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

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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 26,780 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 $^5/_8$-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 σ_a = stress amplitude

σ_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 σ_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 σ_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 occured.

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 33,000 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, ΔK, was computed for these voids and the stresses computed by FEM analysis. For the original bearings, $\Delta K \approx 1.8$ ksi $\sqrt{in.}$ For the new bearings, $\Delta K \approx 0.9$ ksi $\sqrt{in.}$ The threshold value of ΔK 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] $\Delta K_{th} \approx 2.0$ ksi $\sqrt{in.}$ [5]. Therefore, since the ΔK 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.

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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.

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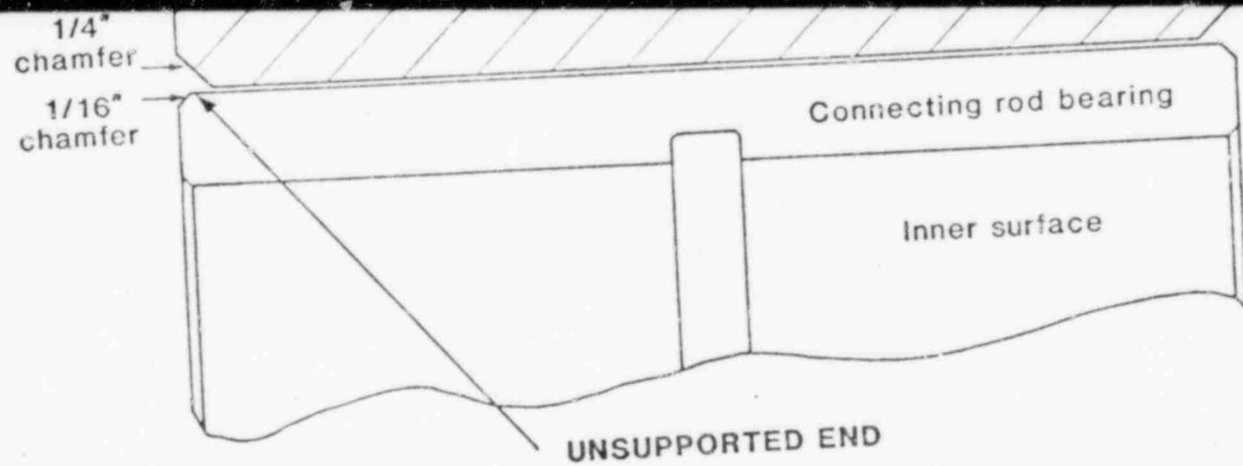

Figure 1. Bearing: Connecting rod configuration with original 11" journals.

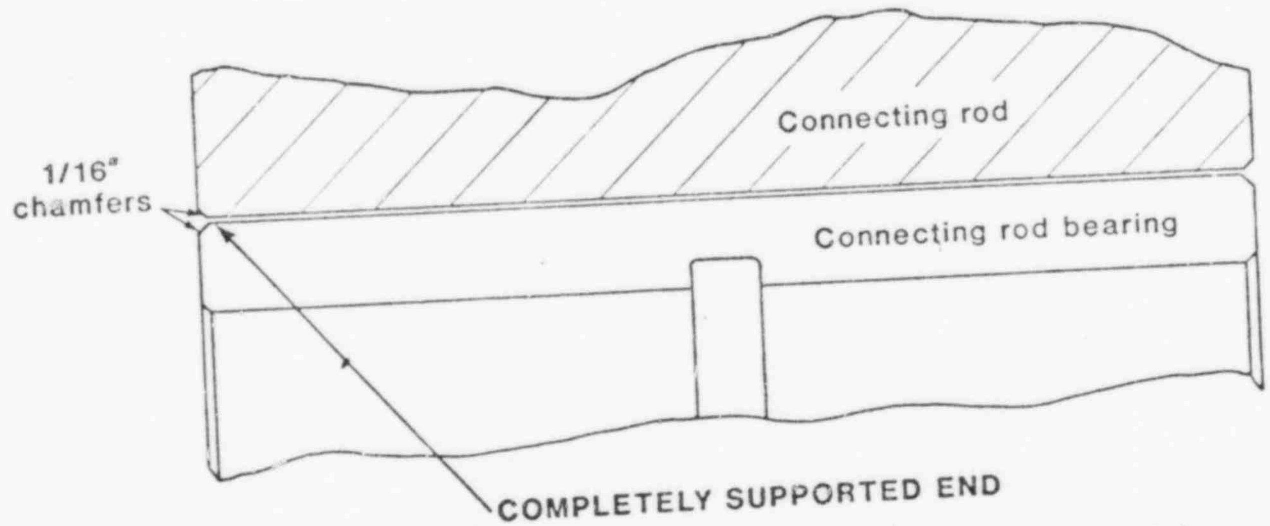

Figure 2. Bearing: Connecting rod configuration with replacement 12" journals.

