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UNITED STATES OF AMERICA
NUCLEAR REGULATORY COMMISSION

Before the Atomic Safety and Licensing Board

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

SUFFOLK COUNTY AND STATE OF NEW YORK PROPOSED FINDINGS OF FACT AND CONCLUSIONS OF LAW ON EMERGENCY DIESEL GENERATORS (REPLACEMENT CRANKSHAFTS)

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I. INTRODUCTION

A. Background and Summary of Conclusions

1. Statement of Contention and Scope of Partial Initial Decision.

1. This Partial Initial Decision addresses the Suffolk County and the State of New York contention that the replacement crankshafts on the Shoreham emergency diesel generators ("EDGs") are not adequate for operating at full load (3500 kW) or overload (3900 kW). The Intervenors thus urge this Board to find that the EDGs fail to satisfy General Design Criterion ("GDC") 17.

2. The first paragraph of the EDG Contention states:

> Contrary to the requirements of GDC 17, the emergency diesel generators at Shoreham ("EDGs") manufactured by Transamerica Delaval, Inc. ("TDI") will not operate reliably and adequately perform their required functions because the EDGs are overrated and undersized, improperly designed, and not satisfactorily manufactured. There can be no reasonable assurance that the EDGs will perform satisfactorily in service and that such operation will not result in failures of other parts or components of the EDGs due to the overrating and insufficient size of the EDGs or design or manufacturing deficiencies. The EDGs must therefore be replaced with engines of greater size and capacity, not designed or manufactured by TDI.

Anderson, et al., ff. Tr. 23,826, at 11.

3. With respect to the Shoreham crankshafts, the EDG Contention alleges that its first paragraph is supported because:

> (a) The replacement crankshafts at Shoreham are not adequately designed for operating at full load (3500 kW) or overload (3900 kW), as required by FSAR Section 8.3.1.1.5, because they do not meet the standards of the American Bureau of Shipping, Lloyd's Register of Shipping, or the International Association of Classification Societies. In addition, the replacement crankshafts are not adequately designed for operating at overload, and their design is marginal for operating at full load, under the German criteria used by F.E.V.
>
> (b) The shotpeening of the replacement crankshafts was not properly done as set forth by the Franklin Research Institute report, Evaluation of Diesel Generator Failure at Shoreham Unit 1, April 6, 1984, and the shotpeening may have caused stress nucleation sites. The presence of nucleation sites may not be ascertainable due to the second shotpeening of the crankshafts.

Id. at 106.

4. This Partial Initial Decision addresses only the foregoing crankshaft-related portions of the EDG Contention. The Board will address the remaining portions of the contention (chiefly concerning the adequacy of the EDG cylinder blocks) in a subsequent decision.1/

1/ In the context of this Partial Initial Decision, the parties have sponsored the following testimony and witnesses:

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2. Background to Litigation of Crankshaft Adequacy

5. The Shoreham EDGs are Transamerica Delaval, Inc. ("TDI") model DSR-48 diesel engines with 8 cylinders in line, having a 17-inch bore and 21-inch stroke. The EDGs constitute the onsite electrical power system for the Shoreham plant. The EDGs are intended to provide reliable onsite emergency power to the Shoreham plant in conformity with 10 C.F.R. Part 50, Appendix A, GDC 17. Hubbard and Bridenbaugh, ff. Tr. 23,826, at 12, 14.2/

(Footnote cont'd from previous page)

LILCO presented testimony on the adequacy of the replacement crankshafts by Drs. Pischinger, Chen, Johnston and McCarthy, and Messrs. Montgomery and Youngling; and on the shotpeening of the replacement crankshafts by Drs. Wells, Johnson and Wachob, and Messrs. Cimino, Seaman and Burrell. This testimony follows pages 22,610 and 23,122 of the transcript, respectively. Suffolk County presented testimony on the adequacy of the replacement crankshafts and shotpeening by Dr. Anderson, Professor Christensen and Messrs. Eley, Bridenbaugh and Hubbard. This testimony follows page 23,826 of the transcript. The Staff presented testimony on the adequacy of the replacement crankshafts by Professor Sarsten and Mr. Henriksen; and on the fabrication process of the replacement crankshafts and shotpeening by Dr. Bush. This testimony follows page 23,126 of the transcript.

2/ In pertinent part, GDC 17 specifies that in the assumed absence of the offsite electrical power system, the EDGs must:

> provide sufficient capacity and capability to assure that (1) specified acceptable

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6. The adequacy of the Shoreham replacement crankshafts is a significant issue in this proceeding because the original EDG crankshafts failed during testing. The EDGs now have replacement crankshafts with 13-inch diameter main bearing journals, 12-inch (nominal) diameter crank pins, and 3/4-inch crank pin fillet radii. The original crankshafts had 11-inch (nominal) diameter crank pins and 1/2-inch crank pin fillet radii. The replacement crankshafts were installed after the original crankshaft on EDG 102 fractured into two pieces during an engine test run on August 12, 1983. Subsequent inspections identified cracks in the crankshafts of EDG 101 and EDG 103 as well. LILCO's consultant, Failure Analysis Associates ("FaAA"), later concluded that the original crankshafts were inadequately designed and failed due to high cycle torsional fatigue. Anderson, et al., ff. Tr. 23,826, at 106-07; Johnston and McCarthy, ff. Tr. 22,610, at 7-8.

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> fuel design limits and design conditions of the reactor coolant pressure boundary are not exceeded as a result of anticipated operational occurrences and (2) the core is cooled and containment integrity and other vital functions are maintained in the event of postulated accidents.

10 C.F.R. Part 50, Appendix A, GDC 17.

7. The Shoreham FSAR specifies that the EDGs must have sufficient load carrying capability to satisfy a continuous and overload performance rating. Section 8.3.1.1.5 of the FSAR requires each EDG to be rated to operate continuously (8,760 hours, or one year) at full load of 3500 kW (with maintenance intervals as required by the manufacturer) and for 2 hours per every 24 hours at overload of 3900 kW (without reducing the maintenance interval established for the continuous rating). Hubbard and Bridenbaugh, ff. Tr. 23,826, at 14-16; FSAR § 8.3.1.1.5.

8. The purpose of the rating requirement for the EDGs is to provide necessary conservatism and confidence that the maximum actual power demands will reliably be met and that, accordingly, the requirements of GDC 17 will be fulfilled. Therefore, the proper criterion for whether the EDGs and their components can satisfactorily withstand operating conditions is whether they can be expected to operate at the rated levels without experiencing failures or incipient failures. Hubbard and Bridenbaugh, ff. Tr. 23,826, at 16. Thus, in judging the adequacy of the replacement crankshafts, this Board has considered whether the crankshafts can operate reliably at the 3500/3900 kw load levels specified in the FSAR.

3. Summary of Decision

9. This Board has carefully considered whether LILCO has established by a preponderance of the evidence that the replacement crankshafts are adequately designed for operating in the EDGs at full load and overload.3/ For reasons set forth below, we hold that LILCO has failed to meet its burden and thus we rule in favor of the Intervenors on the replacement crankshaft issue.

10. In reaching this decision, the Board has been confronted by a difficult problem of attempting to identify a standard or standards against which the adequacy of the replacement crankshafts should be measured. GDC 17 is not prescriptive, i.e., it does not set forth a definitive, quantifiable standard by which to judge the adequacy of a crankshaft. In the absence of such a definitive standard in the regulations, the parties have suggested a number of means for assessing the adequacy of the replacement crankshafts. For

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3/ Once a prima facie case is established (as clearly has been done in this instance), LILCO must establish by a preponderance of the evidence that the replacement crankshafts are adequate. See Tennessee Valley Authority (Hartsville Nuclear Plant, Units 1A, 2A, 1B, and 2B), ALAB-463, 7 NRC 341, 360 (1978); Louisiana Power & Light Co. (Waterford Steam Electric Station, Unit 3), ALAB-732, 17 NRC 1076, 1093 (1983).

example, Suffolk County has suggested that the rules of classification societies (such as Lloyd's Register of Shipping ("Lloyd's"), the American Bureau of Shipping ("ABS"), and the International Association of Classification Societies ("IACS")) have appropriate criteria against which to judge the Shoreham crankshafts. LILCO urges that we ignore the classification societies and rely instead on the recommendations of the Diesel Engine Manufacturer Association ("DEMA") and on engineering fatigue analyses. The NRC Staff has generally urged that we rely on DEMA recommendations and that the adequacy of the crankshafts can also be measured by operating the EDGs at full rated load for approximately 740 hours.

11. We do not completely accept any of these suggestions. First, we do not adopt the classification society rules as particular standards which must necessarily be satisfied in order to comply with GDC 17. However, we do find that assessment of LILCO's compliance with these rules is relevant in reaching an overall judgment on the adequacy of the Shoreham crankshafts. In particular, these rules represent years of accumulated experience on crankshaft adequacy. Although the classification rules pertain primarily to diesels in marine service, the evidence indicates that marine diesel generators and nuclear plant diesel generators are subjected to basically the same stresses.

We would be remiss, therefore, if we ignored data regarding whether the Shoreham crankshafts would comply with any of these rules.

12. The evidence is clear that the Shoreham replacement crankshafts comply with few, if any, of the rules of the classification societies. In most cases, the noncompliance is undisputed. In a few instances, the status of compliance is unclear; at best, the compliance would be marginal, which is hardly the level of conservative design one would seek for a diesel in nuclear service. While marine diesel generators vary in some details from diesel generators used in nuclear service, we find that a diesel generator to be used in a nuclear power plant generally should be required to meet at least as stringent requirements as marine diesel generators. Therefore, the fact that the replacement crankshafts fail to satisfy classification society rules constitutes strong support for our view that LILCO has failed to establish by a preponderance of the evidence that the replacement crankshafts are satisfactory for nuclear service.

13. Second, based on the evidence of record, we cannot find that the replacement crankshafts comply with the DEMA recommendations. LILCO's proof was seriously deficient in this

regard. For example, the record is confused regarding whether the DEMA recommendations are up-to-date, whether the DEMA recommendations even constitute an appropriate standard for judging crankshaft adequacy, and how the DEMA recommendations are to be interpreted. On this record, the Board cannot find even that the DEMA recommendations are appropriate to be considered. In addition, even assuming arguendo that it is appropriate to attempt to judge the Shoreham replacement crankshafts against the DEMA recommendations, we find that, at best, the evidence is in sharp conflict as to whether the replacement crankshafts comply with the DEMA recommendations, with the LILCO and Staff experts taking opposite positions. We find that the Staff's interpretation of the DEMA recommendations is logical and more reliable than LILCO's and thus find, based on the Staff testimony, that the replacement crankshafts do not satisfy DEMA. Therefore, this evidence cannot support a finding that the replacement crankshafts are adequate.

14. Third, LILCO also has urged that a fatigue analysis performed by FaAA provides sufficient assurance of the adequacy of the replacement crankshafts. On the basis of this fatique analysis, FaAA testified that the replacement crankshafts had a 1.48 safety factor which it claims would be sufficient to support operation of the EDGs with the replacement crankshafts.

We note that the Staff does not support LILCO on this point, arguing that more than just a fatigue analysis is required in order to demonstrate that the crankshafts are adequate. We agree with the Staff. The FaAA analysis is heavily based upon the failure of the original crankshafts and thus represents only a single point of reference, rather than a widespread analysis based on extensive operating data. The analysis also inadequately considers the effect of the fabrication process on the material properties of the replacement crankshafts. Finally, some of the input data used in the FaAA analysis appears to be inconsistent with and substantially less conservative than similar data prepared by other LILCO consultants, making it impossible to assess the accuracy of the FaAA safety factor calculation. We therefore find that the FaAA fatigue analysis is not sufficient to demonstrate the adequacy of the replacement crankshafts.

15. Fourth, LILCO has also urged us to find that the replacement crankshafts are adequate because they have been shotpeened -- a technique for increasing the fatigue strength of metals. We decline to give weight to this evidence, particularly because there is no reliable expert testimony on how much, if at all, the shotpeening process increases the strength of the crankshaft material. Indeed, LILCO's lead consultant

specifically declined to quantify the amount of increased strength attributable to the shotpeening of the replacement crankshafts. Without any reliable quantification, we cannot find that shotpeening has resulted in any significant increase in the capability of the replacement crankshafts.

