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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	)	Docket No. 50-322-OL
	)	
(Shoreham Nuclear Power Station, Unit 1)	)	

LONG ISLAND LIGHTING COMPANY'S PROPOSED FINDINGS OF FACT

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I. Background

1. The crankshaft in a diesel engine such as the Emergency Diesel Generators (EDGs) at Shoreham converts the reciprocating motion of the pistons and connecting rods into rotary motion. In this process the crankshaft converts the inertial and gas pressure firing forces into torque. The torque from the crankshaft drives the electrical generator to provide emergency power. (McCarthy et al, ff Tr. 22,610 at 7).[1/]

2. The original crankshafts provided by TDI had a 13-inch main journal and an 11-inch crankpin and the fillet regions were not shotpeened. On August 12, 1983, the original 13-inch by 11-inch crankshaft on EDG 102 fractured through the crankpin and rear web under cylinder No. 7. Subsequent investigation revealed that the crankshaft on EDG 101 was cracked at the No. 5 and No. 7 crankpins and the crankshaft on EDG 103 was cracked at the No. 6 crankpin. (Montgomery, ff Tr. 22,610 at 7-9).

3. The failure of the original crankshafts was caused by high cycle vibratory fatigue. The crankshafts were unable

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1/ The prefiled testimony of Messrs. McCarthy, Johnston, Montgomery and Chen and Messrs. Pischinger and Youngling were not numbered when bound into the transcript. Both sets of testimony may be found following page 22,610 of the transcript.

to withstand the torsional stresses imposed upon them during operation of the engine. (McCarthy, Johnston, ff Tr. 22,610 at 7-8).

4. The original crankshafts did not comply with the Diesel Engine Manufactuers Association (DEMA) recommendations for allowable crankshaft vibratory stress. (Tr. 22,840 Johnston; Tr. 22,841 Chen).

5. LILCO replaced the crankshafts in all three engines with crankshafts of a different design. The replacement crankshafts have a 13-inch main journal and a 12-inch crankpin. The crankpin-to-web fillet radii of the replacement crankshafts have a larger radius of curvature than the fillet radii of the original crankshafts, and the fillet regions of the replacement crankshaft have been shotpeened. (Montgomery, ff Tr. 22,610 at 8-9).

6. The minimum ultimate tensile strength of the replacement crankshafts is over 100,000 psi. The average ultimate tensile strength of the original crankshafts was approximately 93,500 psi. (Montgomery, ff Tr. 22,610 at 9).

7. The replacement crankshafts have undergone extensive engineering analyses and testing to determine their adequacy for service in the Shoreham EDGs. Failure Analysis Associates (FaAA), Power and Energy International (PEI) and FEV have analyzed the replacement crankshafts, and all concluded

that the crankshafts are adequate for service in the Shoreham EDGs at 3500 KW and 3900 KW. FaAA reviewed TDI's torsional analysis and the results of torsiograph tests to determine that the crankshafts comply with DEMA. FaAA also conducted a full scale fatigue analysis based on testing to determine a true factor of safety for the crankshafts. FaAA found the factor of safety was 1.48. PEI performed a torsional analysis typically used by the diesel engine industry to determine whether a crankshaft complies with DEMA. The nominal torsional stresses in the replacement crankshafts are well below the DEMA limits. FEV calculated a factor of safety for the replacement crank-shafts under the Kritzer-Shahl criteria. The crankshafts have a margin of safety of 24%. (Montgomery, ff Tr. 22,610 at 8-9; Chen, ff Tr. 22,610 at 21; Tr. 23,004 Pischinger).

II. The Crankshafts are Only Required to Comply with DEMA. The Crankshafts do not Have to Comply with the Requirements of any Other Design Society

8. The purchase specifications for the Shoreham EDGs, Spec. No. SH1-89, Revision 2, January 26, 1983, required that the replacement crankshafts meet the DEMA recommendations for allowable vibratory stress. (Montgomery, ff Tr. 22,610 at 10; LILCO Diesel Exhibit C-2).

9. NRC Regulatory Guide 1.9, Revision 2, (Reg. Guide 1.9) addresses the design requirements for diesel generator units used as standby electric power systems at nuclear power plants. Reg. Guide 1.9 provides that conformance with the requirements of IEEE Std 387-1977 is acceptable for meeting the design criteria and qualification testing of diesel generator units used as onsite electric power systems for nuclear power plants. IEEE Std 387-1977 provides that diesel generators should comply with the standards of DEMA's Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines. (Montgomery, ff Tr. 22,610 at 11-12; LILCO Diesel Exhibits C-3, C-4; Henriksen, Sarsten, ff Tr. 23,126 at 10-11).2/

10. DEMA is not a design code in the sense that DEMA does not provide detailed rules that tell an engine manufacturer how to design a crankshaft. However, DEMA does provide specific stress limits to measure the adequacy of a crankshaft. Engine manufacturers have used DEMA for years on stationary diesel generator installations to determine whether a crankshaft is adequate for its intended service. (Chen, ff Tr. 22,610 at 13-14).

2/ The prefiled testimony of Messrs. Berlinger, Bush, Henrikson, Laity and Sarsten was not numbered when bound into the transcript. The testimony may be found following page 23,126 of the transcript.

11. Suffolk County asserts that the crankshafts do not comply with the maximum horsepower requirements of Lloyd's Register of Shipping (Lloyd's), do not meet the torsional or web thickness requirements of the American Bureau of Shipping (ABS), do not comply with the minimum safety factor under CIMAC, and do not comply with the criteria used by FEV. (Christensen, Eley, ff Tr. 23,826 at 114-21, 123-32). The record demonstrates that the crankshafts have unlimited life at 3500 KW and at least 1200 hours of life at 3900 KW under the Kritzer-Stahl criteria used by FEV. (Pischinger, ff Tr. 22,610 at 5). A CIMAC calculation performed by ABS shows that the crankshafts exceed the minimum CIMAC factor of safety of 1.15. (Suffolk County Diesel Exhibit 43 at 29). ABS has determined that the crankshafts meet ABS torsional requirements. (LILCO Diesel Exhibit C-13). ABS has approved the sizing of the webs (Montgomery, ff Tr. 22,610 at 17) and the NRC Staff's consultants believe the webs meet the ABS requirements. (Henriksen, Sarsten, ff Tr. 23,126 at 11; Staff Diesel Exhibit 1).