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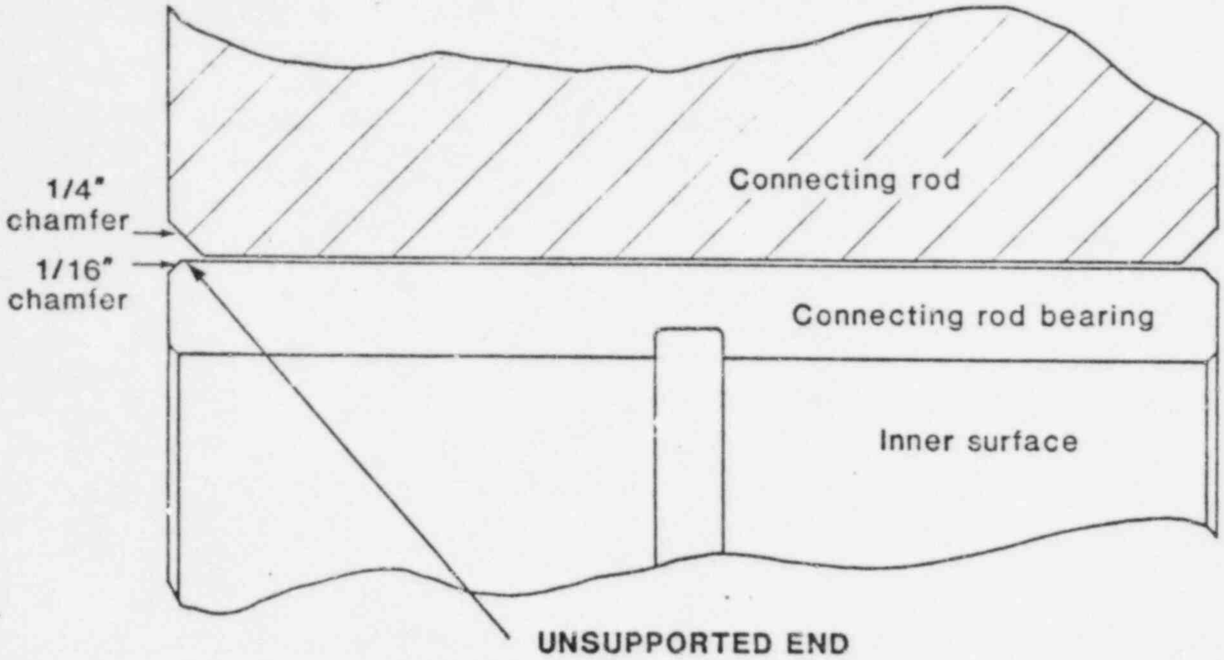
NOTE: drawing at .74 of original



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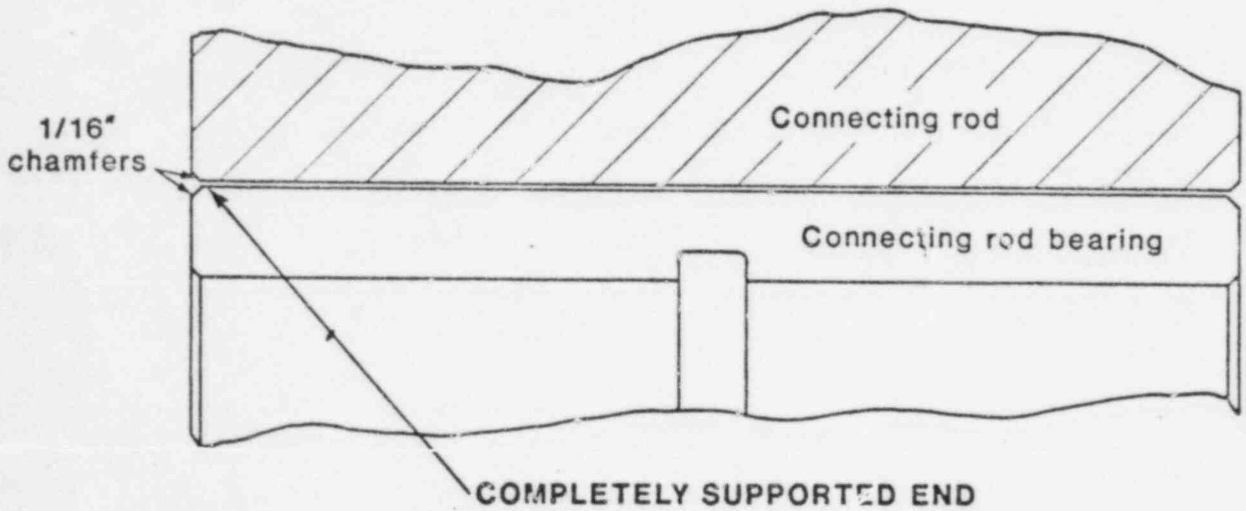

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