMENT

The requalification testing is described in the following section.

11.3.1 To perform TDI's recommended breakin run, following the installation of the "AE" piston skirts, the engines were started and run at 300 rpm and no load for about 15 minutes. During this run the D/Gs were inspected to ensure that the rocker arms, valves, push rods, fuel injection pumps, nozzle holders, high pressure fuel injection lines and drip return headers were secure, functioning properly and that there were no fuel leaks. The engines were then stopped, the crankcase side door covers removed and various internal components checked for indication of excessive heat. The covers were replaced and the engines run at 20% load for about one hour. After this run the engines were inspected as above. The engines were then run at levels varying between 25% to 100% load for approximately 8 hours. After this run hot crankshaft web deflection checks were performed. The engines were then allowed to cool and another inspection as above was performed.

11.3.2 The load rejection tests were accomplished by performing Test #3 of Surveillance Procedure Nos. 06-OP-1P75-R-0003 and 06-OP-1P75-R-0004 "Standby Diesel Generator (SDG) 11 (12) 18 Month Functional Test". These tests demonstrated the capability to reject a full load (7000 kw) without exceeding speeds or voltages which could cause tripping, mechanical damage, or harmful overstresses.

11.3.3 In addition to the required testing, 24 hour run tests were performed; 2 hours at 110% load followed by 22 hours at 100% load. These tests demonstrated the capability of the D/G to carry the rated load for an extended period.

11.3.4 The starting, load acceptance and design load tests were accomplished by performing Surveillance Procedure Nos. 06-OP-1P75-M-0001 and 06-OP-1P75-M-0002, "Standby Diesel Generator (SDG) 11 (12) Functional Test". These tests demonstrated the ability of the D/G to start and reach rated frequency and voltage within 10 seconds after the start signal, the capability to be loaded to at least 100% load within 60 seconds and to operate for at least one hour at full load.

11.4 TESTING NOT REQUIRED FOR REQUALIFICATION

The main consideration in developing the requalification test program described in Section 11.3 above was that any engine component or subsystem that was replaced, modified or reworked would be adequately tested, followed by an integrated testing of the total diesel generator system. Accordingly, an engine component or subsystem that was not affected by the maintenance activities and was previously qualified, was not tested individually or in conjunction with engine testing.

11.5 D/G RELIABILITY ENHANCEMENT TESTING

MP&L has developed comprehensive maintenance programs and established operating practices to assure a high level of diesel generator reliability. This program was developed using vendor recommendations as well as good engineering practice and operating experience. This program covers the diesel generator as well as its supportive equipment.

Critical diesel generator parameters such as jacket water temperature, lube oil temperature, jacket water standpipe level, generator bearing oil level, turbocharger lube oil flow, starting air pressure, heater operation, and alarm checks are performed once per 8 hours; while other various supportive equipment is checked once per day by the Operations Department. These checks will assure that the diesel generators are in a satisfactory state, and that potential problems are identified.

During the monthly surveillance test, operating parameters are checked to verify that the diesel generator is operating as required. The generator operating parameters monitored are voltage, amperes, frequency, VARS, DC volts-field, DC Amps-field, RPM and watts. The engine operating parameters monitored are lube oil temperature and pressure, jacket water temperature and pressure, turbocharger lube oil pressure, lube oil filter differential pressure, fuel oil pressure, fuel oil filter differential pressure, combustion air pressure, crankcase vacuum, RPM cylinder temperatures, and exhaust stack temperatures. The monitoring of these parameters aids in detecting any problems which would affect engine operation and reliability.

11.6 ADDITIONAL DEMONSTRATION TESTS

Since the discovery of the failed crankshafts at Shoreham, additional testing/monitoring of the D/Gs at Grand Gulf has been implemented (Table [illegible]). This includes the completion of a 7-day equivalent test run on both D/G units, 24 hour run tests (22 hours @ 100 percent and 2 hours @ 110 percent power), monitoring of vibration levels by Technology for Energy Corporation (Reference 8), increased emphasis on pre-action planning sessions for persons involved in planned operational and maintenance activities and an improvement in the working relationship with TDI (Reference 10 and 11).

11.6 (Continued)

Additional testing also includes the completion of a 100 hour run on the GGNS Unit 1 D/Gs. The Division II D/G performed well during a 100 hour 100% load run. As a precautionary measure, the Division I D/G was shutdown 21 hours into the 100 hour 100% load run when a maintenance inspection revealed that two bolts were missing from the left bank turbocharger. The bolts were replaced and the 100 hour run restarted. Seventy-two (72) hours into the run, three bolts were again discovered broken or missing from the left bank turbocharger and as a precautionary measure the engine was shutdown. Following the second incident, an extensive maintenance effort was undertaken to ensure that the left bank turbocharger was aligned correctly.

Following this effort the Division I D/G performed well during another 100 run with 32 hours at 100% load and 68 hours at 75% load. The loads carried by the D/Gs during these runs are substantially larger than those that could be experienced during emergency engine service. During a loss of power event the Division I loads are 52% of the D/G full load rating and Division II 68% of the D/G full load rating.

The successful completion of these runs following the correction of the alignment of the Division I left bank turbocharger demonstrates that the GGNS TDI D/Gs will reliably carry loads substantially larger than those experienced during emergency engine service.

TABLE 11-1

SUMMARY OF QUALIFICATION AND VALIDATION TESTING

TESTING PRIOR TO INSTALLATION OF "AE" PISTON SKIRTS	REQ	ADDITIONAL DEMONSTRATION SURVEILLANCES
Qualification Testing (1)	X	
Preop Testing (2)	X	
Tech Spec Testing (3)	X	
18 Month Functional Test, Division I D/G		X
7-Day Equivalent Test		X
Vibration Test Runs		X
"AE" PISTON SKIRT INSTALLATION		
Piston Inspection		X
Crankshaft Inspection		X
Rod Bearing Inspection		X
Cylinder Head Inspection		X
TESTING FOLLOWING INSTALLATION OF "AE" PISTON SKIRTS		
Break-In Run	X	
Load Rejection	X	
24-Hour Run - 2 Hr @ 110%, 22 Hr @ 100%		X
Monthly Surveillance	X	
Additional 100% Power Runs (Div I 101 Hrs, Div II 100 Hrs)		X

(1) Qualification Testing includes 300 Start Prototype Tests Performed by TDI.

(2) Includes Starting, Load Acceptance, Overload, Design Load, Rejection, Reliability, Electrical and Subsystem Tests.

(3) Includes Monthly Surveillance and 18 Month Functional Tests.

12.0 SUMMARY

Specific actions have been taken to correct problems identified during testing of the Division I and II TDI diesel generators and to also evaluate and resolve problems identified to MP&L as a result of experience with TDI diesel generators at other nuclear installations. Significant actions that have been completed, or are planned, are as follows.

- The suspect modified type "AF" piston skirts in the Division I and II D/G have been replaced with new type "AE" pistons. The new type "AE" piston skirts were inspected prior to installation to assure they were free of the type of rejectable indications found on the type "AF" piston skirts and to establish documented baseline data for the new skirt. The FaAA report (Reference 21) for the TDI D/G Owners Group on "AF" and "AE" piston skirts concluded that, based on both analysis and test results, the type "AE" skirt attachment would not fail in fatigue.

- During removal of cylinder heads on the Division II D/G the stellite overlays on the exhaust valve seats on the #5 right bank cylinder were discovered to have cracks. There was also incomplete fusion on the intake valve seat of the #1 left bank head. Inspection of the Division I heads found six with rejectable indications. The eight heads with rejectable indications were replaced with heads that had no rejectable indications. To address a long term concern, a failure investigation has been initiated to determine the cause of the crack initiation and the crack propagation mode.