12. The rules, standards and design methodologies of marine classifications societies vary widely and, in fact, provide differing acceptance criteria for the same crankshaft design parameters (e.g., journal/pin sizing, allowable horsepower, allowable torsional stress levels, etc.). A crankshaft may not meet the criteria of certain codes and be perfectly adequate under other codes. (Chen, ff Tr. 22,610 at 14).

13. A crankshaft may be entirely adequate for its intended service and not comply with the rules of ABS, Lloyd's or CIMAC. While compliance with one of the codes generally provides assurance that a crankshaft is adequate, noncompliance does not necessarily mean a crankshaft is inadequate. Rather, noncompliance merely means a crankshaft does not meet the design requirements of a particular code. If a crankshaft is not required to meet that code by specification or other requirement (e.g., insurance purposes, licensing requirements, etc.), and there is assurance from other sources that the crankshaft is adequate, noncompliance is not significant. (Chen, ff Tr. 22,610 at 15-16).

14. Suffolk County witnesses argue that the crankshafts should be required to comply with the design criteria of all major classification societies. (Christensen, Eley, ff Tr. 23,826 at 113-14). There is, however, no regulatory requirement that the crankshafts comply with any standard other than DEMA. (Montgomery, ff Tr. 22,610 at 11-12; Henriksen, Sarsten, ff Tr. 23,126 at 10-11)

15. Good design practice does not require that diesel generators in nuclear standby service meet any of the rules or requirements established by various marine classification societies. The rules of the classification societies are for engines designed to operate in marine applications. Marine

engines are exposed to conditions far different from those for standby engines at nuclear power plants. (Henriksen, Sarsten, ff Tr. 23,126 at 11; Chen, ff Tr. 22,610 at 15-16; Tr. 22,708 Pischinger).

III. <u>The Crankshafts Comply with DEMA</u>

16. The DEMA recommendations for allowable crankshaft vibratory stress provide:

> 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, connecting shafts, flange or coupling components, etc., made of conventional materials, torsional vibratory conditions 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.

DEMA's Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines was last revised in 1972. The allowable limits for torsional vibratory stresses, however, have not been revised since at least 1958. (LILCO Diesel Exhibit C-14; Tr. 22,710-12 Chen).

17. The DEMA allowable stress limits are based on the assumption that the crankshaft is manufactured from

conventional materials. At the time the DEMA allowables were established, the average ultimate tensile strength of conventional material used in crankshafts was between 60,000 and 70,000 psi. (Tr. 22,711 Chen). In contrast, the minimum ultimate tensile strength of the replacement crankshafts is over 100,000 psi. (Montgomery, ff Tr. 22,610 at 9).

18. In a four-stroke engine such as the Shoreham EDGs, harmonics of the order 0.5, 1.0, 1.5, 2.0, 2.5, 3.0 . . . exist. These orders continue infinitely. (Johnston, Chen, ff Tr. 22,610 at 23; Tr. 23,048 Johnston). At the time the DEMA stress allowables were first adopted, the only available method of calculating combined stresses involved laborious, time-consuming hand calculations. (Tr. 22,742 Pischinger). Not until the development of sophisticated, high speed digital computers in the mid-1960s was it possible as a practical matter to calculate the combined stresses of more than six orders. Current computational techniques permit the summation of 24 orders. (Tr. 22,989-90 Pischinger). Stresses are typically not calculated for orders higher than number 12 (a total of 24, including half orders) because the stresses caused by the higher orders are insignificant. (Tr. 23,253 Sarsten).

19. With the development of sophisticated computers that permit the calculation of combined nominal stresses of 24 orders in all modes, and the development of better quality

steel, the trend among classification societies has been to become less conservative in their rules for allowable stress. (Tr. 22,995-96 Pischinger). DEMA, on the other hand, has not revised its allowable stresses since the 1950s. (Tr. 22.710-12 Chen). This indicates that DEMA has a large built-in margin of safety.

20. The DEMA allowables were established as a result of experience with many crankshafts. The allowables have to be correlated with the analytical techniques that were used to analyze the stresses in those crankshafts. Since the allowable limits are based on stress values that were calculated by a certain technique (i.e., Holzer forced vibration calculations), it is appropriate to perform that type of calculation to determine whether a crankshaft meets the DEMA limits. It is not appropriate to use a technique that did not exist at the time the limits were established. (Tr. 22,851-53 Johnston). The Holzer forced vibration calculations performed by TDI are standard techniques used to determine whether a crankshaft complies with DEMA. (Tr. 22,755-56 Johnston). TDI's single order Holzer calculations show the replacement crankshafts comply with DEMA. (Johnston, ff Tr. 22,610 at 24).

21. The standard practice in the diesel engine industry is to sum four or six orders for purposes of comparison with the DEMA allowables. (Tr. 22,729-30, 22,832 Chen). In certain

instances, as few as two orders may be summed. For example, ABS summed only two orders when reviewing the stresses in the replacement crankshafts. (Tr. 22,738 Johnston). The TORVAP C computer program used by PEI, which is a common domain software program widely used by the diesel engine industry to calculate nominal stress, is designed to sum only six orders. (Tr. 22,745, 22,747 Chen).

22. The calculation PEI performed using the TORVAP C computer program is a modal superposition calculation. (Tr. 22,716-18 Chen). FaAA and Professor Sarsten also calculated nominal torsional stresses using modal superposition or harmonic synthesis analyses. The distinction between PEI's calculations and the calculations performed by FaAA and Professor Sarsten lies primarily in the number of orders summed. (Tr. 23,034-35 Chen). FaAA and Professor Sarsten did not sum major orders. They summed all (24) orders. PEI performed two calculations, one in which six orders were summed and one in which twelve orders were summed. The calculation that is most appropriate to compare with the DEMA allowables is the PEI calculation that sums six orders. (Tr. 22,729 Chen). FaAA and Professor Sarsten summed many more orders than are included in the term major orders. (Tr. 22,734 Johnston).

