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September 22, 1983

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MHC  
Book

Alan R. Dynner, Esq.  
Kirkpatrick, Lockhart, Hill,  
Christopher & Phillips  
1900 M Street, N.W.  
Washington, D.C. 20036

Richard Goddard, Esq.  
U.S. Nuclear Regulatory  
Commission  
Washington, D.C. 20555

Dear Messrs. Dynner and Goddard:

In its Order dated June 22, 1983, the Board directed LILCO to discuss with the Staff and County the testing or inspection of the new rocker arm shaft bolts installed on the Shoreham diesel generators. As you will recall, the rocker arm shaft bolts currently installed on the diesel generators are of an improved design installed following the failure of one of Shoreham's original rocker arm shaft bolts. Although there was only one failure, all of the 96 rocker arm shaft bolts on the Shoreham diesel generators were replaced. As the Board noted, the new bolts, unlike the old ones, were subjected to magnetic particle testing and, in addition, substantial testing hours were accrued on the Shoreham diesels since the installation of the new bolts with no problems. In its consideration of the rocker arm shaft bolt problem, the Board concluded that the only remaining question related to LILCO's plans to inspect a sample of the new bolts. Specifically, the Board stated

The one remaining long term question regarding the bolts directly should be easily resolved without resort to litigation. That question is the scope of LILCO's plans (undecided at the time of the conference of parties. Tr. 21,385-89

HUNTON & WILLIAMS

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Page Two

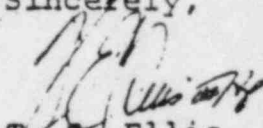
(Youngling)), if any, to prudently inspect a sample of the new bolts at reasonable intervals of operation of the diesels (not too soon or too long), perhaps including before any fuel-loading given the substantial testing hours accrued with the new bolts. Tr. 21,368-69 (Goldsmith). We direct LILCO to discuss such a testing proposal with the Staff and the County, reporting any agreement or disagreements. Subject to our approval of such surveillance, or approval of any position by LILCO as to why such sampling should not be done, we find there is nothing left to litigate regarding the reliability of the rocker arm bolts.

In connection with the replacement of Shoreham's original cylinder heads with new cylinder heads, LILCO took the opportunity to conduct an inspection of a random representative sample of the bolts. More specifically, seven bolts from each engine, 21 bolts total, were randomly selected for inspection. Each bolt was cleaned with solvent to remove any lubrication and facilitate the inspection. The inspection consisted of visual observation for cracks, laps or seams on the threads, shanks or heads of the bolts. Attention was also given to the internal threads and to the junction of the head and shank portion. The visual inspection performed on the bolts is the same as that described in ASTM A614. No indications or problems were observed.

Given the results of this inspection and the fact that all of the new bolts that have been installed were subjected to magnetic particle inspection before installation, LILCO does not consider that any further inspection or examination of the bolts is necessary.

Pursuant to the Board's Order of August 29, 1983, please let us know whether you agree that further inspection is unnecessary so that we can report to the Board as required.

Sincerely,

  
T. S. Ellis, III

75/403

bc: Edward J. Youngling

**NEI****Parsons Peebles - Electric Products, Inc.**

1725 Clarkstone Road  
Cleveland, Ohio 44112  
Telephone: (216) 481-1500  
Telex: 241664

November 7, 1983

Transamerica Delaval, Inc.  
Engine & Compressor Division  
850 85th Avenue  
Oakland, California 94621

Attention: Mr. John Witt

Subject: Long Island Lighting Co.  
Repair on Rotor

Gentlemen:

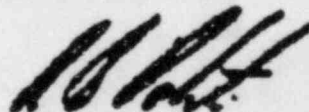
We refer to our report on the failure analysis of the rotor pole on a Long Island Lighting Co. generator and would like to amplify on their statement "typical example of marginal workmanship".

What is meant by the statement is that we take every precaution to insure good workmanship and good quality control. However, even with all the inspection and testing performed, in some isolated instances faults do occur in machines. These faults are random and difficult to find even after failure. In this particular case we applied a higher voltage to the pole to cause the fault to increase so that we could easily detect the exact location of the fault. When such a random fault is present, in the position of this particular fault, due to the continual heating and cooling of the coil together with the normal vibration present in any rotating body, a failure is likely to occur over a period of use. This is not a generic fault and in order to put this fault in perspective, we would state we have built approximately 170 Class 1E generators in the past 18 years, and this failure is the first such pole failure we have experienced.

This fault relates to one pole only, and we have no reason to expect any further failures on this or any of the other machines based on our past experience.

We trust this will explain the situation fully.

Yours very truly,



Ron B. Politi  
Manager of Marketing

December 7, 1983

M. H. Milligan/W. M. Judge

Integrity of EDG Engine Bases  
Shoreham Nuclear Power Station - Unit 1  
W.O. 44430/48923

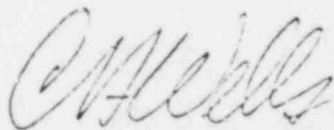
Cracks have been reported in the engines of DG 102 and DG 103 following disassembly. Cracking has been primarily confined to the upper surface of the base between the main bearing saddles and the bearing cap stud holes and to the bore of the stud holes adjacent to the bearing saddles. The separation between the saddles and bolt holes at the top of the bases of DG 102 and DG 103 is approximately 0.125 inches, with some as small as 0.100 inches, and the edges of the holes are not chamfered. The separation is approximately 0.250" in the case of the DG 101 base, and the holes are chamfered approximately 0.125". Thus the DG 101 base, which was found not to contain cracks, is inherently stronger than the other two.

Concern was initially focused upon the possibility of growth of these cracks under operating loads. Apparently other non-counterweighted DSR-48 engines in U.S. Coast Guard service have cracked severely. Dynamic loading is especially severe at the main journals of cylinders No. 4 and No. 5, at which the connecting rod pins are simultaneously at top dead center, when the throws are not fitted with counterweights. The cracking problem in the Coast Guard engines has been attributed by TDI to insufficient main bearing cap stud preload. Torque requirements for the bearing cap nuts were subsequently increased. The SNPS diesel engines have reportedly always been operated with this increased torque. We analyzed the forces on the main bearing caps according to the results of a journal orbit analysis conducted by TDI. With the specified torque, the stud preload is sufficient to prevent motion of the bearing caps, i.e., the side load is too small to overcome static friction. Therefore, the studs cannot hammer the bore of the stud holes. Moreover, our calculations show that the frictional shear stress and normal stress in the vicinity of the stud holes are too low to cause growth of the cracks, and we have not been able to predict any other operating loads that would result in crack growth in this area.



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Subsequently we attempted to determine the cause of the observed cracks. In the case of DG 102 it was logical to associate the cracks with high impact loads on the bearing caps resulting from fracturing of the crankshaft and destruction of the No. 7 connecting rod bearing shells. Explanation of the numerous crack indications in the DG 103 base is still uncertain, and TDI has not yet provided their conclusions. However, we have observed use of a torque multiplier to aid in removal of locking pins in the lower nut pockets of certain main bearing stud holes. Calculations show that the side loads developed between the studs and the 1/8" wall adjacent to the bearing saddle are more than sufficient to fracture this wall. Furthermore, visual examination of the studs showed several to exhibit scratches and dents coinciding approximately with the top of the base, and in at least one instance a stud hole was found to be deformed by radial force directed towards the bearing saddle. We concluded that although other sources could have contributed to cracking, the cracks had probably been initiated in the course of disassembly and, in any event, could not be attributed to engine operation.



C. H. Wells

CHW:mp

cc: A. Earley  
C. K. Seaman  
SR2

December 15, 1983

M. H. Milligan

Inspection of Jacket Water Pumps  
from Transamerica Delaval Diesel  
Engines, SNPS, FaAA No. PA07396  
Shoreham Nuclear Power Station - Unit 1  
W.O. 44430/48923

On October 12-14, 1983, Dr. Donald O. Cox inspected jacket water pumps from the Transamerica Delaval Incorporated "Enterprise" diesel engines used to drive standby generators at the Shoreham Nuclear Power Station.

A typical water jacket pump is shown in Photographs 1 and 2. The pump is driven through a gear at one end of the shaft which engages a gear on the crankshaft. The impeller is mounted at the opposite end of the shaft and held in position by an interference fit, retaining washer and nut as shown in Photograph 2. The jacket water pump has apparently undergone two major redesigns due to pump shaft fatigue failures in the region where the impeller is attached.

The "original" design incorporated a straight shaft and key to prevent relative rotation of the impeller. However, several units with this design failed during operation in Saudi Arabia, allegedly due to impeller looseness on the shaft.

A redesign was incorporated which used a tapered shaft to attach the impeller to shaft; use of a key was retained. However, LILCO experienced three pump failures in pumps with the tapered shaft/key attachment design.

TDI again redesigned the attachment, incorporating three major changes: 1) the impeller material was changed from red brass to ductile iron, 2) the diameter of the tapered portion of the shaft in the area under the impeller was increased, and 3) the key method of antirotation was eliminated. Thus, the only force attaching the impeller to the shaft in the current design is the friction force due to the interference fit used during assembly.

During tear down of diesel engine No. 102 after a crankshaft failure, the impeller of the water jacket pump was found to have spun on the shaft. It was postulated that this pump could have been exposed to unexpected impulse loads during failure of the crankshaft. Therefore, an investigation was initiated to establish that the current impeller/shaft attachment insures the impeller will remain firmly attached to the shaft. This effort involved

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disassembly of the pumps from DG 101 and 103, which had provided approximately 200 hours of service.