- As a result of the connecting rod bearing failure identified at Shoreham, MP&L initiated an inspection of the connecting rod bearings and connecting rods during the scheduled piston replacement on the Division II D/G.

  The inspection results indicate that the integrity of the bearings is good and not affected by previous service. A final analysis for chemical and physical properties is planned.

  The FaAA report (Reference 22) for the TDI D/G Owners Group conservatively concluded that the 12 and 13 inch bearing shell fatigue life should be approximately 38,000 hours at full load. This is over 10 times the useage expected over the 40 year service life of the nuclear standby diesel generators.

- Numerous weld failures between the D/G connector push rod ball and tube have been discovered. MP&L concluded that it was not likely that a failed push rod would result in engine failure. However, during a recent inspection one of the push rod balls was cracked in addition to the weld cracks. At this point a new replacement design was pursued. The new design has been determined to be acceptable by MP&L and replacement connecting and main push rods have been installed in the Division I and II diesels.

12.0 (Continued)

- o Due to the crankshaft failure at Shoreham an engineering evaluation of differences in design between the Shoreham and GGNS TDI diesel crankshafts was performed. This evaluation shows that the potential for the type of failure experienced at Shoreham does not exist at GGNS. During the piston skirt replacement the Division I and II crankshafts were inspected. These inspections did not indicate defects of the type found at Shoreham.

- o A fire in the Division I D/G room on September 4, 1983 was determined to be caused by the break of a low pressure fuel oil line. Analysis indicated that the line break was caused by a combination of unusual vibration loads imposed by a turbocharger that had been replaced several weeks before the fire, and the absence of any supports to isolate the Swagelok connector from vibration loads. Tubing supports were designed by MP&L and installed on the Division I and II diesel-generators.

- o A 10CFR21 notification to the NRC by TDI dated July 20, 1983 identified a possible draw seam on the ID of high pressure fuel oil lines supplied on the Division I and II D/Gs. A high pressure fuel oil line on the Division I D/G also similarily failed. An analysis of the failed tubing attributed the failure to a manufacturing flaw (draw seam) in the tubing. The TDI letter of July 20, 1983 indicated that the failure occurred at approximately one million operating cycles and that fuel lines that have in excess of ten million operating cycles without failure are acceptable. Using this rationale, all of the original lines on the Division I and II D/Gs were considered to be free of flaws of this type, however, the replacement for the failed line and one line on the Division II diesel were not original lines and were considered suspect. Replacement lines were ordered and installed in place of the two suspect lines.

- o The generator on the Division II D/G was damaged and replaced in mid-year of 1982 when a head from a capscrew on the rear crankcase cover sheared off and entered the generator via the generator air gap. Protective screens have been installed on the Division I and II generator air gaps to prevent recurrence of damage to the generator from an incident of this type. Subsequent testing indicated that the problem of the capscrew shearing was unique to the Division II D/G and that the failure was due to low stress high cycle fatigue, however, test results were inconclusive as to the root causes of the vibration sources. High strength capscrews and tab washers were installed to extend the life of the capscrews. Periodically a capscrew will be removed from the crankcase covers and subjected to destructive analysis in an attempt to obtain further information for identifying the root cause.

12.0 (Continued)

- o Following piston skirt replacement, qualification/reliability testing in accordance with IEEE Std. 387-1977 was performed on the Division I and II diesel generators. Testing of the D/Gs prior to this maintenance included the satisfactory completion of a 7-day equivalent test run on both D/Gs. Post maintenance testing included breakin runs, twenty-four hour runs, load rejection tests and surveillance tests. Additional post maintenance demonstration testing has resulted in approximately 158 hours at 100% load on the Division I diesel generator and 131 hours at 100% load on the Division II diesel generator.

13.0 CONCLUSION

In conclusion, the specific corrective actions, engineering evaluations and testing that have been completed, enhance the reliability of the D/Gs and provide assurance, with a reasonable level of confidence, that the GGNS TDI engines will adequately perform their required safety function.

14.0 REFERENCES

1. FaAA Preliminary Report on GGNS Modified AF Piston Skirts.

2. TDI Letter Dated 12-15-83 "Nuclear Power Plant Standby Diesel Generator User's Group Minutes of November 30, 1983, Meeting".

3. Metallurgical Evaluation of Diesel Engine Push Rod Weld From Grand Gulf Nuclear Station-Unit 1 Emergency Diesel Generator (Division I), prepared by Middle South Services.

4. TDI Response to MP&L for NRC Request of Additional Information on TDI D/Gs, Dated November 2, 1984.

5. Preliminary Standby Diesel Generator Crankshaft Design Analysis Review Grand Gulf Nuclear Station, prepared by Bechtel Power Corporation.

6. Metallurgical Evaluation of Diesel Engine Fuel Oil Line Failure from Emergency Diesel Generator - Division 1, Grand Gulf Nuclear Station - Unit 1, prepared by Middle South Services.

7. Metallurgical Evaluation of Diesel Engine Fuel Injection Tube from Grand Gulf Nuclear Station - Unit 1 Emergency Diesel Generator Prepared by Middle South Services.

8. Test Evaluation Report on the Grand Gulf Nuclear Station Division I and Division II Diesel Generators (TEC Report No. R-83-033), prepared by Technology for Energy Corporation.

9. Engineering Investigation of the Failure of Rear Crankcase Cover Capscrews for the Delaval Standby Diesel Generators at MP&L, GGNS, LETCO Job No. G-8847, Dated August 17, 1982, by Law Engineering Testing Company.

10. PMI 83/12569, J. P. McGaughy to J. B. Richard Letter on D/G Enhancement.

11. PMI 84/0210, J. E. Cross to J. F. Pinto Letter on Plant Staff Response to NRC D/G Questions.

12. AECM-83/0689 - GGNS Diesel Generator Reliability Report, October 26, 1983.

13. AECM-83/0724, GGNS Diesel Generator - NRC Request for Additional Information, November 15, 1983.

14. AECM-84/0030, GGNS Diesel Generator - NRC Request for Additional Information, January 18, 1984.

15. FaAA-83-10-2 PA07396 - Emergency D/G Crankshaft Total Stress Analysis Summary, February 2, 1984.

14.0 (Continued)

16. Bechtel Standby Diesel Generator Crankshaft Total Stress Analysis Summary, February 2, 1984.

17. FaAA-83-10-02 PA07396, Analysis of the Replacement Crankshafts for Emergency Diesel Generators, Shoreham Nuclear Power Station, October 31, 1983.

18. FaAA-83-10-16, PA07396, Emergency Diesel Generator Connecting Rod Bearing Failure Investigation Shoreham Nuclear Power Station, October 31, 1983.

19. AECM-83/0653, Applicability of Shoreham Diesel Generator Crankshaft Failure to GGNS, October 14, 1983.

20. AECM-84/0103, GGNS Standby Diesel Generator, Comprehensive Reliability Report and Status, February 20, 1983.

21. FaAA-84-2-14, FME-R-6/7389, Investigation of Type AF and AE Piston Skirts, February 27, 1983.

22. FaAA-84-3-1, PAO 7389/LAS-M&T-3A, Design Review of Connecting Rod Bearing Shells For Transamerica DeLaval Enterprise Engines, March 12, 1984.

ATTACHMENT 1 TO THE
UPDATED REPORT ON GGNS
DIVISION I AND II TDI
DIESEL GENERATORS

RESPONSES TO SIXTEEN POTENTIALLY SIGNIFICANT PROBLEMS
IDENTIFIED IN TDI OWNERS GROUP
MEETING WITH THE NRC ON
JANUARY 26, 1984

April 1984

Z3rg52

1.0 INTRODUCTION

A meeting of the Transamerica Delaval, Inc. (TDI) diesel generator (D/G) owners group with the NRC Staff was held on January 26, 1984. During the meeting the owners group presented a slide summarizing significant potential problems with TDI diesels. These potential problem areas are detailed below:

- o Crankshaft
- o Connecting Rod Bearings
- o Pistons
- o Cylinder Heads
- o Cylinder Liners
- o Cylinder Block
- o Engine Base
- o Head Studs
- o Push Rods
- o Rocker Arm Capscrews
- o Connecting Rods
- o Electrical Cable
- o Fuel Injection Lines
- o Turbocharger
- o Jacket Water Pumps
- o Air Start Valve Capscrews

Further details of these concerns, their applicability to Grand Gulf, and their resolution are described in the following sections.