23. The NRC Staff asserts that the crankshafts do not comply with DEMA. Professor Sarsten's calculations show that

the combined nominal stresses in the crankshafts slightly exceed 7000 psi at full load and at overspeed and underspeed. Professor Sarsten, however, made no attempt to sum the major orders. Rather, he summed all 24 orders in his calculations. (Sarsten, ff Tr. 23,126 at 13-14; Staff Diesel Exhibit 2).

24. Professor Sarsten has no experience with the application of DEMA. He has never used DEMA prior to his involvement in this case. (Tr. 23,255 Sarsten).

25. DEMA does not require that the combined stress of all (i.e., 24) orders be less than 7000 psi. Rather, DEMA only requires that the combined stress of the major orders of vibration be less than 7000 psi. DEMA does not define major orders, because they are different for every engine and depend on a number of variables. The determination of what the major orders are for each crankshaft is left to sound engineering judgment. (Tr. 22,741, 22,832 Chen).

26. The major orders in the replacement crankshafts at Shoreham are the 4.0, 5.5, 4.5 and 2.5 orders. (Tr. 22,739, 22,747-48 Johnston; Tr. 22,741 Chen; LILCO Diesel Exhibit C-17 at 3-14; LILCO Diesel Exhibit C-18 at 11). The major orders are those that give the largest free end vibrational amplitudes. (Tr. 22,741 Chen).

27. PEI's torsional calculations were performed in two parts. First, PEI calculated the single order stresses for the

first 20 orders (1 through 10, including half orders) using the TORVAP R program. TORVAP R is a classical Holzer forced vibration calculation. (Tr. 22,728-31, 22,843 Chen). Second, PEI selected the six largest orders and calculated the combined stresses using the TORVAP C program. TORVAP C is a modal superposition calculation. As an added measure of conservatism, PEI designed a special subroutine for TORVAP C and calculated the combined stresses for the twelve largest orders. (Tr. 22,745-47 Chen).

28. The six orders used by PEI, in its TORVAP C calculation include the 0.5, 1.5, 2.5, 4.0, 4.5 and 5.5 orders. (Tr. 22,749 Johnston; LILCO Diesel Exhibit C-18 at 10).

29. Dr. Chen, the president of PEI, was chairman of the DEMA technical committee from 1971 to 1973 and has worked in the American diesel engine industry since 1952. (Chen, ff Tr. 22,610 at 4, 30). The calculations performed by PEI are typical of a calculations performed by the diesel engine industry to check the adequacy of a crankshaft to withstand torsional stress. These calculations show that the crankshafts comply with DEMA at full load (3500 KW), overload (3900 KW), overspeed (105%) and underspeed (95%). (Chen, ff Tr. 22,610 at 28-30).

30. PEI's single order calculations show the crankshafts comply with DEMA. The nominal stresses for the number 4.0 order (the largest single order) are well below the DEMA allowable of 5000 psi.

ENGINE SPEED (RPM)	LOAD (KW)	NOMINAL STRESS (PSI)
450	3500	3455
450	3900	3740
427.5	3500	3071
472.5	3500	4010

(Chen, ff Tr. 22,610 at 29).

31. PEI's combined stress calculations for the six largest orders show the crankshafts comply with DEMA. The nominal stresses for the major orders are well below the DEMA allowable of 7000 psi.

ENGINE SPEED (RPM)	LOAD (KW)	NOMINAL STRESS (PSI)
450	3500	5101
450	3900	5401
427.5	3500	6232
472.5	3500	5673

(Chen, ff Tr. 22,610 at 29-30).

32. As an added measure of conservatism, PEI summed an additional six orders and created a special subroutine for TORVAP C. The combined nominal stress for twelve orders (at least double the number typically summed for a DEMA calculation) at full load (3500 KW) and rated speed (450 rpm) is 6020 psi, well below the 7000 psi allowable. (LILCO Diesel Exhibit C-18 at 10).

33. The nominal stresses in the original 13-inch by 11-inch crankshafts significantly exceeded the DEMA limits.

The single order stress for the number 4.0 order was 6200 psi. The combined stress, based on the summation of only four orders, was 9000 psi. (Tr. 22,841, 22,969-70 Chen).

34. One of the inputs to any calculation of nominal torsional stress is the gas pressure tangential effort, or Tn values. PEI used Tn values provided by Lloyd's Register of Shipping in its calculations. Lloyd's Tn values were the highest Tn values published in an available common domain reference. At the time PEI performed its calculations, the FaAA Tn values were not available. In addition, PEI did not want to use private data that it had not generated. In determining the nominal stresses on a crankshaft, it is commonly accepted practice to use common domain Tn values such as Lloyd's. (Tr. 22,853-56 Chen). Lloyd's Tn values are higher than the Tn values used by TDI for orders above number 4.0 and lower than the values used by TDI for orders below number 4.0. (LILCO Diesel Exhibit C-18 at 13).

35. FaAA measured the pressures in the Shoreham EDGs to obtain a pressure versus time curve. This curve allowed FaAA to develop accurate Tn values for the replacement crankshafts. (Tr. 22,814, 22,850 Johnston). FaAA's Tn values are higher than Lloyd's Tn values. (LILCO Diesel Exhibit C-18 at 13; LILCO Diesel Exhibit C-17 at 3-13). However, even if PEI had used FaAA's Tn values, the nominal stresses would still be safely within the DEMA allowable limits. (Tr. 23,035-36 Chen).

36. Stone & Webster performed torsiograph tests on the replacement crankshaft in EDG 103 in January, 1984. The torsiograph tests measured the total torsional vibrations resulting from all orders. FaAA converted the torsional vibrations into stresses for comparison with DEMA. (Johnston, ff Tr. 22,610 at 24; Tr. 22,814 Johnston).

37. The torsiograph provides the angular displacement response of the free end of the crankshaft as a function of time. This displacement may be decomposed into components corresponding to each order. The torsiograph also provides the peak-to-peak response. These responses are used to calculate the nominal stresses. (Johnston, ff Tr. 22,610 at 26).

38. In order to convert the amplitude of free end rotation into nominal shear stresses, those measurements must be multiplied by a factor of 9562 psi per degree. This number (9562) is a stress that occurs on the crankshaft by applying a rotational displacement of one degree at the free end of the crankshaft, assuming that the shape of the crankshaft is in the first mode of vibration. (Tr. 22,837 Johnston).