16. Finally, there is no evidence to support a finding that the adequacy of the replacement crankshafts has been established by testing for a sufficient number of hours at full rated load and overload. In contrast to the 750 hours of operation at full load suggested by the Staff (Sarsten, ff. Tr. 23,126, at 17), none of the replacement crankshafts has operated for more than 205 hours at or above 3500 kW. LILCO Ex. C-7; Montgomery and Chen, ff. Tr. 22,610, at 13. Nor is there any evidence of sufficient testing of the same design crankshaft at full rated load and overload at other installations. Therefore, LILCO has again failed to meet its burden of demonstrating a basis for finding the crankshafts to be adequate.

17. The detailed bases for our decision are set forth below. We deem it appropriate at the outset, however, to express concern about the November 5 proposed findings filed by LILCO. We directed the parties to file findings that did not

merely repeat direct testimony but. rather, which dealt with all sides of each issue.[4] LILCO's proposed findings did not comply fully with the Board's order. In large part, LILCO's findings are merely extracted from its direct testimony with little discussion of the testimony of the witnesses for the County and the Staff or the cross-examination of LILCO's witnesses. Moreover, LILCO frequently fails to state and justify why its proposed findings should be adopted when there is

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4/ In our August 1, 1984 order, this Board directed the parties to file proposed findings of fact and conclusions of law consistent with the format established by the Laurenson Board in Sections I through VII of the "Memorandum and Order Establishing Format and Schedule of Proposed Findings of Fact and Conclusions of Law," (Emergency Planning Proceeding), slip op. at 1-6 (July 27, 1984). In pertinent part, that Memorandum and Order directed:

> [T]he findings and conclusions should be concise, fair and well reasoned. Proposed findings which are complete, accurate, balanced and supported by the evidentiary record have the best chance of being relied upon by the Board. Proposed findings which are extracted from one party's written testimony, with little or no discussion or evaluation of other testimony and the cross-examination, are unlikely to be complete and balanced. Indeed, we expect the parties to state and justify their reasons for a proposed finding that a particular fact should be adopted rather than a contrary fact proposed by another party.

Id. at 1-2.

conflicting testimony of the County and/or the Staff on the same subjects.5/ Thus, this Board has been forced to canvass the record in detail itself.

5/ For example, LILCO's Proposed Finding number 15 states that "[t]he rules of the classification societies are for engines designed to operate in marine applications. Marine engines are exposed to conditions far different from those of standby engines at nuclear power plants." LILCO Findings at 6-7. In proposing this finding, LILCO completely ignores the County's testimony that the rules of the classification societies also are used to evaluate the adequacy of land-based diesel generators (Tr. 23,979-80 (Christensen and Eley)), that there are no major design differences between land-based and marine diesel generators (Tr. 24,207-09, 24,211-12 (Christensen, Eley)), and that marine diesel generators (as opposed to main propulsion systems) are subjected to essentially the same stresses as land-based diesel generators. Tr. 23,981-98 (Christensen, Eley). See also discussion in Section II.A.1, infra. LILCO offers no justification why this Board should choose its proposed finding over contrary evidence in the record.

II. DISCUSSION

18. We have been presented with four related theories regarding how to judge the adequacy of LILCO's replacement crankshafts: the classification societies rules; DEMA recommendations; fatigue analyses; and shotpeening. As noted above, after review of the evidence, we hold that LILCO has failed to establish by a preponderance of the evidence that the replacement crankshafts are adequate. We address below the bases for this decision.

A. The Replacement Crankshafts do not Comply with the Classification Societies' Rules

1. The Classification Societies' Rules are Relevant in Evaluating the Replacement Crankshafts.

19. Classification societies, such as Lloyd's and ABS, and other organizations such as the IACS, have formulated design rules for diesel engines in marine service. These rules represent the experience of each organization on the design/analysis procedures, materials, fabrication techniques, and testing methods that would produce an adequate engine design. These rules have evolved over time as new design techniques, materials, and fabrication methods have developed. Henriksen and Sarsten, ff. Tr. 23,126, at 9-10; Tr. 22,689 (Chen); 24,270 (Christensen).

20. The parties disagree about the relevancy of the classification societies' rules to an evaluation of the adequacy of the replacement crankshafts. The County urged that the classification society standards should be applied to determine the adequacy and reliability of the replacement crankshafts. The County's witnesses stated that these standards embody the only comprehensive collection of meaningful guidelines controlling crankshaft design in diesel engines that are used in applications where reliability is a significant evaluation factor. Christensen and Eley, ff. Tr. 23,826, at 111. Thus, although the County witnesses did not suggest that this Board should adopt the rules of any particular classification society as the ideal standard to evaluate the adequacy of the replacement crankshafts, the County argued that the classification societies' rules do provide pertinent guidance for applications such as at Shoreham where reliability is a significant evaluation factor. *Id.* at 109, 113-14.

21. The Staff's position is that it is generally not necessary "for good design practice" that the EDGs comply with the rules of the classification societies. Henriksen and Sarsten, ff. Tr. 23,126, at 10. According to the Staff, the classification societies' rules apply to engines designed to operate in marine applications, and marine engines are exposed to

different conditions than standby diesel engines at nuclear power plants. *Id*. at 11. Nonetheless, the Staff agrees that the rules represent a large amount of data and experience with crankshafts in diesel engines. *Id*. at 10; Sarsten, ff. Tr. 23,126 at 16-17; Tr. 23,467, 23,525 (Sarsten). Moreover, although the Staff believes that the ultimate test of the adequacy of any crankshaft is testing for approximately 740 hours at full load, the Staff prefers to assess the adequacy of the replacement crankshafts under the classification societies' rules rather than relying on FaAA's fatigue analysis and calculation of a factor of safety (discussed further in Section II.C, *infra*). Sarsten, ff. Tr. 23,126, at 16-17; Tr. 23,479 (Henriksen, Sarsten).

22. LILCO's witness agreed that compliance with the rules of a classification society generally provides assurance of the adequacy of a crankshaft, but argued that noncompliance is not significant if there is assurance of adequacy from other sources, such as testing or detailed engineering analysis. In addition, the LILCO witness also argued that classification society rules for marine diesels are more stringent than the rules for stationary land-based engines because operating conditions at sea are more severe. Thus, LILCO argued that a stationary engine may fail to comply with a classification society

rule but still be adequate for its intended nuclear service. Chen, ff. Tr. 22,610, at 15-16.

23. We find that the classification societies' rules are relevant to an evaluation of the adequacy of the replacement crankshafts. The rules of the classification societies and the IACS are based upon many years of practical experience with allowable stress levels for crankshafts and upon a very large data base of crankshaft failures. Tr. 23,467, 23,525 (Sarsten).6/ In reviewing crankshafts, the classification societies take into consideration many significant factors, including the actual stresses imposed on the crankshafts, the strengths of the material, and the forging process, and evaluate that information in light of the years of experience with successful and failed crankshafts. Tr. 23,526, 23,528 (Sarsten). The rules are based on the premise that crankshafts that comply with the rules will exhibit infinite fatigue life and operate safely below their fatigue limits even when aberrations that may occur are taken into consideration. Tr. 23,526-27 (Sarsten). We thus agree with the County that

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6/ For example, the IACS rules are based on an extensive evaluation of the stress levels, conditions of failure and other aspects of the failure of one hundred crankshafts. Tr. 23,467 (Sarsten).

compliance with the rules of the major classification societies generally provides assurance that the crankshafts in diesel engines are designed adequately. Conversely, if a crankshaft in a diesel generator at a nuclear plant does not comply with such rules, this is persuasive evidence that the crankshaft does not satisfy the stringent requirements embodied in the NRC regulations.

24. We recognize that the classification society rules generally relate to diesel engines in marine applications. However, they also are used for evaluating the adequacy of diesel engines in stationary applications. Tr. 23,979-80 (Christensen, Eley). We find that the differences between marine and stationary diesels are not significant in this context, particularly since there are no major design differences between diesel engines for marine or stationary use. Tr. 24,207-09, 24,211-12 (Christensen, Eley); 23,991 (Eley); 24,095-96 (Christensen).

25. A further issue in deciding the relevancy of the classification society rules is the suggestion that marine diesel generators are subjected to more severe operating conditions than land-based standby diesel generators and, thus, that the rules are not necessarily applicable to our consideration

of the Shoreham EDGs. We agree with the County, however, that differing operating conditions are not a basis upon which to reject the application of the classification society rules. Tr. 23,981-98 (Christensen, Eley).

26. Generally, main propulsion diesel engines on ships are subjected to more severe operating conditions than land-based diesel generators. However, unlike main propulsion engines, marine diesel generators are not connected to the ships' propellors, do not normally use low-quality fuel, are mounted on much stiffer foundations and generally have much shorter crankshaft lengths. Tr. 23,981-82 (Christensen). Because of these differences, marine diesel generators are not subjected to the more severe operating conditions of the main propulsion diesel engines. Indeed, marine diesel generators are not subjected to any significantly different operating conditions than land-based diesel generators. Tr. 23,981-98 (Christensen, Eley).

27. Although Lloyd's, the IACS and the ABS rules make some distinctions between main propulsion engines and engines used for electrical generation, the rules are virtually the same with respect to evaluating the adequacy of crankshafts. Tr. 23,987 (Christensen, Eley); 24,239 (Eley); County Ex. 38.

Accordingly, since marine diesel generator crankshafts are subject to the classification society rules and since marine diesel generators do not vary signficantly in design, stress, or operating conditions from land-based diesels, we are persuaded that it is relevant to assess whether the Shoreham replacement crankshafts comply with these rules. Similarly, we note that the DEMA recommendations relating to crankshafts impose the same limits on torsional vibration levels for marine and stationary applications. Tr. 22,705 (Chen); 22,709 (Pischinger); 24,212 (Christensen). We find, therefore, that the standards for evaluating the adequacy of the design of crankshafts in standby diesel generators in nuclear power plants should be at least as conservative as the standards for evaluating crankshafts used in marine diesel generators. Tr. 24,035 (Christensen). Indeed, it seems contrary to sound design principles to suggest that the margin of safety demanded of a diesel generator aboard a ship should be higher than that demanded for one serving the vital role of providing emergency power to a nuclear power plant.

2. <u>The Replacement Crankshafts do not Comply with the Classification Societies' Rules.</u>

28. As described in greater detail below, Lloyd's, the ABS and the IACS each evaluate the adequacy of crankshafts in different manners. Lloyd's calculates the maximum allowable horsepower that a crankshaft can safely withstand for reliable operation. The ABS calculates (i) the minimum dimensions of crankshaft webs to withstand bending stresses, (ii) the maximum permissible level of torsional (twisting) stresses imposed on a crankshaft, and (iii) when the calculated torsional stresses exceed the ABS limits, the ABS also calculates whether the crankshaft has a sufficient margin of strength over and above that which is needed to withstand the stresses to which the crankshaft is subjected (safety factor calculations). The IACS also performs safety factor calculations in evaluating the adequacy of crankshafts. The replacement crankshafts, however, are no better than marginal under any of these rules and fail to comply with the rules in a number of significant respects. The inadequacy of the replacement crankshafts under these rules is strong evidence that they are inadequate for use in the EDGs at Shoreham.

(a) The Replacement Crankshafts do not Comply with Lloyd's Rules.

29. The County's witnesses addressed in direct testimony whether the replacement crankshafts satisfy Lloyd's rules. Neither LILCO nor the Staff contested the accuracy or validity of the County's calculations under Lloyd's rules. The County's calculations show, and we find, that the replacement crankshafts do not comply with Lloyd's rules for maximum allowable horsepower at full load and overload.

30. Lloyd's rules are the most commonly used criteria for designing the initial dimensions of crankshafts. Tr. 24,001 (Eley, Christensen). Lloyd's evaluates the adequacy of the design of a crankshaft by calculating the maximum allowable horsepower that can be developed safely and reliably in an engine. Christensen and Eley, ff. Tr. 23,826, at 112.7/

7/ Lloyd's rule on allowable horsepower originated in the 1920's and has been continuously updated since that time, based on, among other things, the results of experimental work, including fatigue testing of full-scale and model crankshafts, and field failures. Tr. 24,203-04, 24,269-70 (Christensen). The Lloyd's calculation takes into consideration 26 inputs, including the manufacturing or forging process of the crankshaft, the strength of the crankshaft material, and the existence of fillet radii. Christensen and Eley, ff. Tr. 23,826, at 112.