2.0 CRANKSHAFT

A summary of the concern and its resolution on Grand Gulf is provided in Section 6.0 of the Final Report.

3.0 CONNECTING ROD BEARINGS

A summary of the concern and its resolution on Grand Gulf is provided in Section 4.0 of the Final Report.

4.0 PISTONS

A summary of the concern and its resolution on Grand Gulf is provided in Section 2.0 of the Final Report.

5.0 CYLINDER HEADS

A summary of the concern and its resolution on Grand Gulf is provided in Section 3.0 of the Final Report.

6.0 CYLINDER LINERS

6.1 DESCRIPTION

A concern has been raised regarding cylinder liner damage in TDI D/Gs. One incident was listed for GGNS, AECM-82/157, dated April 15, 1982, which transmitted the final report on PRD-81/45 dealing with the separation of piston crown from the piston skirt during testing of the Division II D/G. An additional deficiency noted in this report was damage to a cylinder liner on the Division I D/G. The damaged cylinder liner was discovered during disassembly of the Division I D/G for corrective action for the piston skirt/crown separation.

The damaged Division I cylinder liner was found to be grooved in three places. These grooves were approximately 10 inches long and 1/16 inch deep. As indicated in the PRD final report, the grooving was probably caused by debris that entered the cylinder during assembly or initial startup.

6.2 ENGINEERING EVALUATION AND CORRECTIVE ACTION

The grooved cylinder liner was replaced with a new liner. The Division I lube oil was flushed and replaced and the lube oil sump was cleaned.

At a meeting between MP&L, LILCO and TDI on February 2, 1984, TDI indicated that the only case of a cylinder liner failure occurring without some other initiating event causing it, occurred on the ship

6.2 (Continued)

Columbia. This damage was attributed to the high vanadium content of the light-heavy fuel oil and the high ash content of the lube oil (heavy oil). Despite the cracking of the liner which resulted from the use of these oils, the engine continued to perform its function. The GGNS TDI diesels use light fuel oil with a lower vanadium content and utilize light lube oil.

During the recent piston skirt changeout on the GGNS Unit 1 TDI engines, the cylinder liners were subjected to a close visual inspection before and after honing the liners to receive the new rings. No obvious damage was discovered during these inspections.

6.3 CONCLUSIONS

Based on lube oil cleanup efforts, recurrences of the subject problem is considered to be resolved. To date, no known cylinder liner failure has been the root cause of a TDI engine failure.

Neither liner material, manufacturing process nor design are considered to be the root cause of the damage on the Division I GGNS engine.

Inspections of the Division I and II D/G cylinder liners during the recent piston skirt changeout in December, 1983 did not reveal any indication of liner damage.

Based on the above conclusions and root cause, cylinder liner failure is not expected to occur at GGNS.

7.0 CYLINDER BLOCK

7.1 DESCRIPTION

The non-nuclear industry has reported cracks occurring in the area around the cylinder liner landing. Cracks may also propagate from the head stud/stud bore to the jacket cooling water passage. MP&L has also been recently advised of the discovery of cracks in the area of engine head studs on a cylinder block used in a nuclear application.

7.2 ENGINEERING EVALUATION

If this cracking were to occur and propagate into the jacket water passage it would be possible for an extremely low flow of jacket cooling water to come into contact with the head studs and cylinder head. This flow would be prevented from entering the cylinder by two spiral wound head gaskets. It is unlikely that jacket cooling water would enter the firing chamber (cylinder) and only a very slow loss of jacket cooling water to the outside of the engine would be evident.

To prevent this cracking, TDI has indicated that proper torque must be placed on the cylinder head studs.

7.3 CORRECTIVE ACTION AND CONCLUSIONS

MP&L considers that no corrective action is required for these conditions, since the postulated condition would not interfere with the operation of the engine and because the proper torque of 3600 foot-pounds, as recommended by TDI, has been applied to the head studs of the GGNS Unit 1 TDI D/Gs. Even if cracks were to occur they would be expected to propagate very slowly because of the large mass of metal in the cylinder block. However, MP&L will continue to closely follow the investigation findings of the TDI D/G owners group.

8.0 ENGINE BASE

8.1 DESCRIPTION

Linear indications have been found on the bearing base journal of several marine diesel engines. These indications were apparently caused by improper torquing of the bearing holddown studs during assembly of the engine.

8.2 CORRECTIVE ACTION

TDI issued SIM #286 to correct this problem. This fix resulted in an increased preload being placed on the holddown studs.

8.3 CONCLUSIONS

Grand Gulf's TDI D/Gs were assembled after SIM #286 was issued. GGNS installation of main bearing bolt nuts, as witnessed by GGNS Plant Quality, indicate that correct preload values were verified during recent engine disassembly at the site on all main bearing studs. This problem, therefore, is not expected to occur at Grand Gulf since no defects have been reported to have occurred in engines using the proper torque.

9.0 HEAD STUDS

This concern is related to the cylinder block concern described in Section 7.0 of this Attachment. Refer to this section for further details.

10.0 PUSH RODS

A summary of the concern and its resolution on Grand Gulf is provided in Section 5.0 of the Final Report.

11.0 ROCKER ARM CAPSCREWS

11.1 DESCRIPTION

Shoreham has experienced problems recently with fatigue failure of a rocker arm capscrew.

11.2 ENGINEERING EVALUATION AND CORRECTIVE ACTION

The failure at Shoreham was apparently caused by undertorqued poor quality capscrews. New capscrews made of ASTM A-193 material were installed and torqued to specified torque values to correct the problem at Shoreham.

11.3 CONCLUSIONS

GGNS rocker arm capscrews have not experienced this type of failure after greater than 10^7 cycles of operation. The "Emergency Diesel Generator Rocker Arm Capscrew Stress Analysis" report, dated March, 1984 prepared for the TDI Owners Group by Stone and Webster Engineering Corporation, concluded that both the original and modified rocker arm capscrews are adequately designed for the given service conditions. The GGNS rocker arm capscrews are original components of the diesels and have been properly torqued to 365 ft-lbs. MP&L considers this issue resolved with no further action required.

12.0 CONNECTING RODS

12.1 DESCRIPTION

TDI has informed MP&L of several incidences of connecting rod failure. At a meeting between MP&L, LILCO and TDI, on February 2, 1984, TDI defined the historical problem with connecting rods. Cracking of the connecting rod link assembly in a master rod-longitudinal plane through the bottom of upper bolt holes (See Figure A12-1) has occurred on several non-nuclear applied diesel engines built by TDI.

12.2 ENGINEERING EVALUATION AND CORRECTIVE ACTION

TDI Vee-type engines of a comparable size to GGNS Division I and II utilize either 1-1/2 or 1-7/8 inch connecting rod bolts. The original design of the GGNS engines (TDI's earlier design) uses the larger of the two bolt sizes. TDI originally specified that these bolts should be torqued to 1800 foot-pounds.

TDI initiated an evaluation of the problem based on the operating history of the engines with failed or cracked connecting rods. For example, several instances of connecting rod cracking were reported to have occurred in a marine diesel on the ship Columbia. The average hours of operation between occurrence was approximately 10,000 hours. Evidence of fretting in the "rack-teeth" almost always accompanied connecting rod failure or cracking.