39. In converting torsiograph data into nominal stress, it is customary to assume a single mode of response. In converting the measurements, FaAA assumed a first mode of response. This type of approach is found in many common textbooks and is the approach used by ABS. (Tr. 22,838, 22,850 Johnston).

40. The conversion factor of 9562 psi was calculated using the first mode of response. It would be possible to calculate a similar conversion factor using the second, third or any other mode of response. The replacement crankshafts, however, vibrate primarily in the first mode. The conversion factor based on the first mode of response was used because it represents a customary way of reducing torsiograph test data. (Tr. 22,838-39 Johnston).

41. While the principle of using the first mode of response to reduce torsiograph data is common, the principle of using a half peak-to-peak is a very conservative approach to reducing torsiograph data. Much of the data in the past has been reduced based on the square root of the sum of the squares (SRSS) of individual orders. If the SRSS method had been used to convert the torsiograph data into stresses the result would have been in the range of 4000 psi. (Tr. 22,839 Johnston).

42. The torsiograph data showed that the single order and combined stresses are below the DEMA allowable limits. At 100% load (3500 KW) the fourth order stress is 3108 psi and the combined stress is 6626 psi. At overload (3900 KW) the fourth order stress is 3287 psi and the combined stress is 6958 psi. (Johnston, ff Tr. 22,610 at 26; LILCO Diesel Exhibit C-17 at 2-11).

43. Suffolk County presented no evidence to show whether or not the replacement crankshafts complied with DEMA. Professor Christensen did not perform any forced torsional vibratory calculations. (Tr. 23,966 Christensen). Mr. Eley is not capable of performing forced torsional vibratory calculations. (Tr. 23,968 Eley). Neither Professor Christensen nor Mr. Eley have had any experience with DEMA prior to this case. (Tr. 23,975-76 Christensen, Eley).

IV. The Crankshafts Have Been Approved by ABS

44. ABS has certified that the material properties of the crankshaft conform to the requirements for ABS grade 4 steel. (LILCO Diesel Exhibit C-12).

45. ABS has approved the dimensional sizing for the diameter of the pins and the journals of the replacement crankshaft and has approved the proportions of the crankshaft webs. (Montgomery, ff Tr. 22,610 at 17). The NRC Staff also agrees that the sizing of the pins, journals and webs comply with ABS requirements. (Henriksen, Sarsten, ff Tr. 23,126 at 11; Staff Diesel Exhibit 1).

46. Neither Suffolk County nor the NRC Staff believe the replacement crankshafts comply with ABS torsional requirements. ABS, however, has determined that the crankshafts comply with ABS torsional requirements. (LILCO Diesel Exhibit C-13).

47. TDI submitted information to ABS concerning the torsional critical speed arrangment of the replacement crankshafts at Shoreham. On the basis of reviewing the information submitted by TDI and performing its own check calculations, ABS approved the critical speed arrangement of the crankshaft, flywheel and generator at Shoreham. (Montgomery, ff Tr. 22,610 at 17-18; LILCO Diesel Exhibit C-13).

48. ABS summed only two orders when it performed its check calculations for torsional stress. ABS used the SRSS method to calculate torsional stress. (Tr. 22,738 Johnston; Suffolk County Diesel Exhibit 43 at 24-28).

V. Fatigue Analysis

49. FaAA performed a fatigue analysis to determine the true margin of safety of the replacement crankshafts. This analysis was independent from the design criteria specified by any code and shows that the crankshafts have an adequate factor of safety. (McCarthy, Johnston, ff Tr. 22,610 at 31).

50. In order to determine a factor of safety for the replacement crankshafts it is necessary to know two things: (1) the maximum stress the crankshaft will see in service; and (2) the endurance limit of the crankshaft material. FaAA's fatigue analysis established both of these factors. (McCarthy, Johnston, ff Tr. 22,610 at 32).

51. The maximum stress on the crankshaft occurs in the fillet regions. The actual maximum stresses were measured during strain gauge testing on the original crankshaft in EDG 101 in September, 1983, and on the replacement crankshaft in EDG 103 in January, 1984. (Tr. 22,814, 22,889 Johnston).

52. The crankshafts were instrumented with strain gauges at the locations of maximum stress. The location of maximum stress was determined analytically by FaAA. First, FaAA performed a dynamic torsional analysis of the crankshaft to determine the true range of torque at each crank throw. Second, using the results of the dynamic torsional analysis, a finite element model of a one quarter crank throw was used to compute the magnitude and location of peak stresses in the fillet region. The calculated peak stresses corresponded closely with the measured peak stresses. (Tr. 22,889-93 Johnston; McCarthy, Johnston ff Tr. 22,610 at 32).

53. The fatigue endurance limit for the replacement crankshafts was established by first obtaining the endurance limit for the failed crankshafts, and then increasing that limit to reflect the difference in ultimate tensile strength between the failed and replacement crankshafts. (McCarthy, Johnston, ff Tr. 22,610 at 32).

54. FaAA developed a dynamic torsional model of the crankshaft to determine the total torque at each crank throw.

The total torque is calculated by a summation of the torque produced by each order and mode. The analytical method used by FaAA (modal superpositon) computes the phase relationship between the various orders and modes, which permits this summation. (Johnston, ff Tr. 22,610 at 33).

55. One of the inputs into FaAA's dynamic torsional analysis was the Tn values based on pressure measurements taken by a quartz transducer placed in the air start valve of cylinder No. 7 on EDG 103. Cylinder No. 7 was chosen for the pressure measurement because strain gauges were placed on crankpins No. 5 and 7 and FaAA wanted a pressure measurement on a corresponding cylinder. Number 7 was chosen over No. 5 because pressure diagrams are typically more accurate the closer the cylinder is to the location where the top dead center marker was measured at the flywheel, and the nearest cylinder that was strain gauged was No. 7. (Tr. 22,866-67 Johnston).

56. Cylinder No. 7 was not chosen for pressure measurement because of any prediction that the pressure would be the highest in that cylinder. The engines are typically balanced so that the cylinder pressures are approximately equal throughout all of them. FaAA sought neither to find the highest nor the lowest pressure measurement. (Tr. 22,868 Johnston).