31. Professor Christensen's calculations (County Ex. 36) under Lloyd's rules for maximum allowable horsepower show that the replacement crankshafts do not comply with Lloyd's rules at 1680 psi, the peak firing pressure assumed by FaAA in its studies at full load (3500 kW). At 1680 psi, the allowable horsepower permitted under Lloyd's rules is just under 4621 HP. At the actual measured peak firing pressure of 1720 psi at full load, the allowable horsepower under Lloyd's rules is 4496 HP. At 1800 psi, the peak measured firing pressure at overload (3900 kW), the allowable horsepower under Lloyd's rules is just under 4252 HP.[8/] Shoreham's horsepower rating of 4890 HP at full load and 5379 HP at overload substantially exceeds the allowables for horsepower under Lloyd's rules. Christensen, ff. Tr. 23,826, at 114; Tr. 24,273 (Christensen).

8/ Lloyd's rules and the other classification rules discussed herein all use the maximum firing pressure in the cylinders as an input. See, e.g., County Ex. 38, at 2 (IACS rules); LILCO Ex. C-41, at 2 (Lloyd's rules). The peak reported firing pressure in the EDGs is 1720 psi at full load and 1800 psi at overload. Christensen, et al., ff. Tr. 23,826, at 30-31; County Ex. 46, at 80, 95; LILCO Ex. P-9, at 6. These firing pressures are appropriately and conservatively used in the County's calculations under the classification societies' rules. In contrast, the lower cylinder pressure measurements taken from EDG 103 with piezoelectric transducers should not be used in these calculations. Those measurements do not purport to be maximum firing pressures but in fact are average pressures. Tr. 22,867, 22,869 (Johnston); LILCO Exs. P-5, P-35.

32. Mr. Eley also performed calculations under Lloyd's rules for maximum allowable horsepower for the replacement crankshafts. County Ex. 37. Those calculations confirm that the replacement crankshafts fail to comply with Lloyd's rules. Mr. Eley's calculations, which vary slightly from Professor Christensen's due to different computational methods, show that at 1680 psi, the allowable horsepower under Lloyd's rules is just under 4636 HP; at 1800 psi, the allowable horsepower is just under 4269 HP. Eley, ff. Tr. 23,826, at 115.9/

33. Thus, the County has demonstrated that the Shoreham EDGs are required to operate at a higher horsepower rating than would be considered acceptable under Lloyd's rules. The failure of the Shoreham EDGs to comply with the allowable horsepower limitations under Lloyd's rules is evidence that the EDGs cannot be operated safely and reliably at their rated power. Christensen and Eley, ff. Tr. 23,826, at 116.

9/ Technically, Lloyd's requires the EDGs to comply with its allowable horsepower rule at their overload condition. Lloyd's rules require that an engine be capable of operating at a 10% overload condition for 15 minutes. Tr. 24,006 (Eley). Because under the FSAR the EDGs are required to be capable of operating in a 10 percent overload condition for longer than the 15 minutes contemplated by Lloyd's (2 hours of every 24-hour period of continuous operation at full load), Lloyd's would require that the EDGs comply with its maximum horsepower rule at 3900 kW. Tr. 24,006 (Eley); 24,012 (Christensen).

(b) <u>The Replacement Crankshafts do not Comply with the IACS Rules.</u>

34. The County also asserts that the replacement crankshafts do not comply with the IACS rules as shown by calculations performed by TDI. LILCO disagrees, asserting that calculations performed by the ABS show compliance. For the reasons stated below, we agree with the County.

35. The IACS is an organization consisting of three minor and nine major classification societies, including Lloyd's and ABS. Christensen and Eley, ff. Tr. 23,826, at ' 6. The IACS has published draft rules (County Ex. 38) to evaluate the adequacy of crankshafts based on the assumption that the most highly stressed areas are the fillet transitions between the crank pin and the web as well as between the journal and the web.10/ Rather than calculating the adequacy of crankshaft

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10/ These rules are based upon a proposal by an international organization of engineers, CIMAC, entitled "Rules on Calculation of Crankshafts for Diesel Engines (4. Draft)" which is still under discussion among IACS members and between IACS and CIMAC. Portions of these rules are being used by the various classification societies. Christensen and Eley, ff. Tr. 23,826, at 116-17. The IACS rules are based on the conservative assumption that the maximum alternating bending stress and maximum alternating torsional stresses within a crankshaft occur simultaneously and at the same point (County Ex. 38, at 14), although generally these stresses do not occur simultaneously or at the same location in all diesel crankshafts. Tr. 24,109 (Christensen, Eley).

dimensions or torsional vibrations, the IACS rules calculate a factor of safety based upon torsional and bending stresses and stress concentration factors. Christensen and Eley, ff. Tr. 23,826, at 113. A crankshaft complies with the IACS rules where the ratio of its fatigue strength to its comparative alternating stress is greater than or equal to a factor of safety of 1.15. Id. at 117.

36. TDI performed calculations for the replacement crankshafts under the IACS rules. County Ex. 39. The County's review of these calculations shows that the replacement crankshafts do not comply with the IACS rules. Christensen and Eley, ff. Tr. 23,826, at 118. The calculated safety factor of the replacement crankshafts at full load is only 1.0422, which is less than the required 1.15. County Ex. 39, at 1, 6. Furthermore, those calculations were performed by TDI using 1650 psi as the maximum firing pressure. When the actual maximum firing pressure of 1720 psi in the Shoreham EDGs at full load is taken into consideration, the replacement crankshafts fail to comply with the IACS rules by an even greater margin. Christensen and Eley, ff. Tr. 23,826, at 118.

37. LILCO did not perform any IACS calculations to attempt to rebut the County's showing. LILCO, however, claims

that the ABS performed a calculation that allegedly shows that the replacement crankshafts comply with the IACS rules. In support of this assertion, LILCO cites only one page from an exhibit to the deposition of ABS employees. County Ex. 43 at 29; <u>see</u> LILCO Findings at 5.

38. We find that there is insufficient evidence to support LILCO's proposed finding. Neither LILCO nor the Staff offered testimony identifying or explaining this ABS calculation. The only evidence concerning the calculation is the oral testimony of the County's witnesses who had not previously reviewed the calculation. They testified that the calculation appeared to be one under the IACS rules, but that it also appeared to be based on a computer program that was not made available for the County to review. Tr. 24,136-37 (Christensen, Eley). Thus, there is no reliable evidence whether this calculation is under the IACS rules, what the inputs were, how it was calculated or what its significance is. Lacking such a foundation, we reject LILCO's proposed finding and conclude that the most reliable evidence indicates that the replacement crankshafts do not comply with the IACS rules.

(c) <u>The Replacement Crankshafts do not Comply with the ABS Rules on Torsional Vibration</u>.

39. Both the County and the Staff contend that the replacement crankshafts do not comply with the ABS rules on torsional vibration. We agree. Further, although LILCO argues that ABS has approved the torsional critical speed arrangement of the EDGs and that this is dispositive of the issue, we find that the ABS approval was based on inaccurate information submitted by TDI to the ABS. Thus, we give the ABS approval no weight. We explain the bases for our holding below.

(i) <u>The County and Staff Calculations Show that the Torsional Stresses in the Replacement Crankshafts Exceed ABS Limits</u>.

40. The County evaluated the adequacy of the design of the replacement crankshafts under Section 34.47 of the ABS rules concerning torsional vibratory stress. The County calculated the ABS maximum allowable stress level for the replacement crankshafts and compared that limit with FaAA's calculated stress level for the replacement crankshafts. The County's evaluation shows that the torsional vibratory stress imposed on the replacement crankshafts exceeds the maximum stress permissible under the ABS rules. Christensen and Eley, ff. Tr. 23,286, at 122-23; Tr. 24,170-71 (Eley); LILCO Ex. C-17 at 3-15; Tr. 22,888 (Johnston).11/

11/ In its prefiled testimony, the County compared the maximum allowable stress levels under the ABS rules to FaAA's

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41. The Staff performed a similar analysis and, like the County, found that the replacement crankshafts do not comply with the ABS rules concerning torsional vibratory stress. Thus, the Staff's calculations show that the stress levels in the replacement crankshafts (3608 psi for a single order and 7096 psi for total vibratory stress), exceed the ABS limits (3357 psi and 5035 psi, respectively, as calculated by TDI). Sarsten, ff. Tr. 23,126, at 15; Tr. 23,289-90 (Sarsten); Staff Ex. 4 at 4.12/

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October 31, 1983 analysis of the torsional vibratory stresses. That analysis, which utilized theoretical harmonic coefficients, or Tn values, calculated the maximum stresses to be 5,640 psi for the replacement crankshafts. Those calculated stresses exceeded the ABS allowable stress level by more than 10%. Christensen and Eley, ff. Tr. 23,826, at 123; Tr. 24,170 (Eley). When FaAA performed its analysis using Tn values derived from actual cylinder pressure measurements from EDG 103, FaAA calculated the maximum stresses to be 7006 psi for the replacement crankshafts. LILCO Ex. C-17 at 3-15; Tr. 22,888 (Johnston); 24,170-71 (Eley). Thus, using the updated Tn values, the total torsional vibratory stress imposed on the replacement crankshafts exceeds the maximum permissible stress under the ABS rules by approximately 40%.

12/ The evidence generally supports LILCO's proposed finding that the ABS summed only two orders when it performed its check calculations for torsional stress. LILCO Findings at 10, 18. We do not find, however, that the ABS only sums two orders of vibration when reviewing an engine's torsional critical speed arrangement. The ABS does not independently perform a summation of major orders of vibration but instead reviews calculations of torsional vibratory stress levels submitted to it by engine manufac-

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(ii) The ABS's Approval of the Torsional Critical Speed Arrangement of the EDGs is Entitled to No Weight.

42. In its direct testimony, LILCO did not contest the County or Staff testimony that the replacement crankshafts do not comply with ABS torsional vibration limits. Rather, LILCO's apparent position is that the ABS has approved the torsional critical speed arrangement of the EDGs (County Ex. 44), and that that approval is dispositive of the issue whether the replacement crankshafts comply with ABS rules. LILCO Findings at 5. We disagree. The ABS approval was obtained on the basis of inaccurate information submitted by TDI concerning the effect of shotpeening on the fatigue endurance limit of the replacement crankshafts. Thus, the ABS approval can be accorded no weight.

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turers. Tr. 23,286 (Sarsten). Indeed, when the ABS reviewed TDI's submission, it performed rough hand calculations based on the information submitted by TDI. Christensen and Eley, ff. Tr. 23,826, at 124; Tr. 24,172-74, 24,231 (Eley); 23,394 (Sarsten). It is highly unlikely that these hand calculations reflect a complete torsional analysis of the replacement crankshafts by ABS because such an analysis would require a computer. Tr. 23,394 (Sarsten); 24,281 (Eley). In any event, because that rough sum of only two orders exceeded the ABS limits, there was no need for the ABS to sum additional orders to determine whether the stresses exceeded ABS limits. Tr. 23,289-90 (Sarsten); Christensen and Eley, ff. Tr. 23,826, at 124-25.

43. TDI filed a submission with the ABS seeking its approval of the torsional critical speed arrangement of the EDGs. Christensen and Eley, ff. Tr. 23,826, at 123-25; County Ex. 45. In reviewing that submission, the ABS found that the calculated torsional vibratory stresses in the replacement crankshafts exceeded the ABS limits for torsional vibrations. Christensen and Eley, ff. Tr. 23,826, at 124.

44. However, like the other classification societies, ABS provides a mechanism whereby a diesel engine manufacturer who develops a design that does not comply strictly with ABS rules can seek approval of the design upon submission of appropriate stress analyses or other supporting data. Henriksen and Sarsten, ff. Tr. 23,126, at 9-10; Tr. 24,093 (Eley, Christensen). Accordingly, the ABS considered supplemental information submitted by TDI, including the alleged effect of shotpeening the crankshafts. TDI represented to ABS that a conservative minimal value of the increase in the fatigue endurance limit of the replacement crankshafts from shotpeening is 20%. Christensen and Eley, ff. Tr. 23,286, at 124, 127; County Ex. 45 at 24. The ABS did not question the 20% shotpeening value; rather, ABS accepted the TDI representation and performed six safety factor calculations based upon TDI's supplemental information. Based upon those calculations, ABS

gave its approval to the EDGs. Christensen and Eley, ff. Tr. 23,826, at 123-27.