The first design change to remedy the situation was a decrease in the connecting rod bolt diameter to 1-1/2 inches. Decreasing the connecting rod bolt diameter effectively increased the amount of base metal where cracking was occurring. Since the cause of the cracking was thought to be relative motion between the rod parts and flexure of connecting rod parts, an increase in the base metal adjacent to the crack initiation site should increase stiffness and hence decrease incidence of cracking or failure.

A decrease in cracking frequency was noted. However, connecting rods using both 1-1/2 and 1-7/8 inch bolts were still reported exhibiting cracking. It was then thought that fretting of the "rack-teeth" was due to lack of clamping force between the connecting rod link and the master rod and box assembly. TDI issued Service Information Memo (SIM) 64 to rectify the suspected clamping force problem. SIM 64 effectively increases the required torque on 1-1/2 and 1-7/8 inch connecting rod bolts from 1200 to 1700 foot-pounds and from 1800 to 2600 foot-pounds, respectively. This design change greatly reduced the reported cases of connecting rod cracking.

The GGNS Division I and II engines were originally assembled at the vendor's shop using the pre-SIM 64 torque values. Therefore, the GGNS engines have been run part of the present total sum times with the connecting rods torqued to pre-SIM 64 torque values and the balance at post SIM 64 torque values. The table below indicates the approximate run times on the Division I and II engines before and after SIM 64 was implemented:

12.2 (Continued)

	Division I	Division II
At Assembly	0	0
Before SIM 64	332	44
After SIM 64	1065	826
Present Run Times	1397	870

At a recent TDI D/G owners group component selection committee meeting, the owners group diesel generator specialists agreed that the type of cracks reported by TDI would propagate very slowly. The cracking of connecting rod parts on non-nuclear diesel engines were reported to have occurred at relatively large run times (greater than 10,000 hours).

12.3 CONCLUSIONS

To date, all engines using the 1-7/8 inch connecting rod bolts exhibiting failures or cracking have been suspected of being undertorqued. Further, no known failures have occurred on connecting rods using 1-7/8 inch bolts that were properly torqued. All torques used on the subject bolts at GGNS have been verified to be in accordance with SIM 64.

Based on low probable propagation rate of incipient cracks, relatively low run hours on Division I and II at pre-SIM 64 torques, and the expected low future run times, (estimated 200 hours/year) deleterious cracking of the GGNS connecting rods is not expected.

FIGURE A12-1: Articulated Connecting Rod Assembly

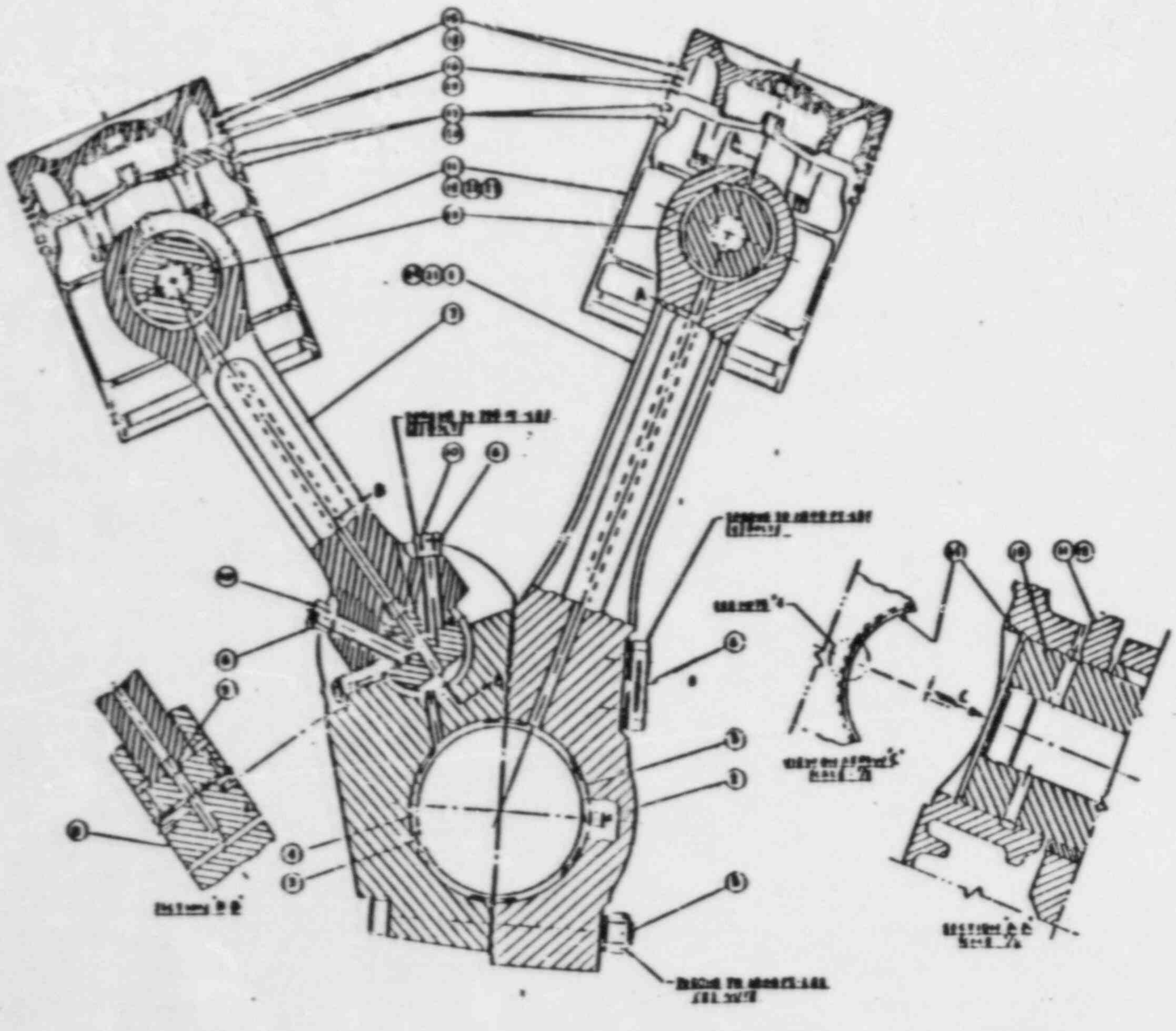


Fig. A12-1

13.0 ELECTRICAL CABLE

13.1 DESCRIPTION

Memo (SIM) No. 361 concerning certain Class 1E cable which failed the IEEE 383-1974 insulation flame test was issued by TDI. The content of this SIM is detailed in Table 10-1. This SIM identified the affected cable as being the shielded cable from the terminal block to the Airpax tachometer relay in the engine control panel, the shielded cable from the Airpax magnetic pickups to the junction boxes on the side of the engine and the multiconductor cable from the engine side mounted junction box to the Woodward governor actuator.

Another notification from Delaval received by MP&L on October 20, 1983 (API-83/0974), idicated that the manufacturer's temperature rating for the cable insulation may be exceeded during operation of the diesel generator. Delaval recommended that these cables be replaced with 90° rated cable.

13.2 ENGINEERING EVALUATION AND CORRECTIVE ACTION

It was determined by Nuclear Plant Engineering that this potential deficiency could create a substantial safety hazard. A Design Change Package (DCP-82/3196) was implemented for Unit 1 in which the installed commercial grade shielded cable on the Division I and II D/Gs was replaced with Class 1E IEEE 383-1974 qualified cable.

Further investigations into the problem subsequently revealed that Bechtel Design Specification M-018.0, Section 6.8.2.6, calls for compliance with Design Specification Appendix N which requires compliance with IPCEA Publication No. S-19-81, Section 6.

In responding to API-83/0974 it was determined that the affected cable had previously been replaced on the Unit 1, Division I and II D/Gs. Therefore, no further action was initiated for Unit 1. Bechtel has issued NCR 6762 to track this concern for the Unit 2 D/Gs.