The significant components of the DG 102 pump include the impeller, shaft, retaining washer and nut which are shown in Photographs 3 through 5. The impeller is fabricated from a ferrous material, specified as ductile iron. There was no evidence of cracking of the impeller, although some casting porosity was observed and there was some corrosion debris at internal portions of the impeller vanes.

Impeller damage was confined to the bore of the hole where it is press fit to the shaft and the bearing surface at the periphery of the hole where the retaining washer contacts. These areas are shown in Photographs 6 and 7. The bore of the impeller hole was severely scored at two major locations consistent with major scoring of the shaft. There was minor damage to the remainder of the impeller bore. The surface around the hole, on the suction side of the impeller, has been severely scored as a result of relative rotation between impeller and retaining washer (Photograph 7).

There was a circumferential lip approximately 1/32 inch in height around the hole, resulting from metal distortion as the impeller rotated relative to the washer. The lip was relatively uniform in height around the hole periphery. There was, however, no visual evidence of heat damage to the bore of the impeller or the hole periphery, suggesting the relative rotation between shaft and impeller occurred for only a short period of time.

The threads at the end of the pump shaft appeared to be in excellent condition, with some evidence of debris in the thread roots (most probably loctite or die penetrant). The tapered surface of the shaft where the impeller seats has been severely scored in two major locations near the center of the contact area. This region is shown in Photographs 8 and 9. The two major areas of scoring on the shaft appear to match with the scores in the bore of the impeller. There was evidence of minor scoring of the taper region over the entire area of impeller/shaft contact. However, there was no damage where the retaining washer was located. As with the impeller, there was no evidence of heat damage to the shaft taper, again suggesting that relative rotation between impeller and shaft did not occur for an extended period of time. Rockwell hardness tests at mid-length of the shaft showed the material hardness to be 84-86 HRB. The material was non-magnetic, indicating an austenitic grade of stainless steel.

The surface of the nut, which was in contact with the retaining washer during operation, has a single circumferential score mark approximately 3/16 to 1/4 inch in length as indicated in

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Photograph 11. The direction in which this mark was produced indicates that the nut/shaft rotated as a unit in their normal driven direction relative to the retaining washer. This suggests the washer rotated with the impeller for at least a portion of one revolution. The washer surface which was in contact with the nut had a circumferential score mark consistent with the damage seen on the nut as shown in Photograph 12.

During a discussion with the mechanic who disassembled the pump, it was indicated that once the torque necessary to overcome the loctite in the threads was applied, the nut turned freely. This indicates that the nut was not holding the impeller in contact with the shaft. Furthermore, when hydraulic equipment was used to push the shaft from the impeller during disassembly, there was no "POP" as would be expected if the interference fit was still present. Thus, it seems quite apparent that at the time of disassembly the impeller was relatively loose on the shaft.

In summary, it is apparent that the impeller from the water jacket pump of the diesel engine from Unit 102 spun on the shaft after the crankshaft failure prior to disassembly. Visual evidence suggests relative rotation between shaft and impeller did not occur for an extended period of time. This opinion is based on two observations: 1) the major damage to mating impeller and shaft surfaces is confined to a small region of the overall contact area, and 2) there is no evidence of damage from frictional heating which would be expected if the impeller had spun on the shaft for a significant period of time. Thus, it is most likely that impulse loads imposed during failure of the crankshaft were great enough to overcome frictional forces in the impeller/shaft joint. The impeller then rotated relative to the shaft for the short period of time between crankshaft failure and engine shutdown.

In an effort to establish conclusively whether the damage to the water jacket pump of Unit 102 was a result of the traumatic events involved in the crankshaft failure, pumps from Units 103 and 101 were also disassembled and inspected. During disassembly of the pump from Unit 103, a torque exceeding 175 foot pounds was required to loosen the nut securing the impeller to the shaft. Furthermore, a significant amount of force was required to separate shaft and impeller and there was a very definite "pop" as the interference fit of this joint was broken. These observations are in direct contrast to the situation in Unit 102 and indicate the impeller was still firmly joined to shaft at the time of disassembly of the Unit 103 pump.

The various components of consequence from the Unit 103 pump are shown in Photographs 13 through 15. There was some corrosion in the vanes of the impeller as well as areas of casting porosity



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similar to those seen in the pump from Unit 102. However, the impeller bore was in excellent condition with no scoring or other evidence that there was any relative rotation with the shaft. The machining marks were still evident and the bore surface was very smooth (Photograph 16). The bearing face on the suction side of the impeller showed no evidence of wear from the retaining washer.

The shaft taper surface was in excellent condition with no evidence of scoring, wear or relative rotation between impeller and shaft (Photograph 17). The only evidence of wear on any component of this pump was at the wear ring of the suction flange. This ring showed wear over approximately 120° of the circumference as shown in Photograph 18. There does not appear to have been any contact over the remaining portion of the wear ring circumference.

Mr. Gary Rogers observed the disassembly of the pump from Unit 101. He likewise did not observe any evidence of relative motion between the impeller and shaft of that pump. Thus, in two of three water jacket pumps from the emergency generator diesel engines there was no evidence that the impeller loosened on the shaft during operation.

Similar negligible amounts of corrosion and porosity were seen on the Unit 102 and 103 jacket water pumps, and were found to be unrelated to the relative motion of the shaft and impeller, and unrelated to the reliable performance of the pumps.

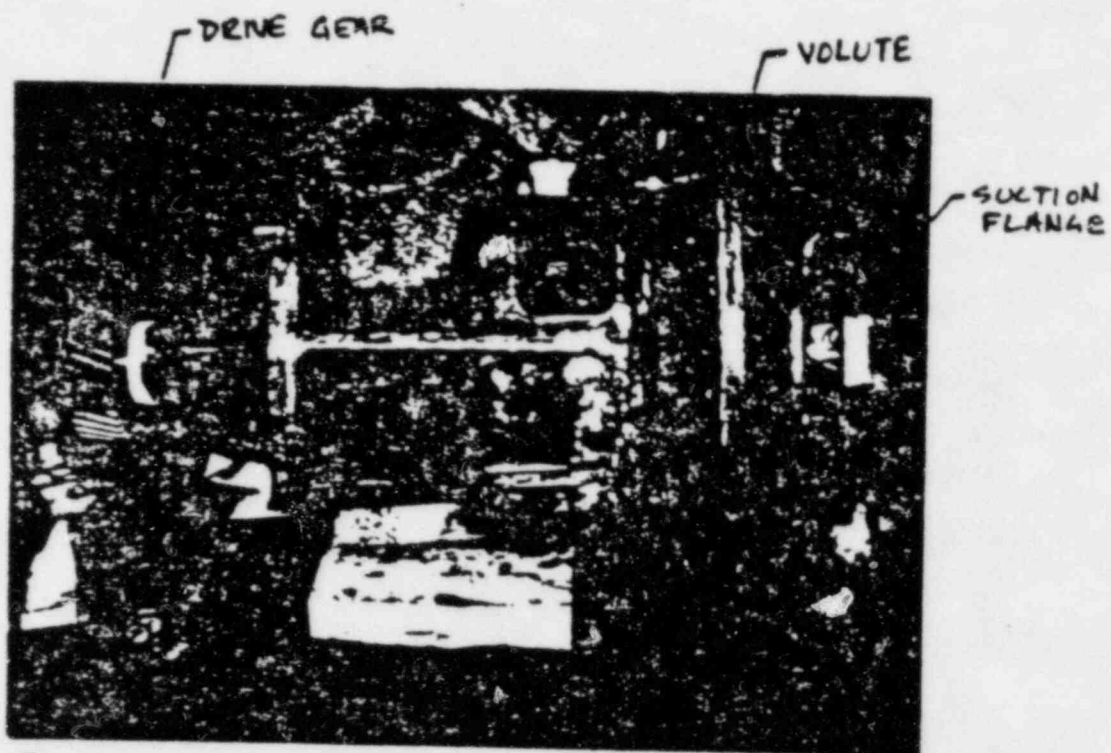
The only instance where definite evidence that the interference fit between impeller and shaft was lost during service is the pump from Unit 102 which sustained the crankshaft failure. On this pump there is no evidence of heat damage, and major scoring is confined to a relatively small area of the shaft/impeller contact region. This indicates that rotation of impeller on shaft did not occur for a long period of time. FaAA therefore, concludes that the impeller did not separate from the shaft until the time of crankshaft failure. When the crankshaft failed, it is most likely that severe impact loads produced forces on the impeller/shaft joint which exceeded the strength of the interference fit.