45. The County asserts that TDI's representation concerning the effect of shotpeening on the fatigue endurance limit of the replacement crankshafts is inaccurate and that if no increase in the fatigue endurance limit is attributed to shotpeening, the replacement crankshafts do not meet the 1.34 safety factor which is the minimum ABS standard for assessing crankshafts that do not strictly comply with its rules for allowable torsional vibrations. Christensen, et al., ff. Tr. 23,286, at 125-29.13/

46. For several reasons, we agree with the County that the 20% shotpeening value cannot be accepted and that, accordingly, the ABS approval was obtained on the basis of inaccurate information. First, regarding the 20% value, we discuss in detail in Section II.D, infra, our finding that there is no reliable evidence to support a 20% increase in the fatigue endurance limit from shotpeening. Indeed, as discussed in that portion of this decision, it is clear that FaAA, LILCO's lead

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13/ The ABS's minimum 1.34 safety factor value is the lowest value for which it previously had approved another crankshaft under the same calculational methods. Tr. 24,282-83 (Eley); County Ex. 72, at 10.

consultant, specifically stated that it could not quantify the amount of increase, if any, in the fatigue endurance limit of the replacement crankshafts due to shotpeening.

47. Further, it is uncontested that, contrary to its representations to the ABS, TDI believed that shotpeening would <u>not</u> substantially improve the fatigue endurance limit of the replacement crankshafts. Christensen and Eley, ff. Tr. 23,286, at 125.[14] In fact, TDI had recommended against shotpeening the crankshafts based upon its experience and upon the opinion of its metallurgical consultant, that shotpeening would not provide more than a 5% increase in the fatigue endurance limit. <u>Id</u>. at 128; County Ex. 10, at 2-5. In addition, TDI had previously been informed by a manufacturer of crankshafts for TDI that shotpeening crankshafts of this size is a "waste of time." County Ex. 48. TDI never informed ABS about this information. Anderson, et al., ff. Tr. 23,286, at 128.

<u>14</u>/ The portion of TDI's submission to the ABS in which TDI represented that shotpeening increased the fatigue endurance limit of the replacement crankshafts by 20% is virtually identical to portion of the April 1984 FaAA crankshaft report that discussed shotpeening. County Ex. 45, at 28. As noted in Section II.D, <u>infra</u>, in its May 1984 crankshaft report, FaAA withdrew from its previous position that 20% was a conservative minimal value of the increase in the fatigue endurance limit due to shotpeening. LILCO Ex. C-17, at 3-11; Anderson, et al., ff. Tr. 23,286, at 128. Thus, the FaAA change in position effectively undermines the bases for TDI's submission to the ABS.

48. Second, we also agree with the County that if no increase in the fatigue endurance limit is attributed to shotpeening, the replacement crankshafts fail to satisfy the ABS's minimum desired safety factor value. In reviewing TDI's submission, the ABS performed six calculations of combined safety factors for the replacement crankshafts and compared those calculated values against its 1.34 safety factor value. Christensen and Eley, ff. Tr. 23,826, at 126; Tr. 24,824 (Eley). When the ABS attributed no increase in the fatigue endurance limit to shotpeening, four of the ABS safety factor calculations showed that the replacement crankshafts did not meet the ABS's safety factor value. Christensen and Eley, ff. Tr. 23,826, at 126-27.

49. The two other safety factor calculations performed by the ABS attributed the full 20% increase in the fatigue limit from shotpeening as represented by TDI in its submittal. Id. at 127; County Ex. 47, at 20. Those calculations showed that the replacement crankshafts exceeded the ABS's safety factor value. In making these two calculations, however, the ABS did not verify whether shotpeening had in fact increased the fatigue endurance limit of the crankshafts by 20%. Christensen and Eley, ff. Tr. 23,826 at 127; County Ex. 43, at 4, 11. Indeed, only when it is assumed that shotpeening does in fact

produce a quantifiable increase in the fatigue endurance limit of greater than 10% do the replacement crankshafts meet the ABS's safety factor value. Christensen and Eley, ff. Tr. 23,826, at 127, 129; County Ex. 47, at 20. If a 5% increase is assumed, the replacement crankshafts would not meet the ABS's safety factor value under one calculation and would only marginally meet that value under the other calculation. Christensen and Eley, ff. Tr. 23,826, at 129; County Ex. 47, at 20.15/

50. Based on the foregoing, we conclude that if ABS had attributed no increase in the fatigue endurance limit to shotpeening, then the replacement crankshafts would not meet the ABS torsional stress standards. Since there is no reliable basis in the record to attribute any such increase due to shotpeening, we must accord no weight to the ABS approval.

15/ The ABS also performed its safety factor calculations using 1700 psi as the value given to it by TDI for the maximum firing pressure in the EDGs. Christensen and Eley, ff. Tr. 23,826, at 124-25. As shown above, the appropriate value should be 1720 psi. *Id*. at 130.

(d) <u>The Replacement Crankshafts do not Comply with the ABS Rules on Crankshaft Web Dimensions</u>.

51. The County and the Staff disagree whether the replacement crankshafts comply with ABS rules on crankshaft web dimensions. LILCO presented no direct testimony on the issue. For the reasons described below, we find that the webs on the replacement crankshafts do not comply with the ABS rules.

52. In order to provide for adequate bending stiffness in crankshafts with solid webs, Section 34.17.4 of the ABS rules provides that:

> The proportions of the crankshaft webs are to be such that the effective resisting moment of the web in bending is not less than 60% of the resisting moment of the minimum required diameter of pins and journals in bending.

Professor Christensen's calculations show that the web strength in bending is equivalent to a crank pin or journal diameter of 10.9337 inches. Using this value, Professor Christensen calculated the maximum allowable firing pressure for the replacement crankshafts. Those calculations show that the maximum allowable firing pressure for the EDGs under the ABS rules is 1746 psi at full load and 1651 psi at overload. Thus, when the actual measured peak firing pressures of the EDGs are considered

(1720 psi at 3500 kW and 1800 psi at 3900 kW), the replacement crankshafts do not comply with the ABS rules for operation at overload and are marginal at full load. Christensen, ff. Tr. 23,826, at 118-20; County Ex. 40.

53. We find that Professor Christensen's calculational method conforms with the method intended to be used by the ABS for calculating the dimension of crankshaft webs. In performing his calculation, Professor Christensen relied on the interpretation of the ABS web rules given by one of the ABS deponents. Tr. 24,145, 24,147 (Christensen). As the ABS deponent explained:

> I believe that our normal practice would be to measure that dimension from the boundary of the actual crankshaft material at one fillet to that at its opposite fillet, rather than constructing the arbitrary lines of a face of the web and going between them. Essentially, it makes sense to count only the metal that is actually there.

County Ex. 72. LILCO Ex. C-42; Staff Ex. 1. We find that Professor Christensen calculated a section of the web measuring the web dimension from metal to metal according to the ABS deponent's interpretation of the ABS rule. Tr. 24,147, 24,148 (Christensen).

54. In contrast, Professor Sarsten's web calculation, which purports to be based upon the same interpretation of the ABS rules relied upon by Professor Christensen (Tr. 23,492 (Sarsten)), does not actually represent the metal to metal boundaries through the full width of the section of the web. Rather, Professor Sarsten focused on a section of the web in the vertical plane. Tr. 24,153 (Christensen).16/ In order to obtain the "effective resisting moment of the web in bending" within the meaning of the ABS rule, one must consider a section of the web in the horizontal plane as Professor Christensen did. Tr. 24,158 (Christensen). Such an approach complies with the ABS method by considering only the metal that actually exists in the web.

55. LILCO asserts that the ABS has approved the dimensions of the web on the replacement crankshafts. LILCO Findings at 17. There is insufficient evidence to support such a finding. LILCO did not introduce into evidence any official ABS certification nor any testimony identifying or explaining

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16/ Professor Sarsten had no previous familiarity with how the ABS interpreted its rules. Tr. 23,492 (Sarsten). Professor Christensen's previous understanding of how the ABS interpreted its rule relating to webs was confirmed by the interpretation given by the ABS deponent. Tr. 24,145, 24,148 (Christensen).

how the ABS purportedly calculated the web dimensions of the replacement crankshafts. The County's witnesses testified only that the ABS deponents assumed that the webs had been approved. None of the ABS deponents, however, performed any web calculations or produced any such calculations which explain the basis for their assumption. Tr. 24,141, 24,162 (Christensen, Eley).

(e) <u>The Replacement Crankshafts are Inadequately Designed for Operating at Overload and their Design is Marginal for Operating at Full Load even under the German Design Criteria used by F.E.V.</u>

56. The County also alleged that the replacement crankshafts are inadequately designed for operating at 3900 kW and that their design is marginal for operating at 3500 kW under the German design criteria used by LILCO's consultant, F.E.V. Christensen and Eley, ff. Tr. 23,?26, at 121. LILCO claims that calculations under these criteria show the replacement crankshafts have unlimited life at 3500 kW and will operate for 1200 hours at 3900 kW. Pischinger, ff. Tr. 22,610, at 5. For the reasons discussed below, we find that LILCO's calculations under these criteria do not provide assurance that the replacement crankshafts are adequately designed for operating at full load and overload. County Ex. 41, at 5; Christensen and Eley, ff. Tr. 23,286, at 121.

57. In evaluating the replacement crankshafts, Dr. Pischinger of F.E.V. performed calculations under the Kritzer-Stahl design criteria. Kritzer-Stahl is a method for calculating stresses in a crankshaft and for comparing those stresses with calculated fatigue endurance limits for the crankshaft material. The ratio of the calculated stresses to the endurance limit gives a factor of safety for the crankshaft. Tr. 22,767 (Pischinger).

58. The replacement crankshafts are "just on the boundary" of compliance with the Kritzer-Stahl criteria at full load. County Ex. 41, at 4. As Dr. Pischinger explained, the calculated stresses (172 Newtons per square millimeter) at full load are just below the calculated endurance limit of the replacement crankshaft material (175 Newtons per square millimeter). Tr. 22,794 (Pischinger). Using these calculated values, Dr. Pischinger obtained a safety factor of slightly less than 1.02 for the replacement crankshafts. Tr. 23,004 (Pischinger). Dr. Pischinger described this safety factor value as "nominal." Tr. 23,004. At overload, the replacement crankshafts do not comply with the Kritzer-Stahl criteria, because the calculated stresses exceed the calculated endurance limit. Tr. 22,792-93 (Pischinger); County Ex. 41 at 5. Thus, the ratio of the stress to the endurance limit gives a factor of safety less than 1.0.17/

17/ The safety factor values for the replacement crankshafts under the Kritzer-Stahl design criteria should actually be

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59. The Kritzer-Stahl design criteria contain no discrete recommended factor of safety. However, a factor of safety of 1.15 is the lowest acceptable value in contemporary industrial practice with which Dr. Pischinger is familiar for the design of a crankshaft in a medium speed diesel engine the size of the EDGs. Tr. 23,012, 23,071-72 (Pischinger). Thus, we find that the calculated safety factors for the replacement crankshafts under the Kritzer-Stahl criteria do not provide assurance that the replacement crankshafts are adequate for their intended service.

60. Dr. Pischinger also calculated additional safety factor values for the replacement crankshafts at full load and overload based upon a comparison of his Kritzer-Stahl

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somewhat lower. A calculation of the endurance limit under Kritzer-Stahl takes into consideration a number of factors, including the ultimate tensile strength (U.T.S.) of the crankshaft material. Tr. 22,791 (Pischinger). Dr. Pischinger used a value for the U.T.S. of the replacement crankshafts that is higher than the minimum measured U.T.S. of two of the replacement crankshafts. Tr. 22,992-93 (Pischinger, Montgomery); LILCO Ex. C-12. This is particularly unsettling because a minor refinement in the inputs to Dr. Pischinger's calculation can substantially affect the results. For example, a very small change in the stress level data used in Dr. Pischinger's calculations resulted in a 100% change in the predicted lifetime of the crankshafts at overload. Tr. 23,043-44 (Pischinger).

calculations on the original and replacement crankshafts. Tr. 23,004 (Pischinger). For the reasons stated below, we find that these additional safety factor calculations do not provide assurance of the adequacy of the replacement crankshafts.