13.3 CONCLUSIONS

This issue is considered closed for the Unit 1 D/Gs. The replacement electrical cable meets the appropriate requirements of IEEE 383-1974 and TDI's recommended temperature rating.

14.0 FUEL INJECTION LINES

A summary of the concern and its resolution on Grand Gulf is provided in Section 8.0 of the Final Report.

15.0 TURBOCHARGER

15.1 DESCRIPTION

Original problems with supports and components adjacent to the Division I D/G left bank turbocharger have been attributed to a turbocharger that had exhibited signs of unusual vibration. This turbocharger was replaced with a spare turbocharger in August of 1983. Both Division I D/G turbochargers were replaced following the D/G fire in September of 1983 because they were located in the fire area and their ability to carry out their design function was in question. Recurring problems with alignment on the left bank turbocharger were experienced following the replacement in August of 1983. This was corrected in February of 1984 when an extensive maintenance effort was undertaken to correct the alignment problem. Following this effort the Division I D/G completed a 100 hour run (32 hours @ 100%, 68 hours @ 75%) with no further problems.

Turbocharger abnormalities resulted in broken mounting bolts. When the turbocharger is not anchored correctly, the intercooler and the jacket water piping are forced to partially support the turbocharger and thereby absorb a larger amount of fatigue stress. This stress would normally be absorbed by the turbocharger mount. Since neither of these two auxiliaries was designed to support the turbocharger, they both developed cracks and broken welds.

Table 15-1 presents a summary of past problems, causes, and corrective action taken. Further details are provided below:

15.1.1 CRACKED WELDS AND BASE METAL ON INTERCOOLERS

Cracks developed in the base metal on the top of the intercooler along an extruded seam. This seam has since been redesigned by Delaval and a piece of flat bar stock welded over the top of the extruded vee shape to stiffen it. The stay rods extend from one side of the intercooler to the other through a heavier block of steel on the outside. The rod is then welded to this heavier block, this is the weld which broke on the right bank intercooler. Several other stay rods were observed to have deficient welds and were also cut out and rewelded.

15.1.2 CRACKED WELDS ON JACKET WATER PIPING

There were several cracked welds which developed on flanges and fittings where the jacket water system ties into the turbochargers. Since more than one repair was necessary the header was refabricated using standard pipe, fittings, and ASME Section III Welding & NDE Criteria in order to work with codes with which MP&L maintenance and engineering personnel were acquainted.

15.1.3 LOW PRESSURE FUEL OIL HEADER FAILURE

On September 4, 1983, the main fuel oil line feeding the Division I engine headers failed due to fatigue. The oil sprayed onto the turbocharger exhaust gas header transformation piece and ignited. All affected components were repaired or replaced. The failed tube and Swagelok fitting were subjected to a metallurgical evaluation, and the cause of the failure was identified as high cycle fatigue compounded by the absence of tubing supports. Further discussion is provided in Section 7.0 of this final report.

15.1.4 TURBOCHARGER MOUNTING BOLT FAILURES

There have been several instances of turbocharger mounting bolt failures on the GGNS Division I D/G left bank turbocharger.

15.1.5 INDUSTRY EXPERIENCE

Turbocharger problems at other nuclear plants have also been experienced. Recently, Shoreham has experienced a failure of turbocharger thrust bearing in two of their engines.

15.2 ENGINEERING EVALUATION AND CORRECTIVE ACTION

The turbochargers on the Division I D/G left bank have exhibited signs of unusual vibration and misalignment in the past. Improper turbocharger alignment and running of the engine with broken/missing turbocharger mounting bolts, has produced conditions conducive to fatigue crack initiation and propagation in adjacent supports and components.

During the rework of the Division I engine after the fire, the replacement turbochargers were removed and re-seated twice before proper fitup was considered attained. The result was an engine that had no noticeable areas of high vibrations, as attested to by Technology for Energy Corporation when they instrumented the Division I and Division II engines after the fire rework was completed.

Thrust bearing failures similar to those at Shoreham have not been identified at GGNS and are not expected because of the differences in design of the lubrication systems. The failures of the turbocharger thrust bearings at Shoreham have been attributed to probable lack of lubrication during manual engine starting.

Shoreham's TDI D/Gs lube oil systems utilize two pumps, one an engine driven pump and the other an electric driven heater pump (See Figure 15-1). The GGNS D/Gs lube oil system utilizes three pumps, one an engine driven pump, one an electric driven heater pump and the other an electric driven auxiliary pump (See Figure 15-2).

15.2 (Continued)

Presently, on a manual start Shoreham does not have the means of supplying oil to the turbocharger thrust bearing other than through a turbocharger lube oil drip system. The turbocharger lube oil drip system is also used at GGNS and is essentially the same as Shorehams. However, prior to a manual start of the D/Gs at GGNS the engine is prelubed for two minutes or less with the auxiliary lube oil pump which pressurizes the turbocharger thrust bearing with lube oil. This precludes the type of failures reported at Shoreham.

Recently, there have been several additional occasions when the Division I D/G left bank turbocharger mounting bolts failed. The main reason for these failures was again considered misalignment of the turbocharger with its associated piping and components. An engineering evaluation of the turbocharger mounting arrangement is being performed and procedures designed to preclude misalignment have been implemented.

The alignment problems with the turbochargers are partially a function of the custom fit arrangement of the turbocharger with the flanged connections on the mating piping and apparatus. Each engine supplied by TDI has slightly different piping due to the fact that it is hand built. The specific problems encountered with the left bank turbocharger on Div. I was the misalignment of the turbocharger and intercooler mating flanges. These flanges were misaligned such that if you tightened down the flanged connection, the turbocharger was cocked on the mounting pedestal approximately 30 mils. (Figure 15-3.) This problem was attacked several times before a satisfactory resolution was achieved. TDI recommended the following:

1) Intercooler flange be cut

2) The turbocharger bolted down

3) The intercooler flange be bolted to the turbocharger

4) The intercooler flange tack welded to the intercooler

5) The turbocharger removed

6) The intercooler flange rewelded

7) The turbocharger remounted

8) The turbocharger bolted down

9) The intercooler flange bolted up

10) The turbocharger mounting bolts removed

11) The mounting plate checked for clearances

12) The turbocharger remounted

15.2 (Continued)

13) Perform a maintenance run

14) The turbocharger mounting bolts removed

15) The mounting plate rechecked for clearances

16) The mounting bolts installed

At the end of this process the engine was tested for 100 hours and the mounting clearances rechecked.

The final conclusion was that the problem had effectively been solved. The final clearances were 0, 0, 0, and 3 mils. on each of the four (4) corners of the mounting plate. These values are acceptable and the successful completion of the 100 hour run demonstrated that the problem has been solved.

15.3 CONCLUSIONS

The susceptible areas in the piping and components around the turbochargers have been identified by past failures. The integrity of these areas has been enhanced by the use of approved ASME code welding, procedures, and materials during rework. Since these enhancements, the weld and component failures have not reoccurred.

The original problems with the left bank Division I turbocharger were due to a turbocharger that exhibited signs of abnormal vibration. Since this turbocharger was replaced there have been alignment problems with the left bank turbocharger resulting in broken mounting bolts. In February of 1984, the turbocharger was removed and carefully refitted with all attached piping and mating surfaces and correctly installed. A 100 hour performance run was performed after which the turbocharger was inspected and no problems were discovered. There have been no further problems experienced since this was done. MP&L feels confident that this problem has been successfully corrected. As a part of the effort by the TDI Diesel Generator Owner's Group, MP&L will fund a thorough study of turbocharger mounting arrangement and will take any corrective actions deemed necessary.