*Donald O. Cox*

Donald O. Cox  
Failure Analysis Associates

DOC:ss

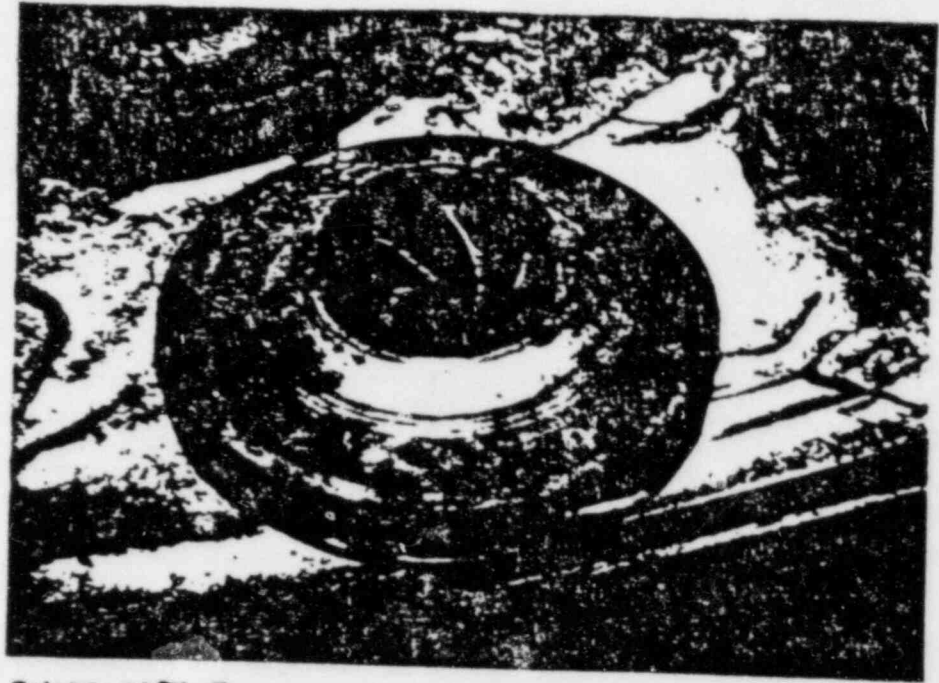
cc: T. Ellis, Hunton & Williams  
G. Rogers, FaAA  
C. Wells, FaAA