57. The type of pressure measurement that is of concern for a torsional analysis is not a peak pressure. What is important is an entire pressure curve. It is important that the pressure curve be typical, because vibrations do not respond to one individual peak pressure, but rather to an accumulation of a series of loadings. That is what causes vibrations to build above a certain level, and is the reason for performing a dynamic rather than a static analysis. The measurement from cylinder No. 7 was taken over many cycles and then averaged in order to calculate an appropriate pressure curve. (Tr. 22,869-70 Johnston).

58. Professor Sarsten agrees that FaAA's Tn values are accurate and used them in his harmonic sythesis calculations. (Sarsten, ff Tr. 23,126 at 13). FEV derived its Tn values from the same pressure data used by FaAA. (Tr. 22,810-12 Pischinger).

59. The total torque calculated by the dynamic torsional model was used as input to the finite element model to determine the actual maximum value and location of stress in the fillet regions. (Johnston, ff Tr. 22,610 at 33).

60. The nominal stresses calculated from the dynamic model are considerably less than the actual maximum stresses in the crankshaft. These nominal values would prevail if the crankshaft were a long circular cylinder. Stresses in the real

crankshaft are greatly influenced by its complex geometry and by stress concentrations, especially at the fillet radii between the main journal and web, and the crankpin and web. In addition, a crankshaft throw is subjected to loads of two basic types: (1) torque transmitted through the throw, which is influenced by the output power level and by the torsional vibration of the crankshaft; and (2) connecting rod forces applied to the crankpin and reacted at bearing supports. (Johnston, ff Tr. 22,610 at 33-34).

61. FaAA used a finite element model of a one quarter crank throw, considering stresses due to torsional loading and stresses due to gas pressure loading, to compute the actual maximum value and location of stress in the crankpin fillet area. The strain gauges used during dynamic testing were placed at the location of maximum stress calculated by the finite element model. (Johnston, ff Tr. 22,610 at 34).

62. FaAA's finite element analysis predicted that the highest stresses would occur in the fillet regions of crankpin Nos. 5 and 7. Two different sets of boundary conditions were used in the finite element analysis. For a determination of the stresses in the crankpin fillet area due to torsional stresses alone, the stresses calculated at the two boundary conditions would be expected to bracket the measured stresses. The stresses measured at crankpin No. 5 are bracketed by the

two finite element models. This is to be expected because the stresses on crankpin No. 5 are due almost exclusively to torsion. (Tr. 22,891 Johnston; LILCO Diesel Exhibit C-17 at 3-17). In regard to crankpin No. 7, the range of principal stress is bracketed by the two boundary conditions, although the range of equivalent stress falls outside the bracket by approximately one and a half percent. (LILCO Diesel Exhibit C-17 at 3-18). This is due to the fact that on crankpin No. 7 there is a small amount of bending stress. The discrepancy is so small, however, that it was not necessary to perform additional analyses using boundary conditions suitable for a bending analysis. (Tr. 22,891-92 Johnston).

63. FaAA performed calculations to compute the maximum bending stresses in the crankshaft. The maximum stress in any crankpin due to bending is 15.5 ksi. The point at which the maximum bending stress occurs is in a different location than the point of maximum torsional stress. The location of maximum bending stress is at the bottom of the crankpin when the pin is at top dead center. The location of maximum torsional stress occurs approximately 45-50° around the crankpin away from the bottom of the pin. The maximum bending stress also occurs at a different time than the maximum torsional stress. The result is that the maximum stress that occurs on the crankshaft, which is the stress of concern when determining a factor of safety, occurs on crankpin No. 5. (Tr. 22,893-94 Johnston).

64. On crankpin No. 7, there is a small overlap in time between the occurrence of the bending stress and the occurrence of the secondary peak of torsional stress, which causes the range of equivalent stress to be 44.5 ksi. This is the number that falls slightly outside the two finite element models. This number is, however, significantly lower than the range of equivalent stress of 49.3 ksi on crankpin No. 5. (Tr. 22,894-95 Johnston; LILCO Diesel Exhibit C-17 at 3-17 and 3-18).

65. The amplitude of equivalent stress is half the range of equivalent stress. The highest range of equivalent stress was on crankpin No. 5 (49.3 ksi). The amplitude of equivalent stress is 24.6 ksi. This amplitude is the value used in the Goodman diagram for purposes of comparison with the endurance limit. (LILCO Diesel Exhibit C-17 at 3-9, 3-32).

66. FaAA's finite element analysis was merely a step in calculating a factor of safety for the replacement crankshafts. The factor of safety was calculated from the stresses measured in the replacement crankshafts. The finite element calculations were performed to demonstrate the location where the strain gauges should be placed on the crankshafts. (Tr. 22,892 Johnston).

67. The strain gauges were placed in the locations of maximum stress that were indicated both around the

circumference of the pin and within the fillet. While the distribution of principal stresses varies by a considerable amount between the two bounding finite element cases, the location of maximum stress is the same under both conditions. It is only the location of the maximum stress that was used as input to the strain gauge tests. (Tr. 22,892-93 Johnston; LILCO Diesel Exhibit C-17 at 3-27 through 3-30).

68. The typical procedure for running a strain gauge test on a crankshaft is to bring the engine to the load of interest and maintain it at that load for approximately ten minutes to stabilize the engine. Data is then taken for approximately two minutes. It is not necessary to stabilize the load for more than ten minutes when taking measurements on a crankshaft because the torsional vibration condition stabilizes very rapidly. The torsional vibration conditions are not dependent on temperature transients or other phenomena that might take a long time to stabilize. This test procedure is also typically followed in taking torsiograph data. (Tr. 22,976 Johnston).

69. The next step in FaAA's fatigue analysis was to compare the measured stresses with the fatigue endurance limit of the replacement crankshafts. The fatigue endurance limit of the replacement crankshaft was established by first obtaining the endurance limit of the failed crankshaft. Since the

endurance limit scales linearly with ultimate tensile strength, the endurance limit of the replacement crankshaft was increased to reflect the difference in ultimate tensile strength between the failed and replacement crankshaft. (Johnston, ff Tr. 22,610 at 36).