TABLE 15-1

ENGINE MOUNTED COMPONENTS PROBLEMS

ATTRIBUTED TO TURBOCHARGER VIBRATION

ITEM	DESCRIPTION OF PROBLEM	CAUSE	CORRECTIVE ACTION
1	Cracked welds and base metal cracks on intercoolers.	Fatigue compounded by high vibration from turbocharger.	Repaired welds and base metal cracks. Reseated turbocharger to eliminate undue stresses caused by misalignment.
2	Cracked welds on jacket water flanges and piping headers.	Fatigue compounded by high vibration from turbocharger.	Repaired welds. Refabricated header to ASME III Class 3 reseated turbocharger.
3	Low pressure fuel oil header failure resulting in Div I fire.	Fatigue compounded by high vibration from turbocharger.	Replaced fuel line and fittings reseated turbocharger.

FIGURE 15-1: Illustration of Shoreham TDI Diesel Generator Lube Oil System
(Reference Telecon Shoreham)

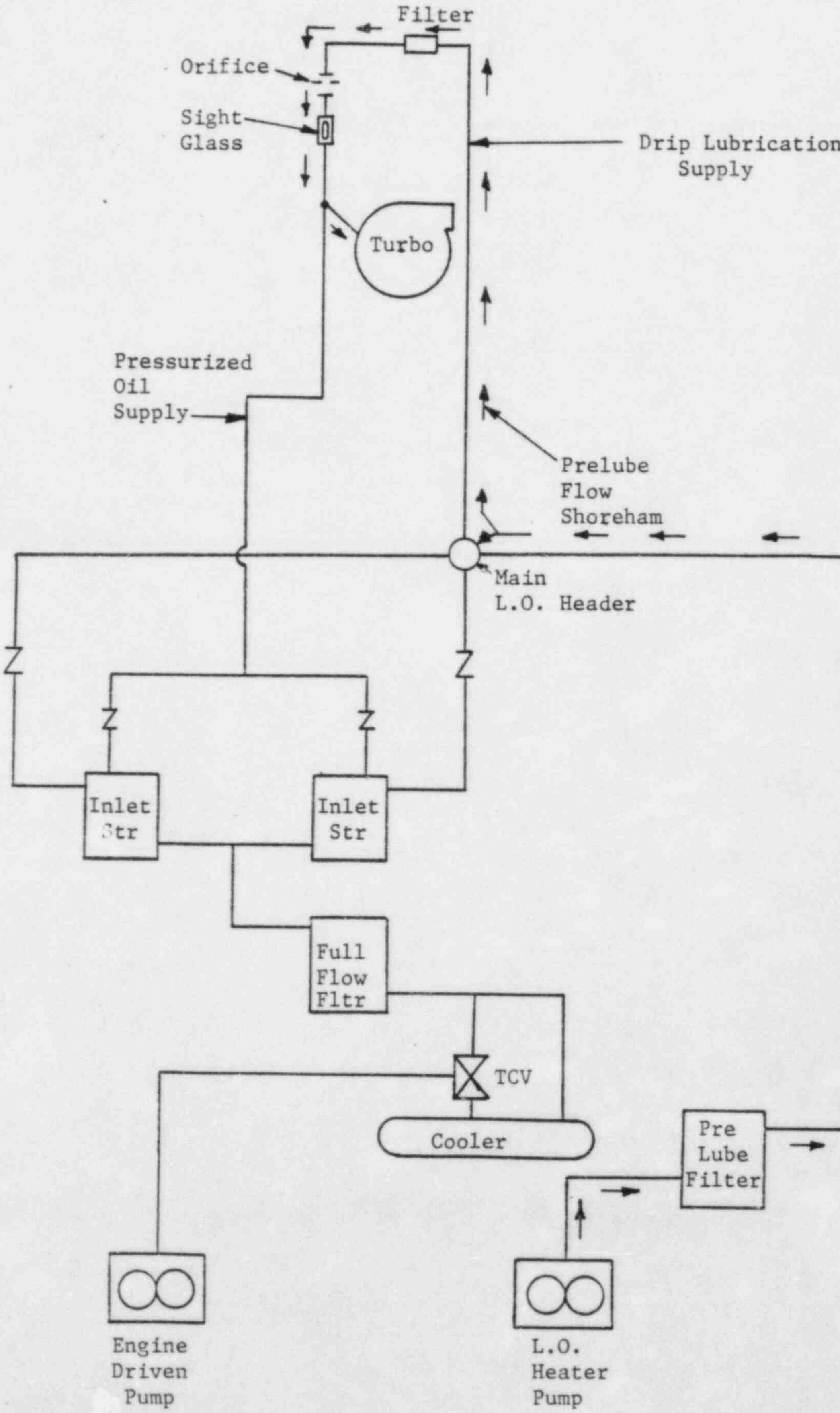


Fig. 15-1

FIGURE 15-2: Illustration of GGNS TDI Diesel Generator Lube Oil System

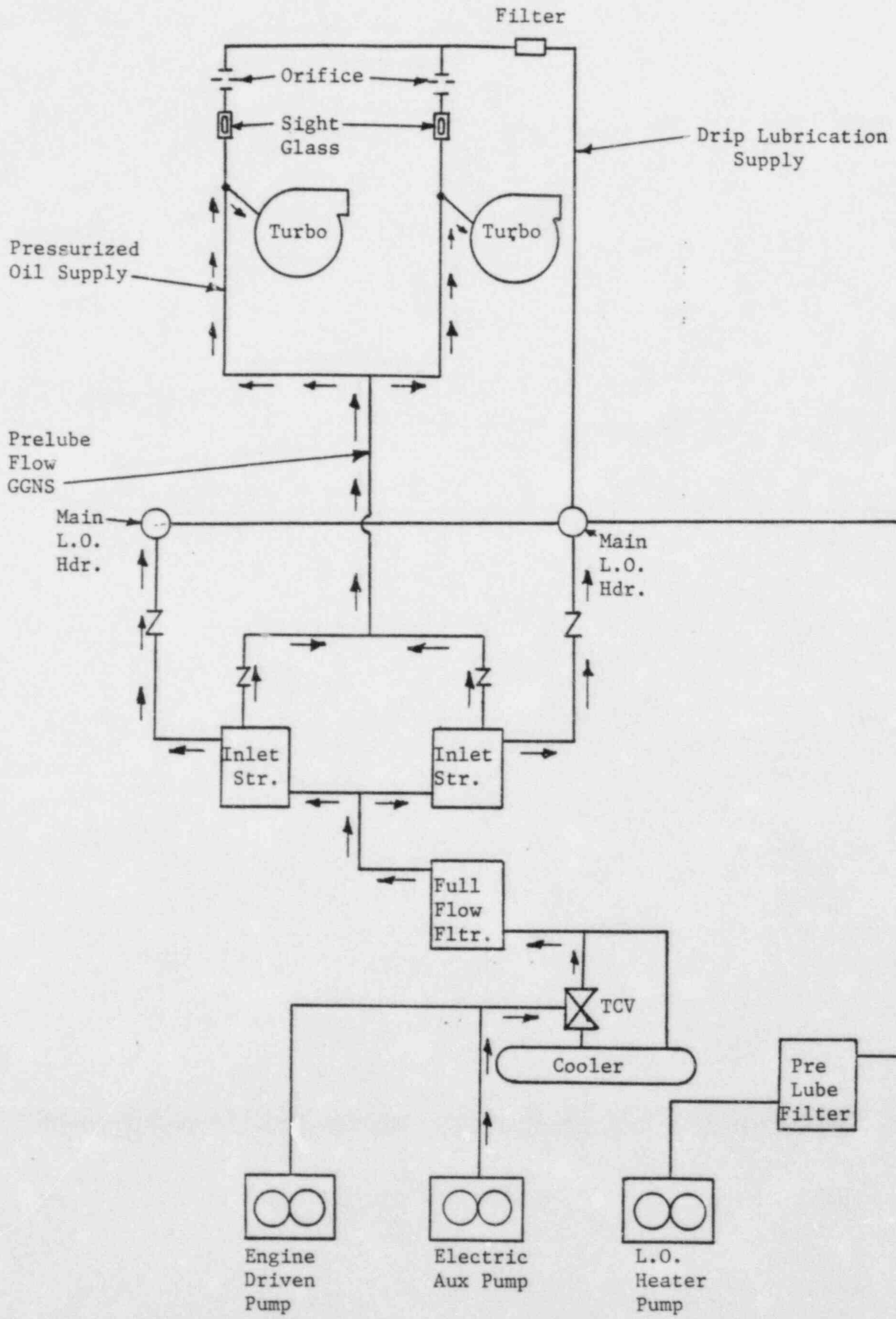
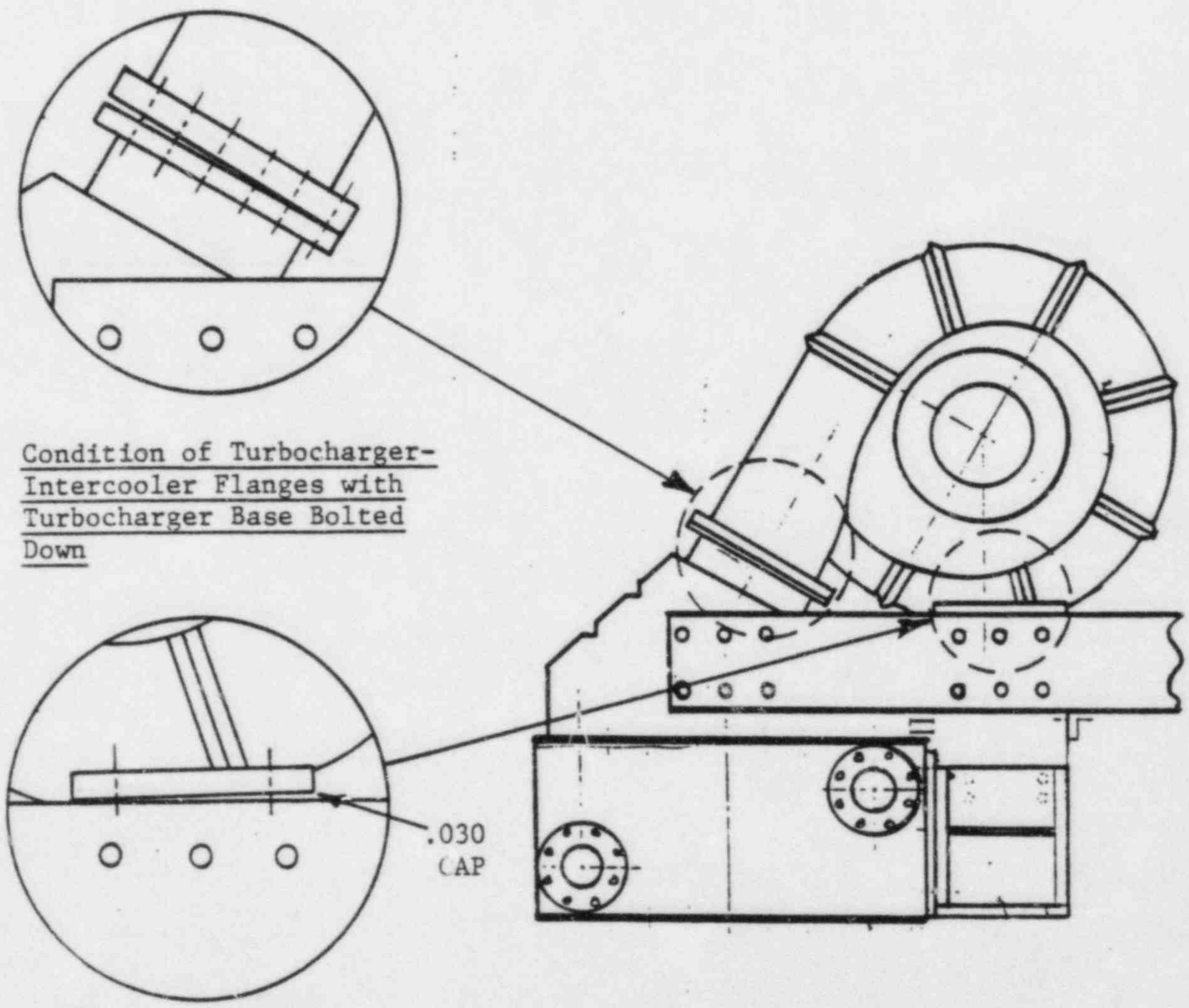


Fig. 15-2



LEFT BANK TURBOCHARGER
MOUNTING ARRANGEMENT

FIGURE 15-3

16.0 JACKET WATER PUMPS

16.1 DESCRIPTION

Shoreham has experienced a jacket water pump shaft failure.

16.2 ENGINEERING EVALUATION

This problem apparently affects only the in-line engines. No jacket water pump shaft failures have been reported on Vee-type engines to date.

16.3 CONCLUSIONS

As of this time, MP&L has been unable to obtain evidence of generic jacket water pump shaft failures on DSRV-16-4 engines.

Since this problem appears to be unique to in-line engines, it has not and is not expected to occur on the GGNS diesel engines.

17.0 AIR START VALVE CAPSCREWS

17.1 DESCRIPTION

On May 13, 1982, TDI reported a potential defect concerning the capscrews that are used to retain the air start valves in the cylinders heads to the NRC under the provisions of 10CFR21. The 3/4-10x3 inch long capscrews were suspected of bottoming out in the tapped holes in the cylinder heads. This could result in insufficient or unequal clamping forces being applied to the air start valve.