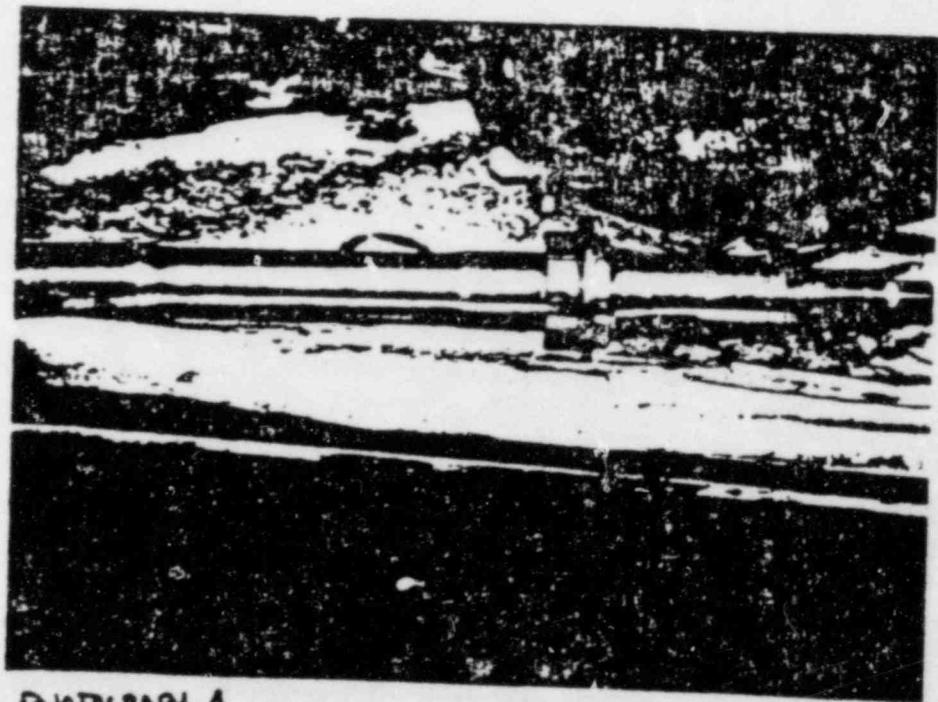
PHOTOGRAPH 1



PHOTOGRAPH 2



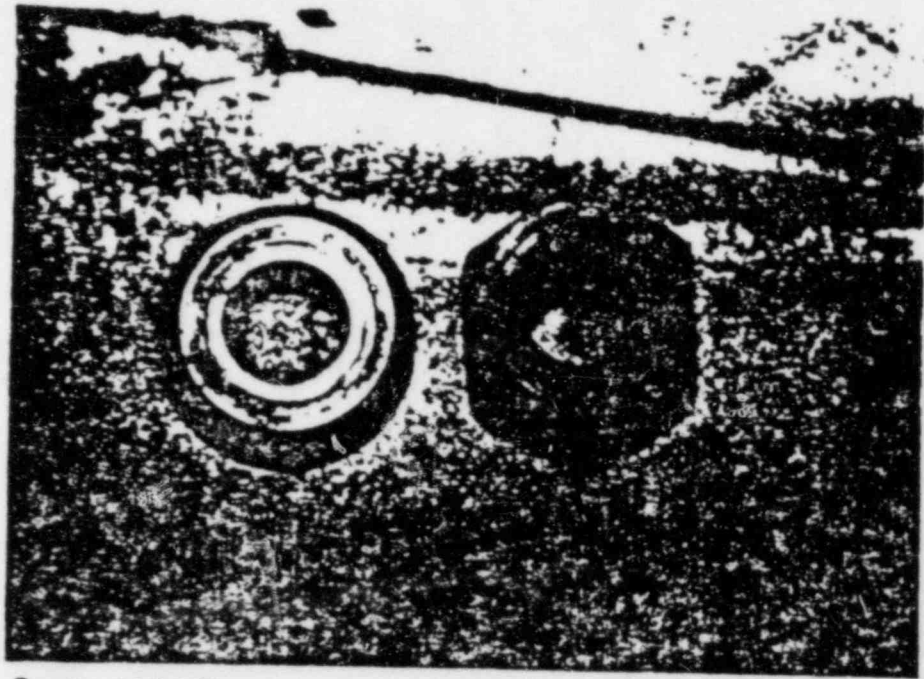
PHOTOGRAPH 3



DRIVEN  
END

IMPELLER  
END

PHOTOGRAPH 4

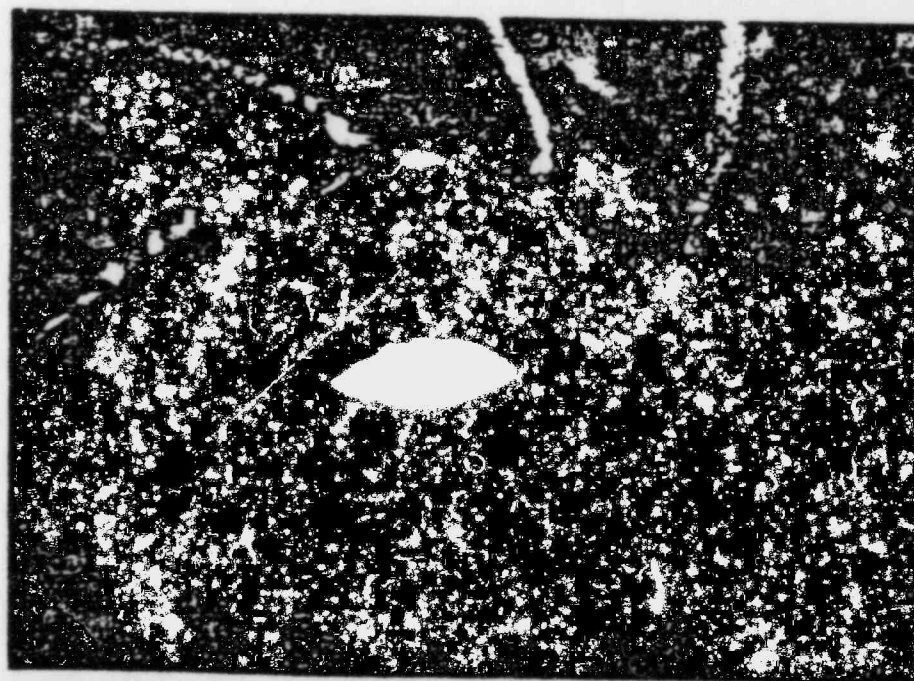


PHOTOGRAPH 5

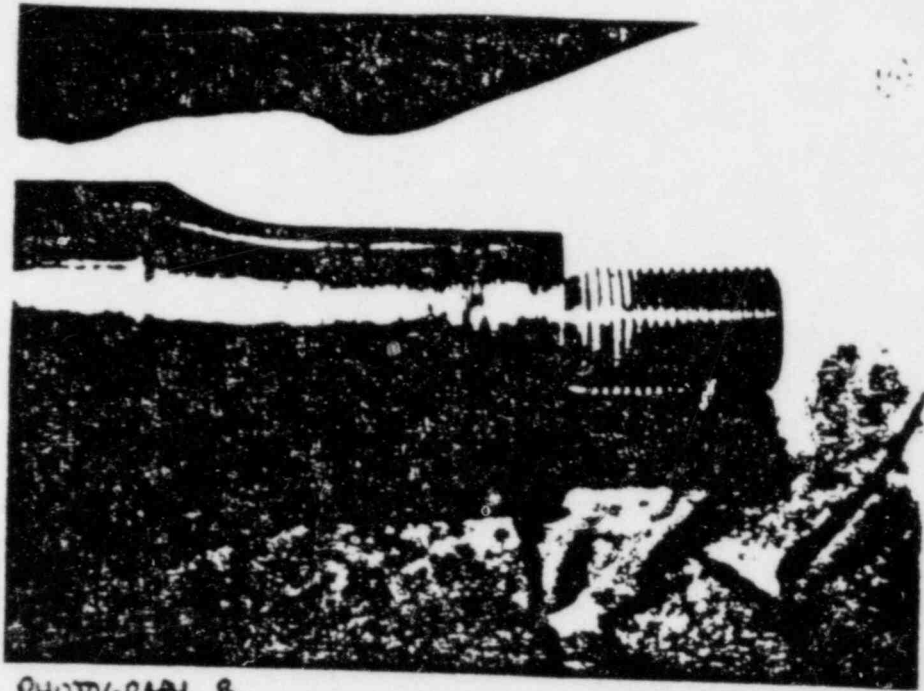




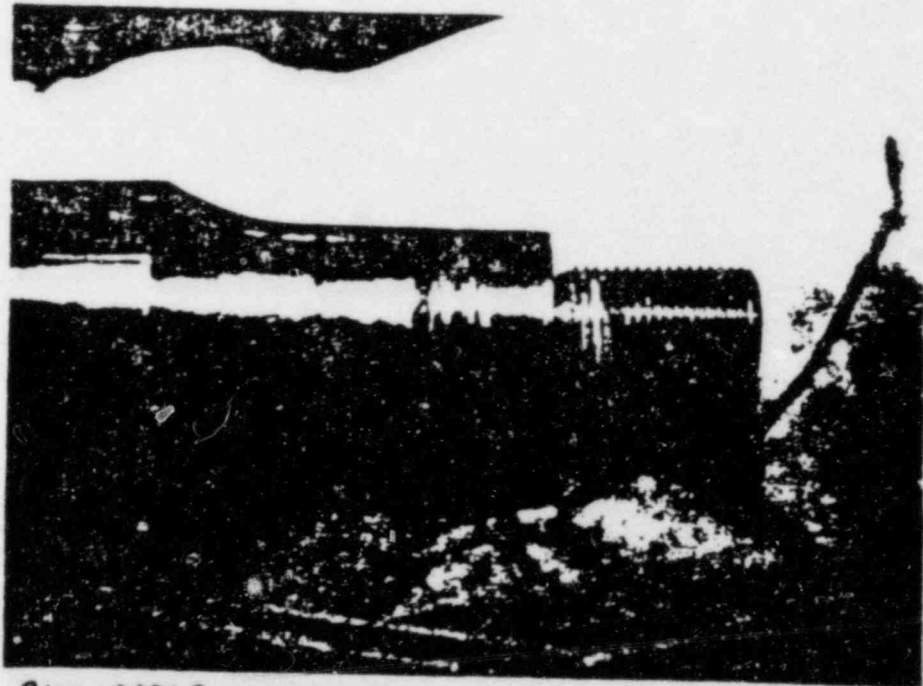
PHOTOGRAPH 6



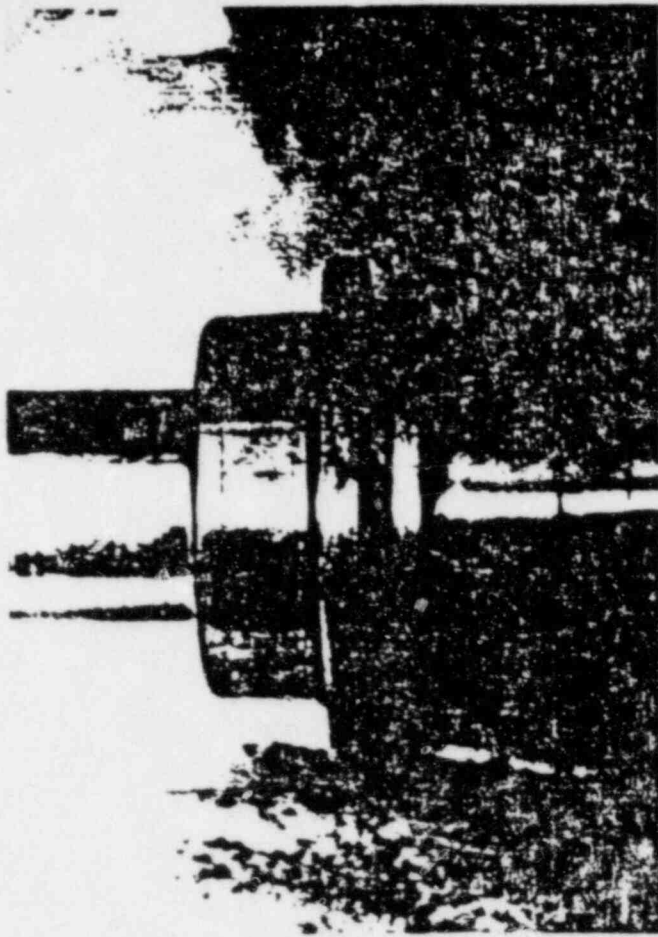
PHOTOGRAPH 7



PHOTOGRAPH 8

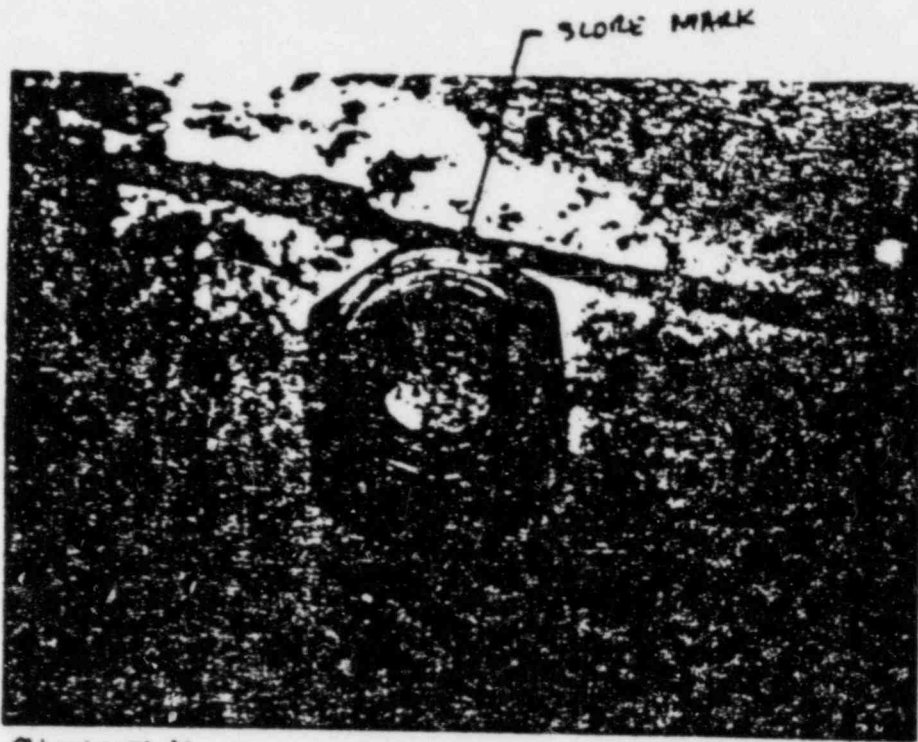


PHOTOGRAPH 9

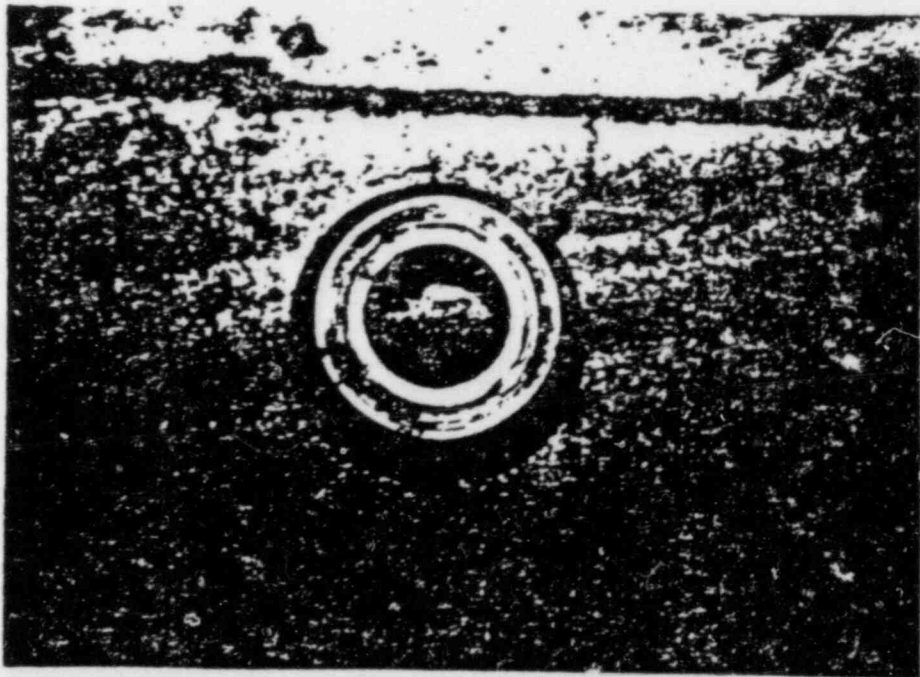


IMPELLER  
END

PHOTOGRAPH 10



PHOTOGRAPH 11

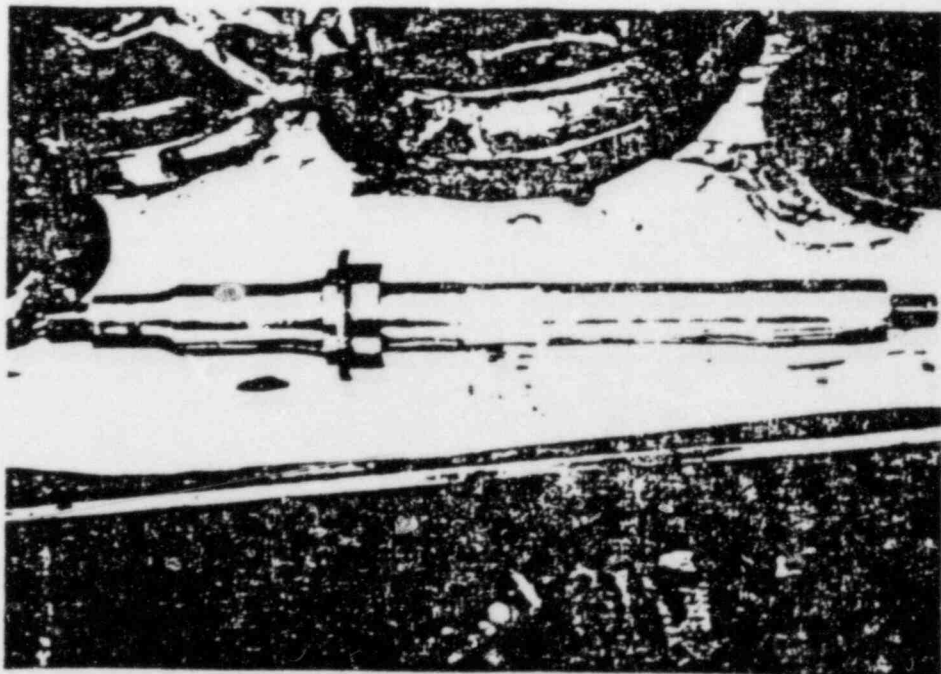


PHOTOGRAPH 12

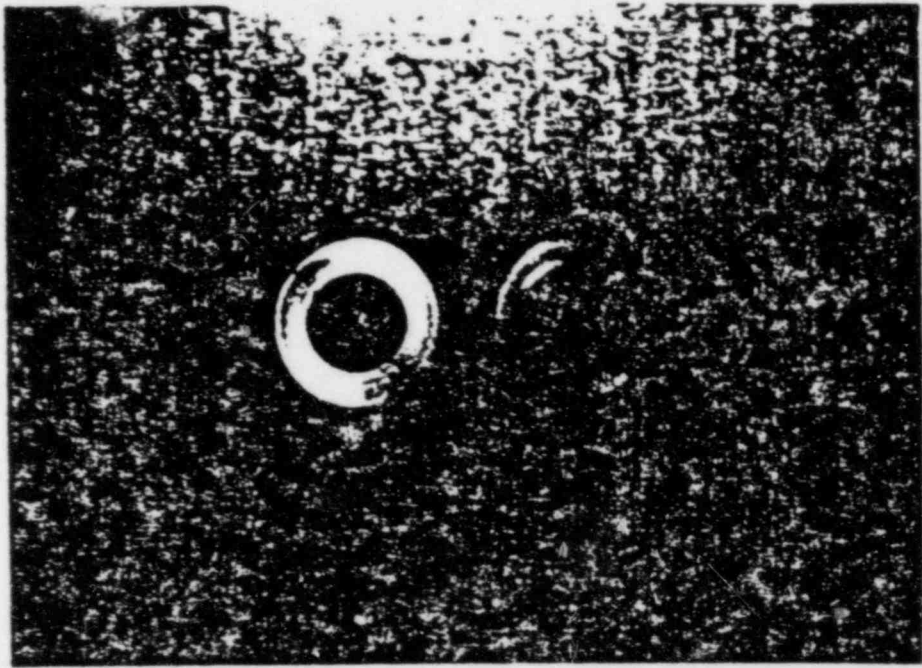




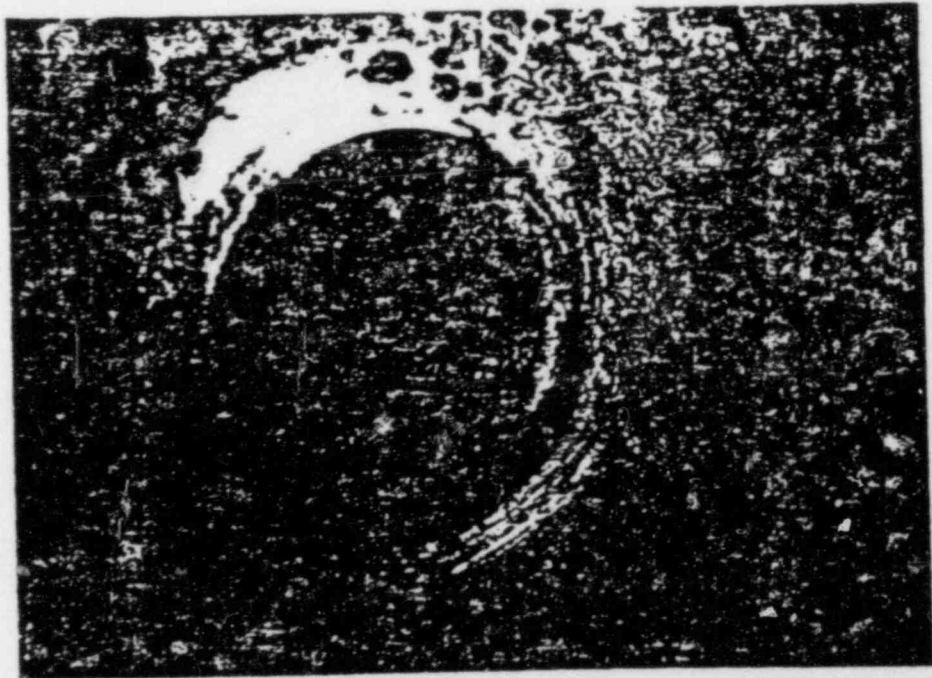
PHOTOGRAPH 13



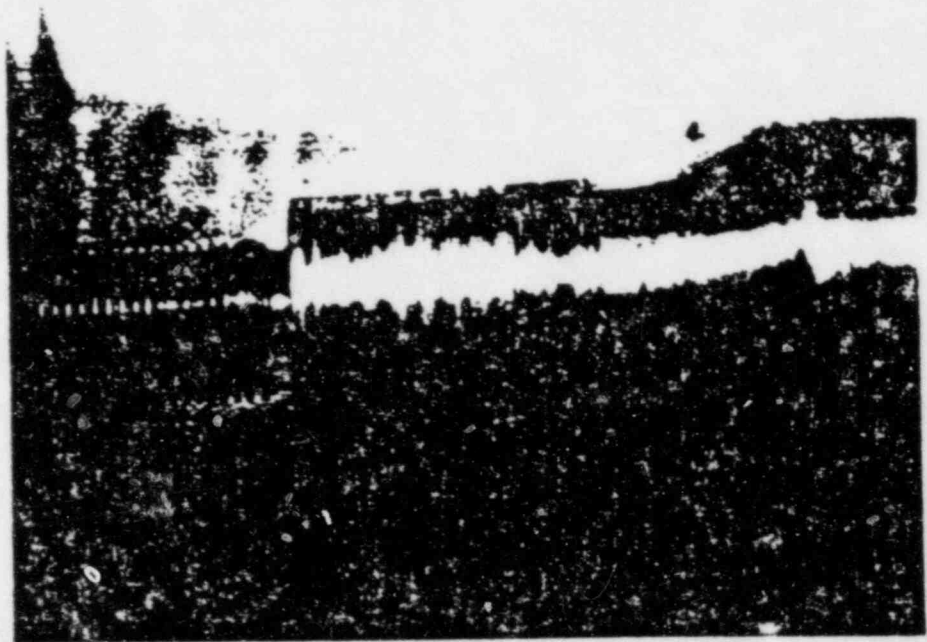
PHOTOGRAPH 24



PHOTOGRAPH 15



PHOTOGRAPH 16



PHOTOGRAPH 17



PHOTOGRAPH 18

## PHOTOGRAPH CAPTIONS

1. Pump from Unit 103.
2. Pump from Unit 103 with suction flange removed.
3. Impeller from Unit 102 pump.
4. Shaft from Unit 102 pump.
5. Retaining washer and nut from Unit 102 pump.
6. Bore area of impeller from Unit 102 pump.
7. Lip around impeller bore of Unit 102 pump.
8. Shaft taper where impeller was mounted, Unit 102 pump.