70. The original 13-inch by 11-inch crankshaft on EDG 101 was instrumented with strain gauges in the fillet location of crankpin No. 5. This fillet had previously experienced a fatigue crack during testing. After the test, the three-dimensional finite element model of a quarter section of a crank throw showed that the strain gauges were placed close to the location of maximum stress. The measured stress range was used to establish the endurance limit in this analysis as a conservative assumption, although the actual maximum stress range was revealed by the finite element model to be about 15% higher at a nearby location. The original crankshaft on EDG 102 had experienced 273 hours at equal to or greater than 100% load, or about 4,000,000 cycles. By using linear cumulative damage techniques, FaAA determined that the endurance limit for the original crankshafts was 36.5 ksi. (Johnston, ff Tr. 22,610 at 37).

71. The fatigue endurance for the replacement crankshafts is 39.2 ksi. This is higher than the fatigue endurance limit for the original crankshafts because the ultimate tensile

strength of the replacement crankshafts exceeds the ultimate tensile strength of the original crankshafts, and the fatigue endurance limit is directly proportional to ultimate tensile strength. (Johnston, ff Tr. 22,610 at 37).

72. FaAA calculated the factor of safety against fatigue failure by plotting the amplitude of equivalent stress (24.6 ksi) on a Goodman diagram constructed using the fatigue endurance limit and the ultimate tensile strength values for the replacement crankshafts. The factor of safety against fatigue failure is 1.48, without taking into account any beneficial effect of shot peening the fillet regions. (Johnston, ff Tr. 22,610 at 38; LILCO Diesel Exhibit C-17 at 3-32).

73. A factor of safety of 1.48 provides sufficient assurance that the replacement crankshafts are adequate for their intended service in the Shoreham EDGs. (McCarthy, ff 22,610 at 38).

74. A factor of safety is an additional margin of strength, in either the fatigue strength (endurance limit), yield strength, or ultimate strength, that is added to a mechanical design to compensate for uncertainties. There is significant confusion often generated by a failure to identify whether a stated factor of safety is with regard to fatigue or endurance limit, yield, or ultimate strength. The factor of safety with regard to these three different failure modes will

generally be different for the same design or part. (McCarthy, ff Tr. 22,610 at 38-39).

75. A factor of safety in endurance limit is the factor of strength the part or design has over that required for the part to be expected to exhibit infinite life, or a life of some specified number of cycles in repeated or cyclic loading. A factor of safety in yield is the factor the yield strength of the part is greater than the expected service load. Similarly the factor of safety in ultimate strength or overload failure is the factor the breaking strength of the part is greater than the expected service load. In older design references it is not uncommon to see a very large factor of safety in overload recommended, and no mention of a factor of safety in endurance limit or fatigue strength, for parts that were cyclically loaded and could fail in fatigue. This was before fatigue and stress concentration effects were as well understood as they are now. (McCarthy, ff Tr. 22,610 at 39).

76. A factor of safety is an allowance for uncertainties as to service load, material properties, stress concentration factors, lifetime, etc., which are directly related to the amount of testing, analysis, and understanding a designer has of a particular part and its service environment. (McCarthy, ff Tr. 22,610 at 39-40).

77. An acceptable factor of safety is determined by the degree of uncertainty and the difficulty or penalties of adding additional strength to the design. Where the design envelope and the nature of the fabricated part are reasonably understood, a factor of safety in fatigue or cyclic loading of 1.3 to 2.0 is generally recommended. When the uncertainty of design factors is greater, higher values will be recommended. Some design texts will recommend that, if the designer is seriously considering a factor of safety of greater than two, he should devote additional time to analyzing the design, rather than accepting the ignorance which is causing him to select a higher factor of safety. A factor of safety of 1.48 in fatigue or endurance limit will produce a much higher factor of safety with regard to yielding or overload failure. (McCarthy, ff Tr. 22,610 at 40).

78. The design of the replacement crankshafts is understood extremely well. Information has been gained from the failure of the original crankshafts, full scale instrumented tests of the actual service loading, material strength tests for the individual parts, torsiograph testing, and extensive three dimensional analytical modeling of the structure. The crankshaft is being run in a temperature controlled, oil filled environment. It is completely guarded from accidental and unanticipated impact by foreign objects by the engine block.

Usually a designer has far less information to work with when assessing a design. This results in uncertainities in the design being reduced substantially. (McCarthy, ff Tr. 22,610 at 41).

79. For well understood designs operating in environments that are not severe, a factor of safety in fatigue or endurance limit of 1.3 to 1.5 is generally accepted. For the replacement crankshafts the degree of understanding permits the use of a safety factor at the lower end of this range, when in fact the actual safety factor is at the high end. (McCarthy, ff Tr. 22610 at 41).

80. The factor of safety of 1.48 is more than acceptable because of the extensive knowledge concerning the design of the replacement crankshafts. All three of the original crankshafts failed at the point they were predicted to fail by FaAA's analytical model. In addition to the analytical model, there is a dynamic model, which permits prediction of the vibrations and deflections of the moving crankshaft. The dynamic model has been verified by torsiograph measurements on the crankshaft. There is a finite element model of the crank throw for both the old and new crankshaft, which has been verified by measurements on the old and new crankshafts in operation. In addition, the analytical model predicts the original crankshaft failure and the replacement crankshaft survival by a wide margin. (Tr. 23,027-28 McCarthy).

81. Factors of safety are based on a comparison of the knowledge of the design and the uncertainty about the expected service. In the case of the replacement crankshafts, the margin of safety is 1.48 and there is extremely detailed knowledge about the expected service. This knowledge allows a confident conclusion that the crankshafts are capable of unlimited operation at 3500 KW and 3900 KW. (Tr. 23,030-31 McCarthy).

VI. Kritzer-Stahl

82. The Kritzer-Stahl design criteria is a method for evaluating the adequacy of a crankshaft by calculating a factor of safety. The method compares calculated stresses with calculated endurance limits, which permits the calculation of a factor of safety. The crankshaft design and engine operating conditions are used as inputs to calculate the stress levels. (Tr. 22,767 Pischinger).