61. Dr. Pischinger's calculations predict that the original crankshafts should have failed after two million cycles (approximately 150 hours) but in fact they failed at about four million cycles (273 hours). Using an S-N curve[18/] and the actual number of cycles to failure rather than the predicted number of cycles, Dr. Pischinger calculated that the ratio of maximum stress to endurance limit for the original crankshafts should have been 22.7% lower. Tr. 23,005 (Pischinger). By adding the safety factor for the replacement crankshafts to this value, Dr. Pischinger obtained an additional safety factor of 24 percent for the replacement crankshafts at full load and 15 percent for operation at 3900 kW. Tr. 23,004, 23,037 (Pischinger).

62. We do not rely on these additional safety factor calculations. The S-N curve used by Dr. Pischinger was based on

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18/ S/N curves show the relationship between the stress for failure and the number of cycles where failure will occur at a particular stress level. Tr. 22,778 (Pischinger, McCarthy).

the number of cycles to the actual severing of one test crankshaft. Tr. 23,008, 23,778 (Pischinger).19/ Similarly, the four million cycle figure used in his calculations is the number of cycles at which the original crankshaft actually severed. Tr. 23,008 (Pischinger). As Dr. Pischinger testified, the four million cycle figure is very important to his conclusion. Tr. 23,007 (Pischinger). If "failure" is defined as the time when the crack actually initiated in the crankshaft, instead of at the time of severance, failure occurred at substantially less than four million cycles. Tr. 23,008 (Pischinger). Although it is not clear precisely when detectable cracks initiated in the original crankshafts, the lifetime of the original EDG 102 crankshaft from the time a detectable indication was present to actual severance was a period of less than 168 hours (approximately two million cycles). Tr. 23,064 (McCarthy). If this approximate value is used, Dr. Pischinger's Kritzer-Stahl calculations correctly predicted that the original crankshafts would fail after about 150 hours.

19/ This S-N curve was based on only eight measurements taken on one test crankshaft. That crankshaft was smaller (10-inch journal diameter) than even the original crankshafts on the EDGs and had a lower ultimate tensile strength than that of the replacement crankshafts. Dr. Pischinger did not know the testing procedures used or the forging method of the test crankshaft. Tr. 22,781, 22,827-29 (Pischinger).

Viewed in this light, the additional safety factors for operation of the replacement crankshafts at full load and overload should be approximately the same as originally calculated by Dr. Pischinger. As previously indicated, those calculations do not provide assurance of the adequacy of the replacement crankshafts.

63. Even if these additional calculations are a normally acceptable method of calculating safety factors under the Kritzer-Stahl criteria -- and there is no evidence in the record that they are -- they hardly provide the degree of conservatism we deem appropriate in evaluating the adequacy of the replacement crankshafts. These additional safety factor calculations merely fall within the low to middle range of acceptable safety factor values in contemporary European industrial practice. Tr. 23,012, 23,071-72 (Pischinger). That, and the fact that the calculations are based upon U.T.S. values that are too high (Tr. 22,992-93 (Montgomery)), do not give us sufficient confidence in these calculations to find that the replacement crankshafts are adequate for use at Shoreham.[20]

20/ The County also asserted that the dimensions of the webs on the replacement crankshafts are inadequate under the design criteria used by F.E.V. Christensen and Eley, ff. Tr. 23,826, at 121; County Ex. 41, at 4, 7. Although the Kritzer-Stahl criteria do not provide any specific guidance on the dimensions of the webs, they do consider the

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B. <u>LILCO Cannot Rely on Alleged Compliance with DEMA to Prove the Adequacy of the Replacement Crankshafts</u>.

64. LILCO has urged the Board to find that the DEMA recommendations are appropriate standards by which to judge the adequacy of the replacement crankshafts and that the replacement crankshafts in fact comply with the DEMA recommendations. Johnston, et al. ff. Tr. 22,610, at 20-30. The County asserts that the adequacy of the replacement crankshafts should be assessed against the rules of the classification societies and that the DEMA recommendations are not reasonable alternative standards. Anderson, et al., ff. Tr. 23,826, at 109-11, 114, 142. The Staff contends that, in any event, the replacement crankshafts do not comply with DEMA. Sarsten, ff. Tr. 23,126, at 13, 17. We agree with the County and the Staff, finding that LILCO has failed to establish that the DEMA recommendations are an appropriate standard or that the replacement

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thickness of the web as an input. Tr. 22,768, 22,783-84 (Pischinger). Dr. Pischinger's own engineering judgment called for the use of a web approximately 1/2-inch thicker. Such a modification also would have required a different sized crankshaft bearing. Tr. 22,783-84, 22,787-88, 23,024 (Pischinger). A thicker web would have beneficially reduced the stress concentration values in the replacement crankshafts and increased their fatigue endurance limit. Tr. 22,784-86, 23,025 (Pischinger).

crankshafts satisfy the DEMA recommendations. Our reasons are set forth below.

1. The DEMA Recommendations are not a Reasonable Standard by which to Judge the Adequacy of the Replacement Crankshafts.

65. DEMA is an American trade association of diesel engine manufacturers. Henriksen and Sarsten, ff. Tr. 23,126, at 10. Unlike the classification societies, DEMA neither approves nor disapproves crankshafts. Tr. 22,688 (Chen). DEMA does not have an in-house staff of engineers to review diesel generator plans, crankshaft drawings or torsional vibration calculations, nor does DEMA have surveyors to inspect crankshafts at specific installations. Tr. 22,687-68, 23,055-57 (Chen); 24,194-95 (Eley). In its publication, Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines, DEMA describes various aspects of the design, operation and testing of diesel engines. For crankshafts, however, DEMA provides only self-policing guidelines for allowable stresses associated with torsional vibratory conditions. Unlike the classification societies, DEMA does not provide any guidance for crankshaft dimensions, material properties, or methods of fabrication. Henriksen and Sarsten, ff. Tr. 23,126, at 10; Tr. 22,688 (Chen).

66. The DEMA recommendations on allowable torsional vibration levels have not been revised in at least 25 years. Tr. 22,692 (Chen).21/ Even Dr. Chen, who chaired one of the DEMA technical committees (Tr. 22,704 (Chen); Chen. ff. Tr. 22,610, at 30), does not consider the DEMA recommendations to be up-to-date. Tr. 22,690 (Chen). In fact, Dr. Chen could not say whether DEMA itself considers the DEMA recommendations to be up-to-date. Tr. 22,689-90 (Chen).

67. Although DEMA allegedly does "provide standards to measure the adequacy of a crankshaft" (Chen, ff. Tr. 22,610, at 14), we cannot find the DEMA recommendations are reasonable standards by which the adequacy of the design of the replacement crankshafts can be measured. Christensen and Eley, ff. Tr. 23,826, at 111. The relevant DEMA recommendations only concern torsional vibration and thus they lack sufficient scope

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21/ In its proposed findings, LILCO states that the trend among classification societies has been to become less conservative in their rules for allowable stress, whereas DEMA has not revised its limits for allowable torsional stresses since the late 1950's. LILCO asserts that this indicates that DEMA has a large built-in margin of safety. LILCO's Findings at 9. The record does not support LILCO's assertion. LILCO cites no testimony or exhibits to support its assertion. In fact, contrary to LILCO's assertion, the evidence indicates that the DEMA limits remain quite high (<u>i.e.</u>, less conservative) in relation to the torsional vibratory stress limits of the classification societies. Tr. 23,364 (Sarsten).

and breadth to be used to evaluate the overall adequacy of the design of a crankshaft. Indeed, LILCO's own expert agrees with the County (id.) that the DEMA recommendations are not a design code, "are not explicit enough to be used as a crankshaft criteria," and cannot be used to design a crankshaft. Tr. 22,689-90, 23,015 (Chen).22/

68. The DEMA recommendations also lack sufficient specificity to be used as reliable standards for evaluating the adequacy of the design of the replacement crankshafts. The DEMA recommendations for allowable torsional vibratory stresses in crankshafts provide that:

> In the case of constant speed units, such as generator sets, the objective is to insure that no harmful torsional vibratory stresses occur within five percent above and below rated speed.
>
> For crankshafts . . . made of conventional materials, torsional vibratory conditions

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22/ This fact also is confirmed by the DEMA recommendations. The forward to the DEMA recommendations explicitly states: "[I]t is not the purpose of this book to attempt to set forth basic design criteria for engines because such approach would be impossible within this volume and yet do justice to the many types of engines on the market, notwithstanding the fact that many technical texts are available to the student who may be undertaking the design criteria aspects of engines in general." Standard Practices for Low and Medium Speed Diesel and Gas Engines, 6th ed., 1972 at iii; Christensen and Eley, ff. Tr. 23,826, at 111.

shall generally be considered safe when they induce a superimposed stress of less than 5000 psi, created by a single order of vibration, or a superimposed stress of less than 7000 psi, created by the summation of the major orders of vibration which might come into phase periodically.

As is readily apparent, the DEMA recommendations do not specify any method to be used for calculating torsional vibratory stress when performing a calculation for comparison with the DEMA limits. Tr. 23,238 (Sarsten). In addition, the DEMA recommendations do not specify the number of major orders of vibration that are to be summed when calculating stresses for comparison with the 7000 psi DEMA limit. Tr. 23,249, 23,297 (Sarsten); Tr. 22,741-42, 22,745 (Chen). Indeed, LILCO's own expert witness testified that he hoped that DEMA would revise its rules because they are not explicit. Tr. 22,701 (Chen).

69. Further, LILCO did not produce any DEMA representative to testify as to how the DEMA recommendations are interpreted by DEMA. There is no evidence in the record that LILCO sought and obtained an interpretation of the recommendations from DEMA itself. Indeed, apparently there is no formal procedure to obtain a DEMA interpretation of the DEMA recommendations. Tr. 22,703-04 (Chen). When Dr. Chen contacted members of DEMA's technical committee concerning the

recommendations, they would not respond to his questions, and were very defensive. Tr. 22,692-93, 22,701-02 (Chen). When Dr. Chen contacted several DEMA member firms, he found that there were various discrepancies in the methods used and the interpretations of the DEMA recommendations. Tr. 22,691 (Chen). This evidence of varying interpretations makes it impossible for this Board to make any reliable findings as to how the DEMA recommendations should be interpreted.

70. Indeed, there is considerable conflicting testimony in the record over how DEMA should be interpreted, thus underscoring our view that we cannot find the DEMA recommendations to be adequate criteria for judging the adequacy of the replacement crankshafts. LILCO claims that the DEMA recommendations should be interpreted in light of the conventional analytical techniques that were used for calculating torsional vibratory stresses when the present DEMA limits were established in 1959 or when the DEMA recommendations were last revised in 1972. Tr. 22,710-12 (Chen). According to LILCO, the conventional analytical technique used at those times was the Holzer method, not calculational methods such as modal superposition. Tr. 22,710-11 (Chen); 22,726 (Johnston). The calculated stresses from the Holzer method fall below the DEMA recommended limits.23/

23/ According to FaAA, TDI's calculations under the Holzer method show that the largest single order stress at full

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71. The Staff disagrees that the DEMA recommendations contemplate only the use of the Holzer method for calculating the torsional vibratory stress levels in crankshafts. Tr. 23,284-85 (Sarsten). Calculational methods such as modal superposition have been available since the mid-1960's and have been conventionally used since approximately 1972. Tr. 23,283-84 (Sarsten); 22,720 (Chen); 22,990 (Pischinger). In fact, even Dr. Chen testified that forced vibration calculations such as TORVAP and similar modal superposition analyses, are typically and conventionally performed by the diesel engine industry to check the adequacy of a crankshaft to withstand torsional stress. Chen, ff. Tr. 22,610 at 28; Tr. 22,720 (Chen).24/

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load and rated speed is 2980 psi. Johnston, ff. Tr. 22,610, at 24; LILCO Ex. C-17 at 2-3 and Table 2.4. In addition, FaAA's calculations using Stone & Webster's torsiograph test data show that the largest single order stress at full load and rated speed is 3108 psi and a total stress of 6626 psi. Johnston, ff., Tr. 22,610, at 24, 26; LILCO Ex. C-17 at 2-4 and Table 2.5. These calculations also show that the largest stresses at 3800 kW are 3242 psi for a single order and 6875 for combined response. By linear extrapolation, FaAA calculated that the corresponding stresses at 3900 kW are 3287 psi and 6958 psi. Johnston, ff. Tr. 22,610, at 26; LILCO Ex. C-17 at 2-4.