9. Shaft taper where impeller was mounted, Unit 102 pump.
10. Shaft flange showing metal transfer, Unit 102 pump.
11. Nut from Unit 102 pump.
12. Retaining washer from Unit 102 pump.
13. Impeller from Unit 103 pump.
14. Shaft from Unit 103 pump.
15. Retaining washer and nut from Unit 103 pump.
16. Bore area of impeller from Unit 103 pump.
17. Shaft taper where impeller was mounted, Unit 103 pump.
18. Wear ring in suction flange, Unit 103 pump.



**PEI** Parsons Peebles - Electric Products, Inc.

725 Clarkstone Road  
Cleveland, Ohio 44112  
Telephone: (216) 481-1500  
Telex: 241564

FAILURE ANALYSIS REPORT - EF-3060

ON

4375 kVA, 3500 kW, 4160V, 3-PH, 60 Hz, 450 RPM  
CLASS 1E SYNCHRONOUS ROTOR SERIAL NO. 17404267  
FOR LONG ISLAND LIGHTING COMPANY  
SHOREHAM PLANT, NEW YORK  
BUILT FOR  
TRANSAMERICA DELAVAL, INC.  
P.O. BOX 2273  
OAKLAND, CALIFORNIA 94614

CERTIFICATION

This is to certify that the contents of this report are a true and accurate statement of findings made during the inspection and diagnostic testing of damaged equipment performed by skilled personnel of Parsons Peebles-Electric Products, Inc., October 6 and 7, 1983.