17.2 ENGINEERING EVALUATION AND CORRECTIVE ACTIONS

TDI recommends replacement with 2 3/4 inch long capscrews or machining 1/4 inch off the existing 3 inch capscrews. A design change was issued by MP&L to implement corrective actions. The air start valve capscrews on the Unit 1, Division I and II D/Gs were modified by machining 1/4 inch off the length.

17.3 CONCLUSIONS

Corrective action is considered complete in regards to the air start valve capscrew problem on the Unit 1 D/Gs.

ATTACHMENT 2 TO THE
UPDATED REPORT ON GGNS
DIVISION I AND II TDI
DIESEL GENERATORS

PISTON MANUFACTURING DETAILS

APRIL, 1984

Z3rg68

ATTACHMENT 2

PISTON MANUFACTURING DETAILS

1.0 GENERAL

As reported by TDI (Reference 2), all 450 RPM rated "Enterprise" R-4 series engines have been furnished with two-piece pistons which incorporate a cast steel piston crown attached to a cast modular iron piston skirt by means of four studs. This piston design has evolved since its inception in 1969 to incorporate design improvements for high reliability and less costly manufacture. As horsepower ratings of engines increased in the mid-1960's, Transamerica Delaval and other medium speed diesel engine manufacturers abandoned the older style single piece piston design. The two piece piston is inherently better equipped to deal with the higher thermal inputs of high Brake Mean Effective Pressure (BMEP) engines, because it allows thermal growth of the crown without causing excessive bending stresses in the skirt. The two piece piston design is also better equipped to handle the higher pressure and inertia loadings of the increased horsepower engines. The modular iron skirt has passed through several design changes. Five different designs have been used and are identified by TDI terminology as "AF", modified "AF", "AN Old Style", "AN New Style", and "AE".