24/ FaAA used modal superposition analysis to determine whether the replacement crankshafts complied with the DEMA limits for off-speed conditions. Modal superposition was

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72. We agree with the Staff. LILCO's argument presumes that the calculational method currently intended by DEMA to be used in assessing a crankshaft against the DEMA limits is the same method that was used over 25 years ago, despite the fact that much more accurate methods have been available for almost 20 years and have been considered conventional for approximately 10 years. Tr. 23,283-84 (Sarsten); 22,720 (Chen); 22,990 (Pischinger). We consider it highly unlikely that DEMA would continue to publish and issue its recommendations if it believed that those recommendations were not applicable to the conventional analytical techniques currently used in the diesel engine industry.

73. LILCO also claims that even if calculational methods such as modal superposition are appropriate, only four or six major orders of vibration should be summed when calculating torsional vibratory stresses for comparison with the DEMA recommended limits. LILCO urges that whether four or six orders

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used to predict the free-end response that would have been measured by a torsiograph test had it been possible to run the EDGs at those off-speed conditions under load. The calculated value of free-end amplitude was then reduced to a nominal stress using the standard torsiograph method. Tr. 22,724 (Johnston).

should be summed depends upon engineering judgment. Tr. 22,858, 23,017, 23,019 (Chen). According to LILCO, when the DEMA limits were established, it was not practical to sum many orders of vibration with any degree of accuracy. Tr. 23,018 (Chen). Thus, even though the calculations by Professor Sarsten, FaAA and Dr. Pischinger show that the torsional vibratory stresses in the replacement crankshafts exceed the DEMA limits (<u>see</u> Section II.B.2, <u>infra</u>), LILCO argues that those calculations are not appropriate for purposes of determining compliance with the DEMA limits because they sum twenty-four orders.25/

74. The Staff disagrees. The Staff interprets the DEMA recommendations as requiring the summation of twenty-four orders of vibration. We agree with the Staff. Major orders

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25/ LILCO also claims that the DEMA recommendations should be interpreted in light of the crankshaft materials conventionally used in 1959 and 1972. LILCO interprets the term "conventional materials" in the DEMA recommendations as referring to materials with ultimate tensile strengths in the range of 60,000 psi to 70,000 psi. Tr. 22,711 (Chen).

We note that the DEMA recommendations on crankshafts do not define the term "conventional materials" or distinguish between grades of conventional materials, (Tr. 23,351-52 (Sarsten)), nor do they permit higher stress levels for crankshafts that have higher U.T.S. values. We find that there is insufficient evidence to reach any conclusions on LILCO's assertion.

within the meaning of DEMA are those that contribute to the accuracy of the calculation. Twenty-four orders include all of the orders of vibration that are significant to the accuracy of the result. Tr. 23,297, 23,299 (Sarsten). Summing twenty-four orders has been the standard practice in the European diesel engine industry for calculating torsional vibratory stresses in four stroke engines since the introduction of powerful digital computers in the 1960's. Tr. 22,989-90 (Pischinger); 23,250, 23,283 (Sarsten).26/ In fact, neither Dr. Pischinger nor Professor Sarsten would sum fewer than twenty-four orders when calculating torsional vibratory stresses. Tr. 22,798 (Pischinger); 23,285 (Sarsten). FaAA also performed its calculations using the sum of twenty-four orders. Tr. 22,724-27, 22,734 (Johnston). The stresses resulting from the sum of additional orders beyond twenty-four are not significant. Tr. 23,253 (Sarsten).27/

26/ By the early 1970's, the universal practice among engine manufacturers submitting computer calculations to one classification society, Det Norske Veritas, was to include forced vibration calculations summing twenty-four orders. Tr. 23,283-84 (Sarsten). Prior to the advent of the digital computer, it was customary to consider only one order because vectorial summation is a very laborious task to perform by hand. Tr. 23,282, 23,284 (Sarsten).

27/ In making these findings, we give little weight to Dr. Chen's interpretation of the DEMA recommendations. First, Dr. Chen possesses no particular expertise in interpreting the DEMA recommendations. He was not a member of the DEMA

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75. LILCO's suggested interpretation of DEMA would permit the summation of fewer than six major orders, depending on the user's engineering judgment. Even Dr. Chen admitted that experts could reasonably disagree over which orders were major. Tr. 22,728 (Chen). The danger in summing fewer major orders than the standard practice of summing twenty-four orders,

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technical committee when the torsional vibratory limits were established (Tr. 22,704 (Chen)), and there is no evidence in the record that Dr. Chen had previously performed a calculation under DEMA. This lack of any special insight is evidenced by his contacting DEMA and its members to obtain an interpretation of the DEMA recommendations. Tr. 22,691-93 (Chen).

In addition, Dr. Chen's testimony on the DEMA recommendations is far from clear and often contradictory. For example, Dr. Chen testified on redirect examination that DEMA is a reliable standard, but previously had testified on cross-examination that DEMA was not explicit enough to be used as a crankshaft criteria. Tr. 22,690, 23,014-16 (Chen). In addition, Dr. Chen testified on re-direct examination that the customary practice in calculating stresses under DEMA is to sum four or six orders, but testified on cross-examination that the DEMA members use various discrepant interpretations and methodologies when making DEMA calculations. Tr. 22,691, 23,019 (Chen). Furthermore, as support for his assertion that the DEMA recommendations are adequate, Dr. Chen testified that he had never seen a crankshaft that complied with DEMA fail primarily from torsional fatigue. On cross-examination, however, Dr. Chen admitted that he has investigated the failure of only three crankshafts (Tr. 23,074 (Chen)), and that the vast majority of crankshafts that fail do not fail primarily from torsional fatigue but from a combination of stresses. Tr. 22,865 (Chen).

however, is that the user could choose whatever number of orders that permits him to comply with allowable stress levels. Tr. 23,297-98, 23,301 (Sarsten). Such an approach would be inconsistent with the notion of a published standard practice. Tr. 23,298 (Sarsten).

76. Based on all of the foregoing, we find that LILCO has failed to establish that the DEMA recommendations are sufficiently comprehensive and well understood by the parties to constitute an adequate criterion for judging the adequacy of the replacement crankshafts.28/

2. The Replacement Crankshafts do not Comply with the DEMA Recommended Limits on Torsional Vibration.

77. Even assuming *arguendo* that the DEMA recommendations constitute a reasonable alternative standard for assessing the adequacy of the design of the replacement crankshafts, we find that the replacement crankshafts do not comply with the DEMA

28/ In the following discussion, we assume *arguendo* that the DEMA rules are sufficiently well defined and understood and then assess whether Shoreham's replacement crankshafts comply with those rules. In that discussion we rely upon the Staff's interpretation of the DEMA rules, including the Staff view that DEMA requires the summation of twenty-four orders of vibration. *See* preceding discussion in Section II.B.1 for our reasons for adopting the Staff interpretation of DEMA over the contrary views of LILCO witness Dr. Chen.

recommended limits on torsional vibration. The Staff and LILCO performed analyses of the torsional vibratory stresses for the sum of twenty-four orders of vibration for EDG operation at 3500 kW. All of these analyses show that the replacement crankshafts do not comply with DEMA. The most accurate of these analyses is the Staff's.

78. The Staff's analysis by Professor Sarsten shows that for section No. 6 of the crankshaft (*i.e.*, the torsional spring constant representing the crankshaft elasticity between cylinders 5 and 6), the torsional stresses for the sum of twenty-four orders exceed the DEMA limit of 7000 psi over virtually the entire speed range called for by DEMA (*i.e.*, from 5% below rated speed to 5% above rated speed of 450 rpm). Sarsten, ff. Tr. 23,126, at 13; Tr. 23,307-08 (Sarsten). At rated speed, Professor Sarsten's calculated stresses equal 7096 psi; at 5% below rated speed, the stresses equal 7051 psi; at 5% above rated speed, the stresses equal 7851 psi. Tr. 23,380, 23,540 (Sarsten).[29/] Only at approximately 440 rpm does section

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29/ All of the values calculated by Professor Sarsten have been adjusted to account for appropriate damping values and to agree with the measured value of free-end amplitude. Tr. 23,307-08, 23,380 (Sarsten). As Professor Sarsten testified, it is often customary to adjust calculated stress values to account for the difference between the calculated and measured values of free-end amplitude. This procedure provides a more accurate calculation of stresses. Tr. 23,344 (Sarsten).

number 6 of the crankshaft comply with the DEMA limits at full rated load. Tr. 23,382, 23,540 (Sarsten).

79. LILCO's analyses by FaAA and Dr. Pischinger also show noncompliance with DEMA. FaAA's analysis shows that for section number 6 of the crankshaft, the nominal torsional vibratory stresses for the sum of twenty-four orders at rated speed equal 7006 psi (LILCO Ex. C-17 at 3-15; Tr. 22,735, 22,888 (Johnston)), which exceeds the DEMA limit of 7000 psi. FaAA's analysis also shows that the replacement crankshafts do not comply with DEMA from 5% below to 5% above rated speed. Although the FaAA crankshaft report states that the calculated maximum torsional stresses at 428 rpm (5% below rated speed) and 473 rpm (5% above rated speed) equal the DEMA limit of 7000 psi within plus or minus 3% (LILCO Ex. C-17 at 2-5), some of the stresses between those speeds exceed the DEMA limit of 7000 psi. Tr. 22,835 (Johnston). Dr. Pischinger's calculations show that at 5% above rated speed, the stresses equaled 7470 psi; at rated speed, the stresses equaled 6890 psi; and at 5% below rated speed, the stresses equaled 6240 psi. Tr. 22,800-01 (Pischinger).

80. Thus, based upon both the Staff and LILCO computations, there can be no finding of compliance with DEMA. We

find in this regard that Professor Sarsten's method of calculating the nominal torsional vibratory stresses is the most accurate of the methods used by the expert witnesses in this case.30/ Indeed, although many of the inputs to the calculations performed by the experts were the same, Professor Sarsten's method calculated a value of free-end amplitude (.690) which was in closer agreement with the actual measured value of free-end amplitude by Stone & Webster (.693) than the values calculated by FaAA (.662), Dr. Chen (.59), and Dr. Pischinger (.665). Tr. 23,443-44 (Sarsten); 22,816 (Pischinger); 22,858 (Chen).31/ This fact in itself is strong evidence that Professor Sarsten's method is more accurate than the others. Further, Professor Sarsten's COMHOL method calculates the steady-state forced vibration of damped linear systems subject to periodic forcing functions represented by a

30/ In giving more weight to the results of Professor Sarsten's calculations, we note that Professor Sarsten participated in the historical development of the methodology used for calculating torsional vibrations. Tr. 23,239 (Sarsten). Professor Sarsten has performed torsional vibration calculations of crankshafts in four stroke diesel engines since 1957, and since 1962 has developed numerous programs for calculating torsional vibratory stress. Tr. 23,260, 23,262 (Sarsten).

31/ Calculations of torsional vibratory stress will be roughly proportional to the calculated value of free-end amplitude. Tr. 23,443-44 (Sarsten).

Fourier series of harmonics. Sarsten, ff. Tr. 23,126, at 14. Where, as here, damping is present in the system, COMHOL more accurately takes into consideration the effects of damping than the modal superposition method and calculates the true vibrations present in the system. Tr. 23.435 (Sarsten).32/

81. In contrast, FaAA used the modal superposition method to calculate the torsional vibratory stresses in the replacement crankshaft.33/ Theoretically, however, that method is not applicable when, as here, damping is present in the system. Tr. 23,430, 23,436 (Sarsten). In practice, modal superposition is an acceptable method of calculation if low damping values are utilized. Tr. 23,430 (Sarsten). FaAA, however, used a relatively large 2.5% damping value, which roughly translates

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32/ COMHOL and modal superposition are two different methods of simultaneously solving complex equations. COMHOL represents the more complex method of calculation. Although the COMHOL and modal superposition methods begin with essentially the same equations, COMHOL solves the equations in the complex plane. Modal superposition, on the other hand, reduces the equations to very simple ones for each mode, solves them for each mode and each order separately, and then sums all of the orders and all of the modes. Tr. 23,435 (Sarsten); 23,050-52 (Johnston).