Engineering interpretations of these findings are correct to the best of my professional judgement and supported by my own qualification as a Registered Professional Engineer, duly licensed to practice in the State of Ohio.

*Peter M. Silverberg*

Peter M. Silverberg, P.E.  
State of Ohio License No. E-041988  
Senior Engineer, Insulation

Parsons Peebles-Electric Products, Inc.  
Cleveland, Ohio  
October 11, 1983

## FAILURE ANALYSIS - ROTOR POLE 7; ROTOR 17404267

### 1. Summary

Engineering evaluation of the Class 1E equipment damage reported below resulted in the following conclusions:

- 1.1 The coil of Rotor pole #7 grounded as a result of mechanical damage to the insulation in the left rear upper corner of the pole. The steel pole body in that location had a sharp corner which was located sufficiently close to the winding.

Continued self-induced vibration allowed an opportunity for aforesaid corner to wear away the insulation resulting in a ground. This is classified as an irrelevant failure within the scope of Chapters 5 and 6 of IEEE Std. 308-1978 and totally unrelated to equipment design or materials.

- 1.2 Two rotor poles had corners knocked off the top washers. This is not a failure as the poles themselves are fully operational. This is classified as minor, repairable mechanical damage.
- 1.3 The equipment was found repairable. The broken washer corners were repaired per Engineering Specification ER-6.1 (Appendix B). Pole #7 was cleaned; its sharp corner rounded off; and its coil was rewound as per L-11027.

### 2. Introduction

- 2.1 Rotor 17404267 was found to have a ground by LILCO personnel during an inspection of diesel engine problems.
- 2.2 This rotor was shipped to PP-EP for repairs under our Service No. 4-4138. It arrived 10/5/83. The incoming inspection report (Appendix A) showed clearly a ground to Pole #7.
- 2.3 Pole #7 was removed from the shaft and a diagnostic test set-up. The diagnostic apparatus is a 1000W resistance in series with the 120 VAC feeder and the pole. Smoke from the ground was marked with an arrow on the head. The outer layers of wire were stripped from the pole down to the first turn. The first turn of wire was gently removed and the grounded spot located at the left rear corner. The three reference photos show: Figure 1 - the first turn. The ground is at the left rear corner. Figure 2 - A close up of the burnt wire. Figure 3 - the grounded spot on the pole body.

- 2.4 The insulating paper was removed from the pole body and a sharp corner was found at the spot of the ground. The sharp corner was located at the edge of the insulating washer which is why it did not cause a ground in manufacturing but required the vibration from running to wear through insulation.

### 3. Probable Mechanism of Failure

- 3.1 Based on the evidence described above, the failure of ground insulation of winding on Pole #7 was caused by accelerated wear of the pole body insulation due to normal self-induced vibration of the equipment. The rate of wear was accelerated by the improperly blended radius of the coil support edge, resulting in a severe concentration of pressure on the pole body insulation material.
- 3.2 Repetitive differential thermal expansion and contraction of the coil (copper) and the pole body (steel) resulted in the fluctuating pressure on the insulating material sandwiched between the pole and coil. This mechanism allowed abrasion due to self-induced vibration during the operation (when hot) and applied increased pressure at the worn spot upon cool-down, eventually leading to puncture.
- 3.3 With the exception of the incomplete blending of coil support surfaces during the edge grinding operation, there was no evidence of any abnormal conditions pertaining to design, material, workmanship or use. The incomplete blending of surfaces in one of the four corners is a typical example of "marginal" workmanship and it is difficult to detect by routine inspection and normal NDT methods.

### 4. Repairs

- 4.1 Chipped washers on poles were repaired as per ER-6.1.
- 4.2 Pole #7 was completely cleaned and the sharp corner carefully smoothed. It was then rewound and vacuum-pressure impregnated in MV-10.15 (Isochem S-100). It is current PP-EP practice to use a wire insulation of heavy enamel plus single daglass instead of double daglass as this substantially reduces the possibility of interturn shorts. This change was made on the new winding. The insulating paper now used is DMD which is preferred to "Duroid" used on the original pole.

3. Conclusions

Ground insulation failure of Pole #7 was due to excessive wear at a high pressure spot that resulted from marginal quality of workmanship in fabrication of the pole piece. The cause was eliminated during the repair.



INSPECTION REPORT

0-6-83

R-Order: 4-4138

: Transamerica Delaval

Type Equipment: Syn. Gen Rotor

11027

Frame: 170

Volts: 125

Amps: 252.6

50

Serial: 17404267-200

CONDITION OF UNIT WHEN RECEIVED

was received mounted inside a wooden crate. The rotor was covered with a sheet. The shaft was supported by wooden clamps, and protected with rubber

1. Megger Pole #7 - 0 Megs  
Remainder of Poles - 200 + Megs
2. Slip rings slightly grooved
3. 3 Fan lockplates are opened
4. Lockplates on field connections to slip rings were opened
5. Broken washers on Poles #13 & 14.
6. Both leads are unsoldered from Pole #6, and one lead is unsoldered from Pole #7.
7. "Separation" on Pole #6 either the washer was broken and when repaired did not return to original setting, or the lower layers of the windings pushed it away, no serious problem.
8. Marks from lifting cable on shaft - S. R. End. (Not ours)

removal of Pole #7 from spider, 1 stud came out.)

TEST RESULTS

Pole .272 OHMS @ 21.5 Deg C )

Pole on spider .282 OHMS @ 21.5 Deg C ) 10-7-83  
(for comparison)

Pole Lead to Ground 30990 OHMS ) WK 10-6-83

Pole Outside Lead to Ground 30990 OHMS)

SUPERSEDES : NEW

SHEET NO. 1 OF 2

SUPERSEDED BY

## REPAIR PROCEDURE (FIELD) FOR BROKEN POLE WASHERS

### Scope

This procedure is limited to repairing rotor pole washers that were broken as a result of mis-handling and where the pole winding itself is intact electrically with no visible signs of damage.

### Materials

MI-10.1 Glass-Polyester Laminate

MV-20.10 Epoxy Coil Sealant

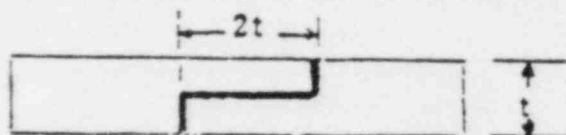
MS-10.2 Solvent - MEK

MS-10.4 Solvent - Acetone

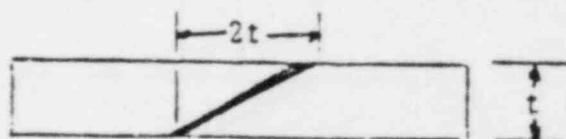
MV-10.5 Insulating Enamel

### Joint Design

The Service Technician shall cut the replacement piece and the remainder of washer to make joints which are sketched in order of desirability. The dark line is the adhesive. The underside of the joint may be reinforced with a thin stiffener piece of MI-10.1 if accessible.



Double buttlap - preferred



Scarf butt - usually practical



Plain butt - acceptable in small inaccessible areas only.

### 4.0 Surface Preparation

- 4.1 Roughen all surfaces for adhesive with 100 grit emery cloth.
- 4.2 Wipe off dust with a soft cloth.
- 4.3 Wash clean with acetone or methyl ethyl ketone.

# REPAIRING SPECIFICATION

CODE DATE SPEC. NO.

ER 1/25/78 6.1

SUPERSEDES... NEW

SHEET NO. 2 OF 2 SUPERSEDED BY

## REPAIR PROCEDURE (FIELD) FOR BROKEN POLE WASHERS

### Adhesion

- 5.1 Mix epoxy per instructions given in MV-20.10.
- 5.2 Spread to make adhesive joint and exposed conductor seal.
- 5.3 Clamp repair piece in place.
- 5.4 Cure for 16 hours at or above 70°F.
- 5.5 Sand any bumps smooth and restore finish coat.

### Quality Check

Lap two scraps of MI-10.1 together with excess MV-20.10 and let cure same as the washer. Try to break the adhesive joint. If of proper strength, the laminate will tear or break.

### Documentation

All field repairs per ER-6.1 performed on Class IE equipment are subject to inspection and certification by the authorized E.P. Service personnel. All materials used in the repairs must be certified as required by the applicable Material Specifications. E.P. warranty is conditional upon compliance with the above.