2.0 GGNS BACKGROUND

Only the "AF", modified "AF" and "AE" piston skirts have been used at GGNS. The "AF" piston skirts were originally supplied by TDI on the Division I and II GGNS engines. When problems were encountered with material quality of the washers (piston crown/skirt bolt) and GGNS experienced a piston crown/skirt separation, MP&L responded by upgrading hardware in accordance with SIM 324 to the modified "AF" piston skirt design. The cracking discovered later on Shoreham modified "AF" piston skirts and rejectable indications found at inspection prompted MP&L to change out all piston skirts at GGNS to the latest "AE" design piston skirts.

3.0 PISTON TYPES

3.1 "AF" AND MODIFIED "AF" PISTON SKIRTS

"AF" piston skirts use spherical washers on the four studs which attach the crown to the skirt. These spherical washers provide fastener flexibility. These commercially supplied washers proved to have inconsistent quality and large variations in heat treatment and manufacturing tolerances. As a result, a small number of the washers failed in service, resulting in piston, skirt/crown separation. One such separation occurred on the Division II D/G during field testing. To solve the spherical washer problem, the design was modified to incorporate a "full stack" Belleville washer arrangement resulting in a modified "AF" piston skirt.

3.1 (Continued)

The "AF" style piston skirt casting received the following heat treatment:

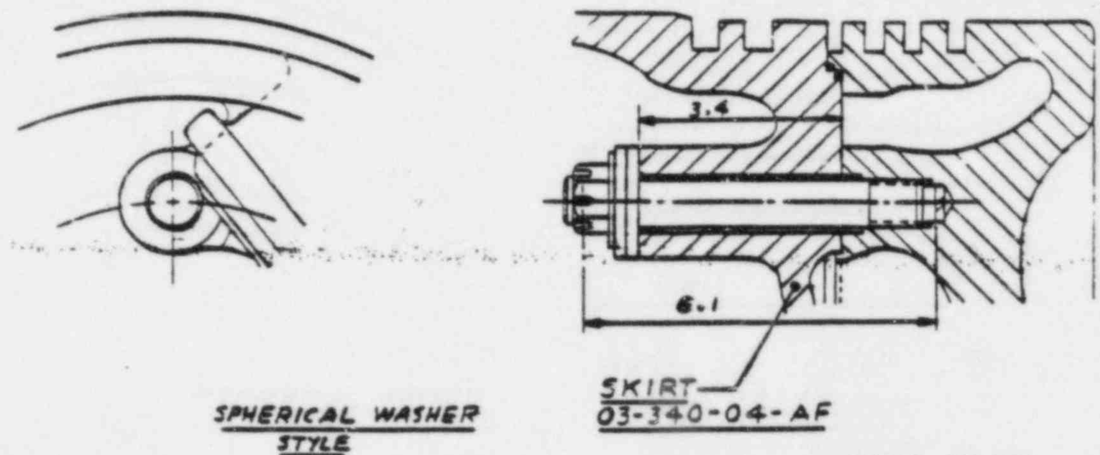
- o Heat to 1750 degrees F. (near the upper critical temperature) for 3 hours. Normalize (air cooled) in still air. This results in a pearlitic structure with 100,000 psi tensile strength.
- o Re-heat to 1050 degrees (slightly below the lower critical temperature) for 3 hours and cool in still air. This tempering process produces the desired ductility in the nodular iron.

3.2 "AE" Pistons

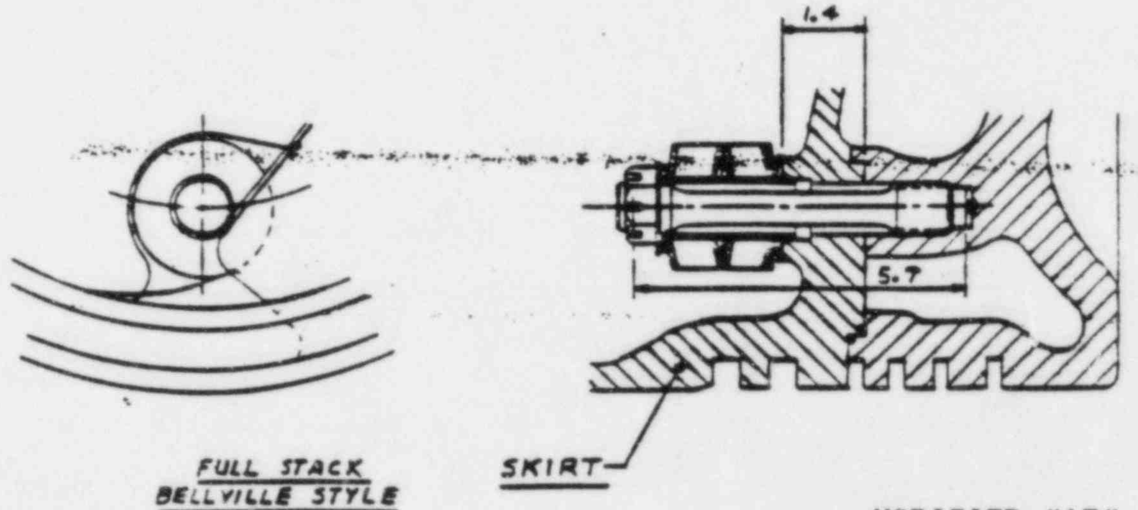
The "AE" piston, the latest R-4 piston skirt design, just concluding research development testing, incorporates the field experience on the R-4 series engine and the R-5 series engine.

The "AE" design utilizes a "half stack" Belleville washer arrangement. All "AE" skirts are heat treated to produce stress relieved 100,000 psi tensile strength nodular iron. All piston skirts in the TDI units at GGNS Unit 1 have been replaced with this design. The "AE" style skirt is interchangeable with existing R-4 piston crowns and requires only minor hardware changes.

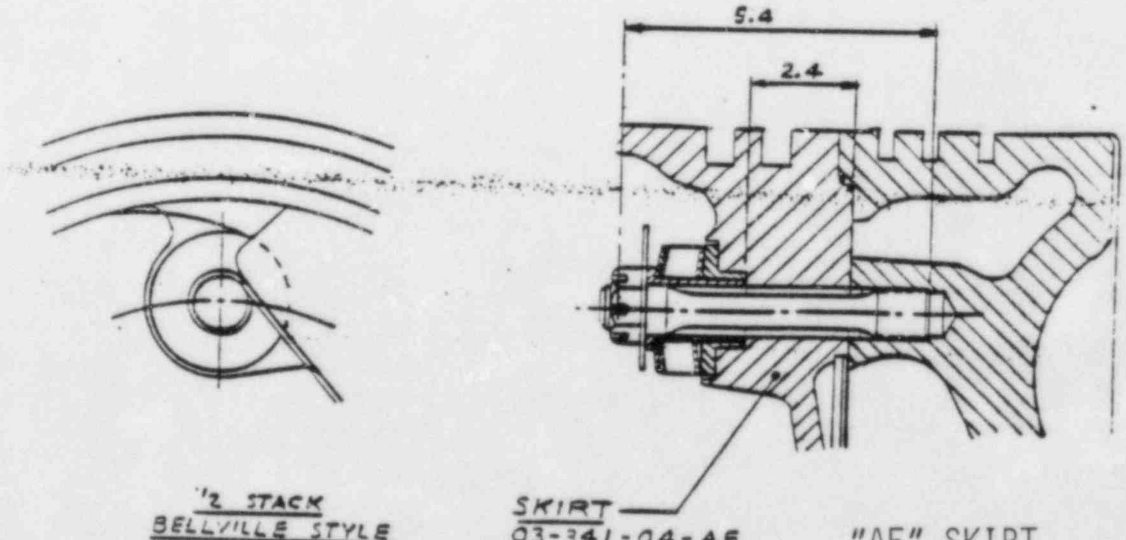
FIGURE A2-1: PISTON COMPARISONS



"AF" SKIRT



MODIFIED "AF" SKIRT



"AE" SKIRT

Fig. A2-1