83. The original research on which the Kritzer-Stahl criteria is based was performed prior to 1961. The criteria has been updated periodically. Additional research that has been conducted since 1961 indicates that the Kritzer-Stahl criteria is very conservative. For example, the stress concentration factors calculated by the more recent Lejkin method are lower than the stress concentration factors calculated by the Kritzer-Stahl method. (Tr. 22,769, 22,772, 22,775 Pischinger).

84. There is a built-in factor of safety in the Kritzer-Stahl criteria of approximately 22%. This is demonstrated by the fact that the Kritzer-Stahl criteria predicts that the original 13-inch by 11-inch crankshaft should have failed after 150 hours at full load (about 2,000,000 cycles). In fact, the original crankshafts did not fail until 273 hours of operation at full load (about 4,000,000 cycles). The original crankshafts actually had twice the life predicted by Kritzer-Stahl. (Tr. 22,776-77 Pischinger).

85. The calculated endurance limit for the replacement crankshafts under the Kritzer-Stahl criteria is 25.4 ksi. The calculated maximum stresses in the fillet radius using the Kritzer-Stahl criteria is 24.9 ksi. (Tr. 22,790-94 Pischinger).

86. There is a very close correlation between the Kritzer-Stahl predicted maximum stresses (24.9 ksi) and the maximum stresses measured during dynamic testing (24.6 ksi). (LILCO Diesel Exhibit C-17 at 3-9). The endurance limit calculated by the Kritzer-Stahl method (25.4 ksi) is extremely conservative when compared to the actual endurance limit determined by FaAA (39.2 ksi). (Johnston, ff Tr. 22,610 at 37).

87. The ratio of the endurance limit calculated by the Kritzer-Stahl method to the maximum stress calculated by the Kritzer-Stahl method is 1.02. Combining that margin with the

built-in 22% margin in Kritzer-Stahl provides a factor of safety of 24% at full load under the Kritzer-Stahl criteria. (Tr. 23,004 Pischinger).

88. The Kritzer-Stahl criteria is highly accurate for predicting maximum stresses. The criteria predicts extremely conservative endurance limits, however. This is demonstrated by the fact that the original crankshafts had twice the life predicted by Kritzer-Stahl. This is the main conservatism of the Kritzer-Stahl criteria. (Tr. 23,006-07, 23,045-46, 22,776-77 Pischinger).

89. The range of acceptable factors of safety in contemporary European diesel industry practice is from 1.15 (15%) to 1.30 (30%). What is acceptable for a factor of safety depends on how much information is available about the crankshaft. If there have been actual test measurements from the crankshaft and there is information from previously failed crankshafts, a lower factor of safety is acceptable. The less information that is available, the higher the factor of safety should be. (Tr. 23,012-13 Pischinger).

90. The replacement crankshafts have a 24% margin of safety at full load and are capable of unlimited operation at 3500 KW. In addition, the replacement crankshafts are capable of operating for at least 1200 hours at 3900 KW. The 1200 hour figure does not include any allowance for the inherent safety

factor in Kritzer-Stahl of 22%. If the inherent conservatism is considered, the crankshafts have a safety margin of 15% for operation at 3900 KW. (Pischinger, ff Tr. 22,610 at 5; Tr. 22,792-93, 23,037-38 Pischinger).

91. The crankshafts will never operate at 3900 KW during a LOOP LOCA. The engines are expected to operate for no more than 60 hours at 3900 KW during testing over the 40 year life of the plant. This is only one twentieth (1/20) of the crankshafts' minimum life at 3900 KW. (Youngling, ff Tr. 22,610 at 5).

VII. <u>Shot Peening</u>

92. Shotpeening is a surface cold-working process that is used primarily to lengthen fatigue life and prevent cracking of metal parts. Shotpeening is also used to shape parts, overcome porosity, work harden surfaces, protect against stress corrosion or corrosion fatigue and for many other purposes. A crack will not initiate in, nor propagate through a compressed layer. As nearly all fatigue, stress corrosion and corrosion fatigue failures originate at the surface of a part, the layer of compressive stress induced by shotpeening produces a significant increase in the endurance limit. The maximum compressive residual stress produced at or near the surface is at least as great as one-half (1/2) the ultimate tensile strength of the

material. Shotpeening is used to eliminate failures in existing designs, or to allow the use of higher stress levels. (Cimino et al, ff Tr. 23,122 at 6-7).3/

93. The replacement crankshaft fillet areas were shotpeened to reduce the mean surface tensile stresses and place the fillet surfaces in compression. Shotpeening makes the surface less susceptible to handling damage, such as the score mark where cracking initiated on the original crankshaft in EDG 102. Additionally, shotpeening eliminates machine imperfections, which prevents initiation of cracks on the machined fillet surface. Shotpeening provides a higher endurance limit for the fillet area and the crankshaft. (Wells, Seaman, ff Tr. 23,122 at 5-6).

94. Two of the replacement crankshafts were initially shotpeened by TDI. Examination revealed that the shotpeening was inadequate. There were areas where coverage was only 80% to 90% and not all peening intensity tests (Almen strips) were accounted for. There was no concern that the TDI shotpeening had damaged the crankshafts. (Wells, Seaman ff Tr. 23,122 at 7).

3/ The prefiled testimony of Messrs. Wells, Johnson, Wachob, Seaman, Cimino and Burrell was not numbered when bound into the transcript. The testimony may be found following page 23,122 of the transcript.

95. Metal Improvements Company (MIC) was hired to re-peen the crankshafts. The crankshafts were placed on pedestals or stands which allowed rotation of the crankshafts so that all fillet areas could be completely saturated with shot. The crankshafts were washed with a chemical solution to remove all traces of oil or other preservatives and the areas on both sides of the fillets were taped. A tent was set up over each of the crankshafts so that shot could be contained within the tent. In addition, Almen strips were set up for measuring shotpeening intensity. Almen strips are flat pieces of metal which are clamped to a solid block and exposed to a stream of shot. Upon removal from the block the Almen strip will be curved. The curvature will be convex on the peened side and the height of the curved arc is measured on a special Almen gauge which serves as a measure of the intensity. A .008-.010 C strip was utilized for the Shoreham replacement crankshafts which provides surface compression to a depth of .027"-.034" on ASTM A-668E metal such as the replacement crankshafts. While MIL Spec. No. 13165B required intensity to be checked by Almen strips every eight hours of peening, MIC, in fact, checked peening intensity every four hours of actual peening. In addition, the shot was screened and examined under a microscope every two hours to ensure uniformity of shot size and shape. (Cimino, ff Tr. 23,122 at 8-9).