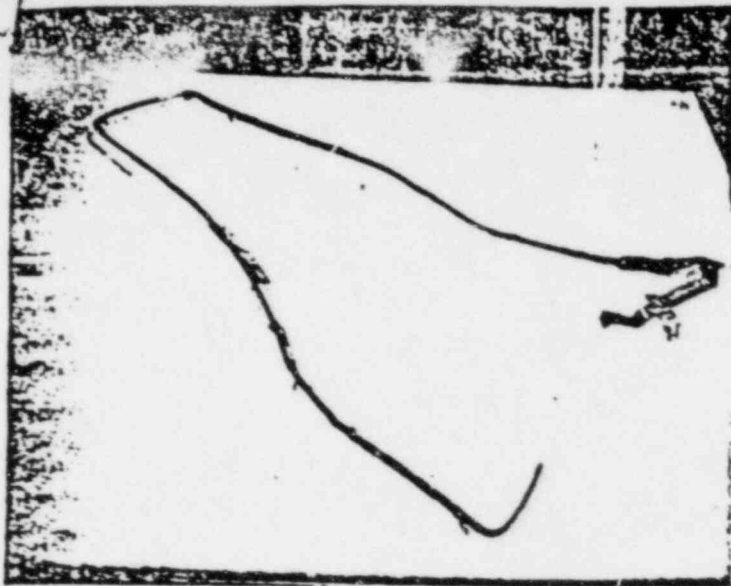


FIGURE 1



FIGURE 2

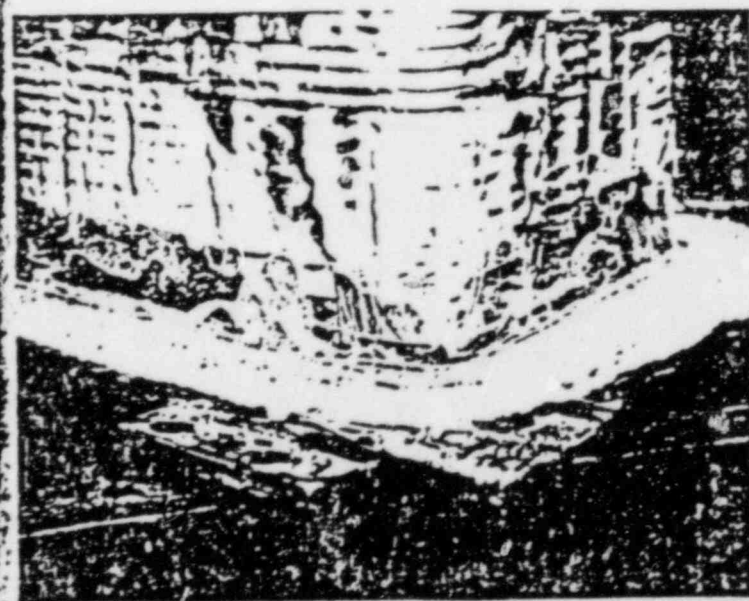


FIGURE 3



Failure  
Analysis  
Associates

ENGINEERING AND METALLURGICAL CONSULTANTS  
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FaAA-83-12-9  
M&T3/7396

ANALYSIS OF THE REPLACEMENT CONNECTING ROD BEARINGS  
FOR EMERGENCY DIESEL GENERATORS, FATIGUE LIFE PREDICTION  
SHOREHAM NUCLEAR POWER STATION

Prepared by  
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2225 East Bayshore Road  
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December 15, 1983

## 1.0 SUMMARY AND CONCLUSIONS

Four upper connecting rod bearing shells in two Transamerica Delaval Incorporated (TDI) Enterprise diesel engines at the Shoreham Nuclear Power Station (SNPS) were found to be cracked after about 250 hours of full load operation.

An earlier Failure Analysis Associates (FaAA) report [1] has been issued analyzing the cracking in qualitative terms. That report cited high peak oil film pressure, lack of bearing shell support at connecting rod chamfers, concentration of load at bearing ends, and voids 0.5mm to 0.7mm in diameter as contributing to the cracking.

Along with new crankshafts of modified design, new connecting rods and new connecting rod bearings have been installed in the the SNPS diesel engines. The new connecting rods have a smaller bore chamfer, eliminating the unsupported bearing ends. The increase in crankpin diameter from 11 inch to 12 inch was shown to reduce peak oil film pressure from 29,745 psi to 26,730 psi. This pressure is slightly above an industry-accepted guideline for peak value and suggests the need for fatigue lifetime calculations.

Subsequent finite element method (FEM) stress analysis and a fracture mechanics analysis of the fatigue cracking of the bearings have shown that the tensile stress in the bearings, that caused cracking in the original bearings, is reduced by 50.5% in the new bearings. The predicted fatigue life of the new bearings is 513,000,000 stress cycles, or 38,000 hours at full load, despite the fact that the peak oil film pressure is slightly above an industry guideline [3]. These industry guidelines are not absolute maximum allowable values. Some engine manufacturers successfully operate engine sleeve bearings above industry guidelines in specific applications, by exercising careful control of engine component design, manufacturing, and operating conditions. LILCO appears to be exercising the degree of control necessary for successful

operation at 26,780 psi peak oil film pressure. In addition, the FEM and fracture mechanics analysis of the connecting rod bearings, performed by FaAA, is a much more detailed analysis than is performed by engine builders and bearing suppliers in the course of normal applications engineering. This detailed analysis provides the basis for the calculated bearing fatigue life.

This expected fatigue life is conservatively calculated in that it does not include any reduction in edge loading of the bearings obtaining from the increased pin diameter and concomitant reduction in torsional yawing of the crankshaft pin.

The expected fatigue life is approximately an order of magnitude greater than the total anticipated full-load test time during the 40 year life of SNPS. Also, the routine maintenance procedures planned by LILCO require periodic inspection of all the surfaces, including nondestructive examination for flaws and bearing thickness measurement, each scheduled plant outage.

## 2.0 INTRODUCTION

An earlier report by Failure Analysis Associates (FaAA) [1] identified the primary causes of damage of some of the connecting rod bearing shells in the TDI Enterprise diesel engines at SNPS. Records indicate that after approximately 250 hours of operation, at or above 100% power, four of the twenty-four upper connecting rod bearing shells had cracked about  $\frac{5}{8}$ -inch from one end. These cracks extended radially through the thickness of the bearing and circumferentially to a length of approximately 4 inches.

Four factors contributing to the cracking were identified in the earlier report. First, the peak oil film pressure in the hydrodynamic oil film separating the crankshaft and the bearing exceeded the guidelines of a major independent supplier of engine bearings by 14% [2, 3]. Second, the geometry of the connecting rod bore left the end of the bearing unsupported, inducing cantilever bending. Figure 1 shows the configuration of the connecting rod relative to the bearing. Third, the contact patterns in the electroplated babbitt overlay on the bearing inner diameter showed that the cracked bearings had been subjected to edge loading, or a concentration of load on the

bearing ends due to lack of parallelism between the crankshaft journal and the bearing surface. The fourth cause was thought to be the presence of voids ranging in size from 0.5mm to 0.7mm.

The failure analysis of the connecting rod bearing shells indicated that voids in the size range of 0.5mm to 0.7mm were the initiation sites for the cracks that formed. However, analysis since issuance of the initial report showed that these voids are not atypical of cast aluminum bearings, and in the absence of abnormally high stresses would not normally be detrimental to bearing life.

The computations described in this report were performed in order to develop a conservative estimate of the expected life of the new connecting rod bearings in the TDI Enterprise diesel engines. Along with the new crankshafts, new bearings and new connecting rods have been put into the engines. Two of the causes of the bearing cracking have thereby been addressed: the unsupported bearing ends have been eliminated with the new components, as shown in Figure 2, and the calculated peak oil film pressure has been reduced to 26,780 psi in the new connecting rod bearings [2]. Through finite element stress analysis and fracture mechanics calculation of fatigue crack growth, the fatigue lifetime of the new configuration can be estimated to determine a suitable inspection or replacement interval for the connecting rod bearings.

### 3.0 BEARING STRESS ANALYSIS

Finite element analysis of both the original and replacement connecting rod bearings was performed using the ANSYS code. The results of the journal orbit analysis [2] were used as the basis for the applied loads on the bearing. Since the journal orbit analysis assumes perfect parallelism between the bearing and the journal [4], the pressure distribution was skewed toward the end of the bearing to correlate with the contact patterns in the babbitt. The loading was skewed so that 82.6% of the applied load is carried on the outer 28.2% of the bearing length. Both the cast aluminum bearing shell and the forged steel connecting rod were included in the finite element model. In addition, the compressive preload on the bearing resulting from the interference fit of the bearing in the connecting rod was included in the model.



The maximum tensile stress was found to occur in the longitudinal direction, at the inner surface of the bearing shell, at a node 0.879 inch from the end of the bearing. The values of these stresses are listed in Table 1.

Table 1

Maximum Tensile Stress:  
Longitudinal direction, on bearing inner diameter

Original bearings, 11 inch diameter crankpins:  
tensile stress - 10,931 psi

New bearings, 12 inch diameter crankpins:  
tensile stress - 5,412 psi

The maximum tensile stress in the new bearings is predicted to be only half the stress in the original bearings that cracked after about 250 hours of full-load operation. About one-fifth of the reduction in stress results from a reduction in the calculated peak oil film pressure, a direct consequence of the larger journal diameter. The remaining four-fifths of the reduction in stress is directly attributable to the elimination of the unsupported bearing ends via reduction of the bore chamfer in the new connecting rods.

#### 4.0 BEARING LIFE PREDICTION

The known behavior of aluminum in response to cyclic stressing can be used to predict the fatigue life of the new bearings installed in the TDI Enterprise diesel engines. In the elastic strain, high-cycle-fatigue region (number of cycles greater than  $10^6$ ), the behavior of aluminum can be described by the equation [5]:

$$\sigma_a = \sigma_f' (2N)^b$$

where  $\sigma_a$  = stress amplitude

$\sigma_f'$  = fatigue strength coefficient

N = number of cycles

b = fatigue strength exponent.

The stress amplitude for the connecting rod bearings is one-half of the maximum tensile stress computed by the FEM analysis described in Section 3.0. The coefficient  $\sigma_f'$  has not been determined for the B850 aluminum bearing alloy, but in using the ratio of the stress amplitudes to compute the ratio of the number of cycles to failure, the coefficient drops out of the expression. The fatigue strength exponent, b, also has not been determined for the B850 alloy, but from work on a wide variety of metals and aluminum alloys, it has been determined that the value for b is in the range of -0.06 to -0.14. The most conservative computation is to use b = -0.14, which yields the smallest change in N for a given change in  $\sigma_a$ .

$$\frac{\sigma_a \text{ (New bearing)}}{\sigma_a \text{ (Old bearing)}} = .495 = \frac{\sigma_f' (2N \text{ (New bearing)})^{-0.14}}{\sigma_f' (2N \text{ (Old bearing)})^{-0.14}}$$

$$\frac{N \text{ (New bearing)}}{N \text{ (Old bearing)}} = 152$$

This calculation predicts that the new bearings should not fail by fatigue until they have experienced 152 times the number of cycles that failed the original bearings.