33/ Although it is not clear from the record, Dr. Pischinger's method appears to be essentially the same as FaAA's, and their calculated free-end amplitudes are in close agreement. The only difference in their methods apparently is the method of reducing the measured cylinder pressure data to Tn values. Tr. 22,814 (Johnston).

to a dynamic magnifier of 20. Tr. 23,434 (Sarsten). The normally accepted damping value for generator engines is a dynamic magnifier of 40 to 45. Tr. 23,438 (Sarsten). Professor Sarsten used a dynamic magnifier of 40 in his calculations which is more consistent with standard industry practice.34/ FaAA's use of distributed damping and a larger damping value (i.e., a lower dynamic magnifier) results in lower, and less accurate, values of the torsional vibratory stresses. Tr. 23,434, 23,439 (Sarsten).35/

82. LILCO urges the Board to adopt Dr. Chen's torsional analysis of the replacement crankshafts which calculated

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34/ Professor Sarsten selected a dynamic magnifier on the low end of the normally accepted range of values in order to obtain a safe lower bound on the torsional vibratory stresses on the replacement crankshafts. Tr. 23,438 (Sarsten). The use of a lower dynamic magnifier slightly underestimates the torsional stresses. Tr. 23,543 (Sarsten). Professor Sarsten did indicate that other analysts often use higher dynamic magnifier values and that one engine firm that deals almost exclusively with generators uses values as high as 90 to achieve good correlation between calculated stress values and measured values. Tr. 23,437 (Sarsten).

35/ FaAA's method also assumes a one node vibratory form as the basis for calculating stresses. Tr. 23,435, 23,442 (Sarsten). This assumption produces only a near approximation of the stresses and thus is slightly inaccurate. Tr. 23,435-36 (Sarsten). In contrast, Professor Sarsten's method is more accurate because it takes into consideration all of the different modes of vibration. Tr. 23,442 (Sarsten).

stresses below the 7000 psi DEMA limit. LILCO Findings at 3, 10, 12, passim. We do not agree with LILCO. Although Dr. Chen's calculational method, TORVAP-C, is apparently similar to Professor Sarsten's COMHOL method, Dr. Chen's calculations do not accurately reflect the nominal torsional vibratory stresses in the replacement crankshafts. First, Dr. Chen accounted for only 12 orders of vibration instead of 24. As previously noted, the stresses from the additional 12 orders contribute to the accuracy of the result. Second, the harmonic coefficients, or Tn values, used in Dr. Chen's analysis are based on a table appearing in Lloyd's standards rather than on the values used by the other expert witnesses which are based on actual cylinder pressure measurements taken from one of the EDGs. Lloyd's Tn values are well known to be too low (Tr. 23,523 (Sarsten)), and are less accurate than the Tn values used by FaAA, Dr. Pischinger and Professor Sarsten. Tr. 23,444, 23,524 (Sarsten). Lloyd's Tn values are more appropriately used for calculating allowable torsional vibration levels under Lloyd's rules. Tr. 23,523 (Sarsten). Indeed, when Dr. Chen calculated the sum of only six major orders on a hand calculator using the Tn values based on actual cylinder pressure measurements, he obtained stresses between 6600 and 6700 psi. Tr. 23,035-36, 23,075 (Chen).

83. Finally, we feel confident in finding the replacement crankshafts do not satisfy the DEMA recommendations because we further find tha the stresses calculated by Professor Sarsten, FaAA and Dr. Pischinger should be higher, as they are based upon Tn values that are inaccurately low. In its proposed findings, LILCO incorrectly asserts that Professor Sarsten agrees that these Tn values are accurate. LILCO's Findings at 21. The testimony to which LILCO cites does not support its assertion. In fact, Professor Sarsten testified that these Tn values are slightly non-conservative and may represent a lower bound on the true Tn values. Tr. 23,412, 23,418 (Sarsten).

84. The Tn values used by Professor Sarsten, FaAA and Dr. Pischinger are based on actual cylinder pressure measurements taken from cylinder number 7 on EDG 103. Tr. 22,866 (Johnston); LILCO Ex. C-17 at 3-2; LILCO Ex. P-5; LILCO Ex. P-35. Using these measurements, FaAA calculated a mechanical efficiency of 1.0 for the EDGs rather than the expected .88. Tr. 22,874 (Johnston); LILCO Ex. C-17 at 3-3. FaAA concluded that the difference is probably explained by either the pressure measurements being too low or top dead center being shifted. Id.; Tr. 22,874 (Johnston).

85. We find that the pressure measurements are too low. Since FaAA obtained a mechanical efficiency of 1.0 from the measurements, the indicated mean effective pressure ("imep") should equal the brake mean effective pressure ("bmep"). Tr. 23,603-04 (Sarsten, Henriksen). However, the County's calculations show that the imep is less than the bmep which indicates that the measurements should be higher. Tr. 23,605 (Henriksen); 24,458 (Eley).36/

C. FaAA's Calculated Safety Factor is Insufficient Proof that the Replacement Crankshafts are Adequate.

86. LILCO has further urged that we find the replacement

36/ The imep can be calculated from LILCO Ex. P-35, the digitalized data of these pressure measurements. Tr. 23,601-03; 23,727 (Sarsten); 24,259 (Eley); LILCO Ex. P-35. The County calculated the imep from these data and obtained an imep of approximately 205, or 91.3% of the full load rating of 225 bmep at 3500 kW. Tr. 24,256, 24,258 (Eley). The fact that this cylinder was not developing full power when the engine was operating at full load indicates that the other cylinders must have been developing greater than full power, *i.e.*, greater than 225 imep. *Id*. The reported firing pressure measured in cylinder number 7 was approximately 1580 psi. *Id*. Had that cylinder been developing full power, its firing pressure would have been approximately 1677 psi. Tr. 24,256-57 (Eley). Since the other cylinders were developing greater than full power, the firing pressures in those cylinders must have been greater than 1677 psi. Tr. 24,257 (Eley). Assuming that all of these cylinders were in perfect balance, the firing pressures in the cylinders would be approximately 1 percent higher, or 1694 psi.

crankshafts adequate based upon a fatigue analysis it has performed. FaAA performed a fatigue analysis of the replacement crankshafts and calculated an endurance limit factor of safety of 1.48 for operation at full load. LILCO argues that a factor of safety of 1.48 provides sufficient assurance that the replacement crankshafts are adequate for their intended service at Shoreham. The Staff contends that FaAA's calculated safety factor value is insufficient proof of the adequacy of the crankshafts. For the reasons stated below, we agree with the Staff.

87. A factor of safety is an additional margin of strength that is added to a mechanical design to compensate for uncertainties such as the service load, material properties, stress concentration factors and lifetime for the design. In general, a factor of safety in endurance limit is the factor of strength that the part or design has over that which is required for the part to be expected to exhibit infinite life. Whether a particular safety factor value is acceptable or not depends in part on the degree of uncertainties and the difficulty or penalties of adding additional strength to the design. McCarthy, ff. Tr. 22,610, at 38-40; LILCO Ex. C-26 at 2, 4, 6.

88. FaAA calculated a safety factor of 1.48 by comparing the endurance limit of the replacement crankshafts with the stresses to which they are subjected. First, FaAA computed the stress levels in the replacement crankshafts from strain gauge test data from EDG 103. Second, FaAA computed the fatigue endurance limit for the original crankshafts from the ultimate tensile strength of the original crankshaft material and from strain gauge test data from EDG 101. FaAA then calculated the fatigue endurance limit of the replacement crankshafts by scaling the fatigue endurance limit of the original crankshafts upward to account for the higher ultimate tensile strength of the replacement crankshafts. Johnston, ff. Tr. 22,610, at 36-38; LILCO Ex. C-17 at 3-8 to 3-10. LILCO claims that FaAA's 1.48 safety factor value provides sufficient assurance that the replacement crankshafts are adequate for their intended service in the EDGs because the crankshaft's design and expected service are well understood. McCarthy, ff., Tr. 22,610, at 38, 41; Tr. 23,030 (McCarthy).

89. We agree with the Staff, however, that FaAA's calculation of a 1.48 safety factor is not sufficient proof that the replacement crankshafts are adequate. Sarsten, ff. Tr. 23,126, at 16. Absent sufficient testing of the replacement crankshafts that establishes their reliability at rated load

(3500/3900 kW), the assessment of the adequacy of the replacement crankshafts should be based on the large amount of data represented by the appropriate classification societies' rules and their experience in the interpretation of those rules. Id. at 16-17. Such an approach provides a conservative basis for evaluating the adequacy of the replacement crankshafts. Id. at 17. Given an accurate knowledge of a crankshaft's material strength, fabrication process and fatigue endurance limit of the material for given stress cycles, more confidence can be given to an assessment of the adequacy of that crankshaft using the guidelines of a classification society rather than an assessment based solely upon a comparison of the calculated endurance limit with the measured stresses in one crankshaft (FaAA's approach). Tr. 23,528-29, 23,548 (Sarsten); 24,193 (Eley and Christensen).

90. Further, FaAA's calculated safety factor value is insufficient proof of the adequacy of the replacement crankshafts because it is based in part on approximate calculations, such as Sine's method and Miner's rule (Tr. 23,403 (Sarsten); LILCO Ex. C-17 at 3-9, 3-10),37/ and its premises are uncertain. Tr.

37/ For example, to convert the measured strains in the replacement crankshafts into a representative stress state which accounted for the simultaneous effects of shear and bending, FaAA used Sine's method to obtain "equivalent

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23,405 (Sarsten). Indeed, FaAA's calculated fatigue endurance limit, which is used as an input to its safety factor calculation, is based in large part upon the failure of the original crankshafts. Although the failure of the original crankshafts provides a data point from which a safety factor can be calculated and permits some conclusions to be drawn about the strength of the replacement crankshaft, those data are not sufficient to conclude that the replacement crankshafts are adequate. Tr. 23,402 (Sarsten). The data reflect only a single point of reference. Sarsten, ff. Tr. 23,126, at 16; Tr. 24,192, 24,242 (Christensen). The failure of the original crankshafts merely indicates one point on the S-N curve for a crankshaft constructed from a different strength material than the replacement crankshafts. Tr. 23,402 (Sarsten).

91. FaAA's calculated fatigue endurance limit for the replacement crankshafts also is heavily based upon the tensile

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stresses." Equivalent stresses are the "alternating and mean uniaxial stresses that can be expected to give the same life as the given multiaxial stresses." LILCO Ex. C-17 at 3-9. Having calculated these equivalent stresses, FaAA used Miner's rule (linear cumulative damage techniques) to calculate an endurance limit for the original crankshaft. LILCO Ex. C-17 at 3-10; Johnston, ff., Tr. 22,610, at 37.

strengths of the crankshaft material and the methods used for arriving at such values. Tr. 23,467 (Sarsten).38/ In contrast, the endurance limit calculated by Dr. Pischinger (which FaAA did not use as an input to its safety factor calculation) takes into consideration many more factors than the tensile strength, including the grain flow, the degree of forge work, the surface roughness and various factors concerning the relative dimensions of the replacement crankshafts. Tr. 22,768, 22,791 (Pischinger). Significantly, FaAA's calculated endurance limit for the replacement crankshafts (39.2 ksi) is much higher (i.e., much less conservative) than the endurance limit calculated by Dr. Pischinger (25.4 ksi), even though they used the same value for the ultimate tensile strength of the crankshaft material. Tr. 22,990-94, 23,007, 23,045-7 (Pischinger); Johnston, ff. Tr. 22,610, at 37. When Dr. Pischinger's calculated endurance limit is compared with the calculated maximum

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38/ FaAA calculated the fatigue endurance limit of the replacement crankshafts merely by upscaling the calculated endurance limit of the original crankshafts to account for the higher ultimate tensile strength (U.T.S.) of the replacement crankshaft material. Johnston, ff., Tr. 22,610, at 37. In doing so, however, FaAA cited 103 ksi as the minimum tested U.T.S. of the replacement crankshafts (LILCO Ex. C-17 at 3-10), when in fact two of the replacement crankshafts have a lower minimum U.T.S. of 100.777 ksi. LILCO Ex. C-12; Tr. 22,993 (Montgomery); 24,127 (Eley).

stresses in the replacement crankshafts (24.6 ksi and 24.9 ksi) (LILCO Ex. C-17 at 3-9; Tr. 22,790-94 (Pischinger)), the margin of safety is substantially reduced. In sum, given Dr. Pischinger's more conservative endurance limit value, and the absence of any evidence to indicate that it is not just as reliable as FaAA's, we are unable to conclude that the 1.48 safety factor calculated by FaAA is in fact reliable.