The connecting rod bearings are subjected to one stress cycle in every two rotations of the crankshaft, or 225 cycles per minute. The original bearings were cracked after approximately 250 hours, or 3,375,000 cycles. The new bearings would not be expected to begin to exhibit cracking until after 513,000,000 cycles, or 38,000 hours of full-load operation have occurred.

## 5.0 DISCUSSION

Calculations have demonstrated that the major contributor to the cracking of the original connecting rod bearings in the TDI Enterprise diesel engines was the unsupported bearing end, the result of a 0.25 inch chamfer in the connecting rod bore.

Eliminating this unsupported end, along with lowering peak oil film pressure by 10%, results in a predicted fatigue life of 38,000 hours at full-load. This life is approximately a factor of 10 greater than the expected time of full-load operation during the life of the plant. Consequently, FaAA is able to conclude that the connecting rod bearings have adequate design fatigue lifetime without the need for replacement during normal plant operation despite the fact that the oil film pressure is still slightly above an industry guideline for peak value [3].

The fatigue life of the new bearings has been conservatively calculated in that no reduction in yawing of the crankpin journals relative to the bearings has been assumed; such reduction is expected as a consequence of the increased torsional stiffness of the new crankshaft. This yawing contributes to the edge loading that was evident on every cracked bearing.

No change in materials properties or structure was assumed. The 38,000 hour predicted life for the new bearings is in the presence of the 0.5mm to 0.7mm voids found in the old bearings. As a check on the influence of the voids, the stress intensity factor range,  $\Delta K$ , was computed for these voids and the stresses computed by FEM analysis. For the original bearings,  $\Delta K \approx 1.8 \text{ ksi} \sqrt{\text{in}}$ . For the new bearings,  $\Delta K \approx 0.9 \text{ ksi} \sqrt{\text{in}}$ . The threshold value of  $\Delta K$  for growth of a pre-existing void in fatigue is not known precisely for this alloy, but in comparison to other aluminum alloys, is estimated to be approximately [5]  $\Delta K_{th} \approx 2.0 \text{ ksi} \sqrt{\text{in}}$ . [5]. Therefore, since the  $\Delta K$  value for the new bearings is below the threshold value for growth of pre-existing voids, this presence of 0.5mm to 0.7mm voids will not have an impact on fatigue cracking.

The state of stress in the old bearings is close to that required to cause the voids to initiate cracks, while the state of stress in the new bearings is well below that necessary to initiate fatigue cracks at the voids.

The inspection procedure at Shoreham planned by LILCO calls for inspection of all surfaces of the connecting rod bearings, including nondestructive inspection for flaws and measurement of the thickness in six places, including the ends subjected to edge loading, during every scheduled refueling outage.



## REFERENCES

1. Failure Analysis Associates "Emergency Diesel Generator Connecting Rod Bearing Failure Investigation, Shoreham Nuclear Power Station, Palo Alto, California, October 31, 1983.
2. Journal Orbit Analyses of TDI Enterprise R-48 Diesel Engine performed by Imperial Clevite Inc. for Failure Analysis Associates, October 6, 1983.
3. W. A. Yanraus (Manager of Product Analysis, Imperial Clevite Inc., Engine Parts Division), private communication with L. A. Swanger (FaAA), October 4, 1983.
4. Ross, J. M. and R. R. Slaymaker, "Journal Center Orbits in Piston Engine Bearings," SAE Paper 690114, Society of Automotive Engineers, Warrendale, Pennsylvania, 1969.
5. H. O. Fuchs and R. I. Stephens, Metal Fatigue in Engineering, John Wiley & Sons, New York, 1980, p. 78.

NOTE: drawing at .74 of original

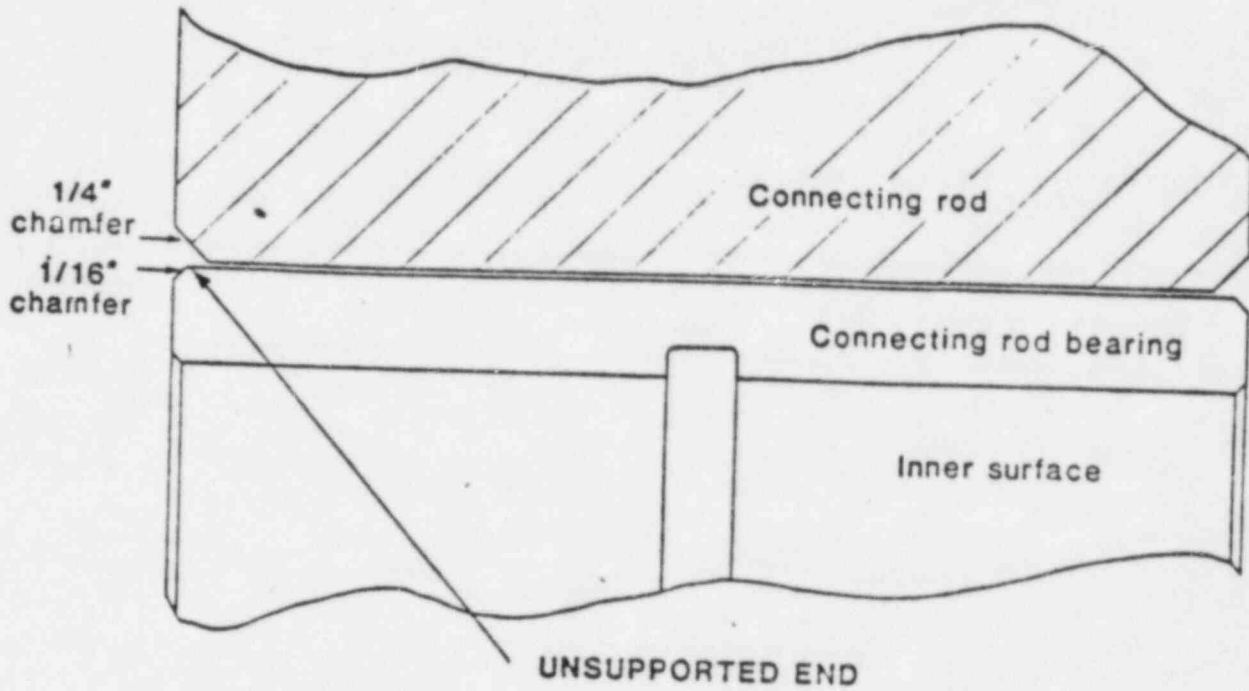


Figure 1. Bearing: Connecting rod configuration with original 11" journals.

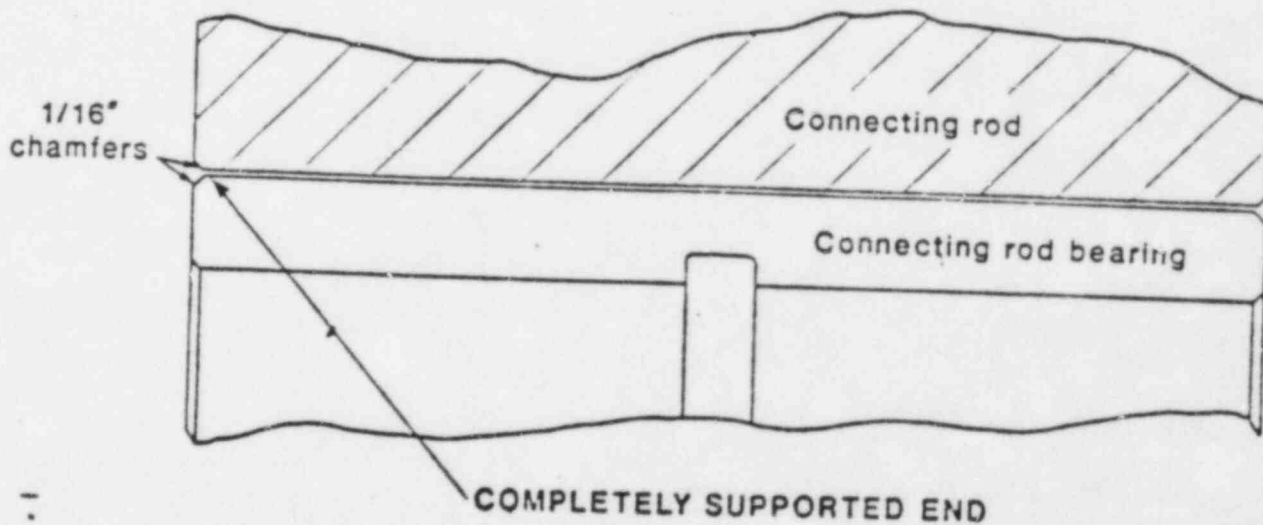


Figure 2. Bearing: Connecting rod configuration with replacement 12" journals.

CERTIFICATE OF SERVICE '84 FEB -9 A10:39

In the Matter of  
LONG ISLAND LIGHTING COMPANY  
(Shoreham Nuclear Power Station, Unit 1)  
Docket No. 50-322 (OL)

OFFICE OF THE  
DOCKETING & SERVICE  
BRANCH

I hereby certify that copies of LILCO's Response to Suffolk County's Motion to Admit Supplemental Diesel Generator Contentions were served this date upon the following by first-class mail, postage prepaid, or by hand as indicated by an asterisk:

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