92. In addition, FaAA's safety factor calculation does not adequately take into consideration the fabrication process for the replacement crankshafts. Slab-forged and hot-twisted crankshafts such as the replacement crankshafts will yield anisotropic (inhomogeneic) mechanical properties throughout the crankshaft. In contrast, crankshafts fabricated by the closed-forged method will have isotropic properties, i.e., the maximum mechanical properties will exist throughout the overall surface. More significantly, slab-forged and hot-twisted crankshafts will display a definite gradient in mechanical properties from centerline to surface. Thus, some areas of slab-forged and hot-twisted crankshafts will display lower mechanical properties. Bush, ff. Tr. 23,126, at 16; Tr. 23,153, 23,173 (Bush).

93. Although LILCO asserted that one of the original crankshafts did not appear to have anisotropic effects (Tr. 23,175 (Wachob)), there is insufficient evidence in the record to find conclusively that the replacement crankshafts do not have anisotropic effects. In fact, the Staff recently issued a letter requesting additional specific information about the replacement crankshafts' forging process, the locations of the tensile specimens relative to forged surfaces, the criteria used in selecting and testing those specimens, the uniformity of tensile properties through the thickness of the crankshaft and the degree of anisotropy. Tr. 25,365 (Berlinger).

94. Finally, as we have already held, the safety factor calculations under the IACS rules and the safety factor calculations performed by the ABS demonstrate that the replacement crankshafts do not comply with these rules. _See_ discussion, Section II.A.2, _supra_. Unlike the ABS and the IACS, which assess the adequacy of crankshafts by calculating safety factor values and comparing those values against safety factor values that were specifically derived through experience with diesel engine crankshafts, FaAA compares its calculated safety factor value against general safety factor values set forth in basic design texts. These general values are applicable to a wide range of machine components and do not necessarily include

crankshafts such as those in the EDGs. McCarthy, ff. Tr. 22,610 at 40-41; LILCO Ex. C-26; Tr. 22,895-99 (McCarthy). Thus, we have no reliable basis upon which to find that the FaAA comparisons are valid as applied to the replacement crankshafts.

95. In conclusion, although FaAA's calculated safety factor value has some significance, we lack sufficient confidence in this calculation to conclude that the replacement crankshafts are adequate. Tr. 23,548 (Sarsten). Our confidence is further diminished by the failure of the replacement crankshafts to comply with the DEMA recommendations as well as the rules of the classification societies and safety factor calculations which are based upon a more extensive body of scientific knowledge about the design of crankshafts. Therefore, we find that LILCO has failed to establish the adequacy of the replacement crankshafts by a preponderance of the evidence.

D. Any Increase in the Fatigue Endurance Limit of the Replacement Crankshafts from Shotpeening is not Quantifiable.

96. A further issue of dispute among the parties is whether the shotpeening of the replacement crankshafts enhanced their fatigue endurance limit and, if so, whether that increase is quantifiable. We find that even if shotpeening enhanced the fatigue endurance limit of the crankshafts, the increase is not

quantifiable. Therefore, LILCO cannot rely upon shotpeening as a basis for seeking approval of the replacement crankshafts.

97. The crank pin fillet regions of all three replacement crankshafts were shotpeened. Shotpeening is a surface cold-working process that produces a shallow layer of residual compressive stress on the surface of the metal being treated. The process generally consists of the bombardment of the metal surface with small beads of metal propelled by air pressure at high velocity. Anderson, et al., ff. Tr. 23,826, at 133, 136. The crankshaft for EDG 101 was shotpeened once, by Metal Improvement Company at the Shoreham plant, while the crankshafts for EDGs 102 and 103 were shotpeened twice, once by TDI in Oakland and once again by Metal Improvement Company. Id. at 136.

98. Although the Staff witness, Dr. Bush, stated that the shotpeening of the replacement crankshafts should have somewhat enhanced their fatigue resistance, he could not quantify any actual increase in fatigue resistance because certain values were not unequivocably known. Given sufficient information and testing, it is possible to determine whether shotpeening has in fact produced a quantifiable increase in the fatigue endurance limit of a particular object. In order to make any such

quantification, one would need to know, for example, the maximum level of torsional stresses and their distribution, the bending moments and stresses, whether those stresses are in phase or out of phase with the torsional stresses, as well as the level of residual stresses in, and condition of, the crank pin fillets. Tr. 23,152 (Bush). Another major unknown variable which makes quantification impossible is that the fabrication process for the replacement crankshafts, the slab-forged and hot-twisted method, may have caused a marked anisotropic effect. Id. The anisotropic effect is a very pronounced change in mechanical properties throughout the thickness of the crankshaft material because of the different degree of forge work in certain areas. Tr. 23,153 (Bush). In crankshafts fabricated by the slab-forged and hot-twisted method, the maximum mechanical properties will not exist throughout the overall surface. Bush, ff., Tr. 23,126, at 16.

99. The FaAA witnesses agreed that any actual increase in the fatigue endurance limit of the replacement crankshafts could not be assessed or precisely quantified based on the available information. They also testified that assessing any actual increase depends on certain factors that were unknown. In particular, such an assessment requires a rather precise knowledge of the residual stresses in the as-machined fillets

of the crank pins, as well as knowing quite precisely the surface conditions of the fillets, including the existence of any machining irregularities, and other unknown factors. Tr. 23,134 (Wells). According to LILCO's witnesses, none of these various factors could be measured when LILCO received the crankshafts. Tr. 23,134 (Wells).39/

100. FaAA itself could not support a quantification of any increase in the fatigue endurance limit from shotpeening. Although FaAA originally stated in its April 1984 draft crankshaft report that a conservative range of values for the expected increase in the fatigue endurance limit of the replacement crankshafts from shotpeening was 5% to 20%, the final May 1984 version of that report deleted that reference and attributed no numerical value to the alleged increase in the fatigue endurance limit. Anderson, et al., ff. Tr. 23,826, at 128-29; LILCO Ex. C-17 at 3-11. When FaAA performed its quality assurance review of the April crankshaft report, it could not locate any documentation in the technical literature or any test data to provide a basis for comparing the shotpeened

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39/ For example, Dr. Wells testified that it was "not practical" to perform x-ray diffraction analyses of the residual stresses because of the size of the crankshafts, whereas other measures would have required some degree of destructive examination. Tr. 23,134 (Wells).

replacement crankshafts with the original unshotpeened crankshafts. Thus, FaAA was unable to support any quantitative improvement in the fatigue endurance limit of the replacement crankshafts from shotpeening. Tr. 23,131 (Wells). In fact, no tests or measurements have been conducted to verify whether shotpeening produced any increase in the fatigue endurance limit of the replacement crankshafts. Tr. 23,131 (Wells).40/

101. Mr. Burrell testified on behalf of LILCO that the shotpeening of the fillet areas of the replacement crankshafts resulted in an increase of approximately 15-20% in the fatigue endurance limit. Burrell, ff. Tr. 23,122, at 22. We give no weight to Mr. Burrell's testimony. First, Mr. Burrell's opinion was based upon his experience with fatigue tests on different sizes and types of crankshafts. The largest crankshaft for

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40/ Dr. Wells and Dr. Wachob both stated in their written testimony that they could not precisely quantify any amount of increase in the fatigue endurance limit of the crankshafts due to shotpeening. They opined, however, that the increase is significant and is "not inconsistent with" the 15-20% range indicated by Mr. Burrell, whose testimony we address in the text. Wells and Wachob, ff. Tr. 23,122, at 22. Dr. Wells testified that his opinion was that the increase should be at least 10% and conceivably as high as 20% or 30% based on his experience at Pratt & Whitney Aircraft. We give little weight to these opinions. Dr. Wells' opinion was based upon his experience with the application of shotpeening to components other than crankshafts for diesel engines. Tr. 23,158 (Wells). Dr. Wachob provided no basis for his opinion.

which he had fatigue test data, however, was one with a journal bearing diameter of 6-1/4 inches. Tr. 23,135 (Burrell). In contrast, the nominal journal bearing diameter of the Shoreham replacement crankshafts is 12 inches. Christensen and Eley, ff. Tr. 23,286, at 106-07. Mr. Burrell had no fatigue test data on crankshafts as large as those at Shoreham. Tr. 23,135 (Burrell).

102. Next, with respect to the fatigue test data on the 6-1/4 inch crankshaft, Mr. Burrell testified that the test was performed about 5 years ago by Standard Pressed Steel of Jenkinstown, Pennsylvania, but he did not know who in that organization performed the testing. Tr. 23,140 (Burrell). Mr. Burrell was told by Standard Pressed Steel that the test showed a 17% increase in the fatigue endurance limit from shotpeening. Tr. 23,142 (Burrell). Mr. Burrell did not know, however, whether Standard Pressed Steel had performed x-ray diffraction on the test crankshaft before or after the testing or whether in fact any x-ray diffraction was performed at all. Nor did Mr. Burrell know what destructive tests were performed on the shafts. Tr. 23,143 (Burrell). Although Mr. Burrell stated that the fatigue test involved running a shotpeened crankshaft and an unshotpeened crankshaft to failure and comparing the fatigue results, Mr. Burrell did not know whether the parameters

of the tests were identical for both crankshafts, even though the parameters would have to be identical to make a proper correlation. Tr. 23,144 (Burrell). Furthermore, Mr. Burrell did not know the horsepower, torsional vibration characteristics or revolutions per minute of the engine for which the 6-1/4 inch crankshaft was intended, other than that it was a 16 cylinder engine. Tr. 23,141 (Burrell).

103. The testing data on which Mr. Burrell relied were contained in an article he authored. Tr. 23,142 (Burrell). Dr. Wells testified that even if FaAA had had a copy of Mr. Burrell's article when preparing its May 1984 report on the replacement crankshafts, it would have been very difficult for FaAA to support a finding that the shotpeening had produced a percentage increase in the fatigue endurance limit based on the information contained in the article because FaAA did not know in what manner the particular crankshaft test specimens referred to in the article had been machined. Tr. 23,146 (Wells). Furthermore, the test results were relative only. Id. While the tests showed an increase in fatigue strength from the as-manufactured unshotpeened condition to the final shotpeened condition, the as-manufactured condition of the test specimen crankshaft before it was shotpeened was not known. Id.

104. Thus, Mr. Burrell did not know the type of information which both the Staff and LILCO agree is necessary to quantify any increase in the fatigue endurance limit. As such, his testimony that the shotpeening increased the fatigue endurance limit of the replacement crankshafts by 15-20% is unreliable.

105. In contrast to Mr. Burrell's experience, TDI recommended against shotpeening the replacement crankshafts, based upon its experience and the experience of its metallurgical consultant that shotpeening would not substantially increase the fatigue strength of the material. County Ex. 10 at 2-5; County Ex. 51 at 4. In addition, TDI was informed by Kobe Steel, Ltd., a Japanese manufacturer of crankshafts for TDI, that shotpeening crankshafts of this size is "a waste of time." County Ex. 48.

106. In sum, we find that there is insufficient evidence in the record to conclude that shotpeening increased in any significant respect the fatigue endurance limit of the replacement crankshafts. Any such increase is not quantifiable on the basis of the available evidence, and thus cannot form the basis for any finding that the replacement crankshafts are adequate.41/

41/ The parties also disagree on whether the first shotpeening of the replacement crankshafts adversely affected the

(Footnote cont'd next page)

III. CONCLUSION

107. Based upon all of the foregoing, we conclude that LILCO has failed to establish by a preponderance of the evidence that the replacement crankshafts in the Shoreham EDGs are adequate for operating at rated loads. We hold, therefore, that LILCO has failed to establish that the EDGs comply with the requirements of GDC 17.

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(Footnote cont'd from previous page)

crankshafts and, if so, whether those adverse effects were corrected by repeening. We find that although the original shotpeening possibly damaged the crankshafts, any damage appears to have been corrected by the second shotpeening.

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November 15, 1984

UNITED STATES OF AMERICA
NUCLEAR REGULATORY COMMISSION

Before the Atomic Safety and Licensing Board

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

CERTIFICATE OF SERVICE

I hereby certify that copies of **SUFFOLK COUNTY AND STATE OF NEW YORK PROPOSED FINDING OF FACT AND CONCLUSIONS OF LAW ON EMERGENCY DIESEL GENERATORS (REPLACEMENT CRANKSHAFTS)**, dated November 15, 1984, have been served to the following this 15th day of November, 1984 by U.S. mail, first class, by hand when indicated by one asterisk, and by Federal Express when indicated by two asterisks.

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