96. MIC utilized a patented process called "peenscan," approved by USA Military Specification, MIL - 13165-B, Amendment 2, to ensure uniformity and full coverage on the area being shotpeened. The area being shotpeened is coated with a flourescent dye-type liquid prior to the shotpeening and allowed to dry. All areas covered with dye will show a green glow under a blacklight. After shotpeening is completed the area is placed under blacklight to see if any green glow remains. If any glow remains the coverage is not 100%. In this case all fillet areas were checked for any green glow and peened until all traces of the dye were completely gone. (Cimino, ff Tr. 23,122 at 10).

97. The shotpeening MIC performed on the two replacement crankshafts was in accordance with MIL Spec. No. 13165B and placed the surface stresses in the fillet area of the crankshaft in compression. (Cimino, ff Tr. 23,122 at 11).

98. The two crankshafts shotpeened by TDI were subjected to magnetic particle testing after machining by the manufacturer and no relevant indications were found. In addition, when the two crankshafts were received at Shoreham, both shafts were subjected to visual examination, magnetic particle testing and liquid penetrant testing. This examination and testing revealed no relevant surface cracks or indications. (Wells, Seaman, ff Tr. 23,122 at 12; Bush, ff Tr. 23,126 at 19). Thus,

the as-received surface condition of the replacement crankshafts shotpeened by TDI was acceptable. (Bush ff Tr. 23,126 at 19).

99. The photographs of the re-peened fillet areas that were reviewed by Franklin Research Center and referred to in its report dated April 6, 1984 are representative of all crankpin and main journal fillet shotpeening. As a result of MIC's work, the peening is uniform, equally dimpled, and the shotpeening at all fillet areas looks exactly as it does in these photographs. (Wells, Seamen, ff Tr. 23,122 at 12).

100. The nondestructive examination of the TDI-peened fillet areas revealed no surface indications or deficiencies which could reasonably be expected to cause a "stress nucleation site." Even if there had been surface "stress nucleation sites," proper repeening of the fillet areas would correct or eliminate any such problem. (Burrell, ff Tr. 23,122 at 13).

101. The re-peening by MIC would have corrected or eliminated any "stress nucleation sites" that may have existed rather than masking them. Any surface "stress nucleation site" small enough to escape detection by magnetic particle testing and/or liquid penetrant testing would be eliminated as a result of the plastic flow of the surface metal caused by the re-peening. (Wells, ff Tr. 23,122 at 13-14).

102. The surface stresses in the fillet areas of the Shoreham replacement crankshafts have been placed in compression and any cut, scratch, flaw, machine mark, etc. no deeper than the compression area itself, will not be the initiation point of a fatigue crack. Any undesirable effects of the previous shotpeening have been corrected. (Burrell, Wells, Wachob, ff Tr. 23,122 at 14-15).

103. The possibility that shotpeening increased the likelihood of subsurface fatigue cracking is quite remote. In almost all instances, fatigue cracks such as occurred in the original crankshafts initiate at external surface areas. Subsurface fatigue cracking is very unusual and requires the presence of a significant void or inclusion at a given stress state for initiation of the fatigue crack. There is always the possibility that any cast or forged piece of metal may contain a subsurface inclusion or void. The only protection against this risk or possibility is the manufacturer's quality control procedures for the melting, casting and forging processes and its quality assurance procedures during and after the manufacturing process. The replacement crankshafts for the EDG's were manufactured by Krupp, and the forging and machining of these crankshafts were certified by ABS. Additionally, Krupp's ultrasonic testing and magnetic particle testing, as well as LILCO's ultrasonic testing, magnetic particle testing

and liquid penetrant testing, revealed no relevant inclusions or voids. This provides as much assurance as is possible that no subsurface voids or inclusions of sufficient size to initiate a subsurface fatigue crack are present in the replacement crankshafts. (Burrell, Wells, Wachob, ff Tr. 23,122 at 15-16). Subsurface fatigue cracking as a result of shotpeening would not occur unless a large embedded flaw was present. Such a flaw would have been detected by the extensive ultrasonic testing during fabrication. (Bush, ff Tr. 23,126 at 19-20).

104. In addition, after 300 hours of operation, of which 100 hours were at 3500 KW or above, the eight crankpin fillet areas of highest torsional stress on each of the three crankshafts were subjected to high resolution eddy current testing. The eddy current test was designed to detect cracks larger than 1/32" long by 1/64" deep. No cracks were found. In addition, the eight crankpin fillet areas of highest torsional stress were subjected to liquid penetrant testing after 300 hours of operation. No relevant indications were found. (Johnson, Wells, Seaman ff Tr. 23,122 at 19).

105. The crankshafts were subjected to more than one million torsional peak stress reversals during this period of operation. Any "stress nucleation site" that had not been detected by previous nondestructive testing would have initiated a fatigue crack that would have been detected by the high

resolution eddy current testing and/or liquid penetrant testing. (Wells, ff Tr. 23,122 at 19-20).

106. The benefits of shotpeening are attributed to the residual compressive surface stress. This region, although small in respect to the crankshaft diameter, is significant with regard to preventing the initiation of a fatigue crack in the surface. Given the residual compressive stresses and the actual operating stresses in the fillet region, a fatigue crack will neither initiate in the fillet area nor will any flaw or defect contained within the shotpeened volume propagate. There is no equation between the hardened depth of shotpeening and its effective depth. (Burrell, Wells, Wachob, ff Tr. 23,122 at 20-21).

107. The cathode-anode electrochemical principal is not operative upon the replacement crankshafts in the Shoreham EDG's because it requires a driving energy that is not present, the presence of electrolytes, which do not exist within the crankcase of the Shoreham diesels. (Burrell, Wells, Wachob, ff Tr. 23,122 at 18-19; Tr. 23,182-83 Wells, Wachob).

108. In order for heat to appreciably affect residual stresses caused by shotpeening, temperature levels of at least 500° F must be attained. This temperature is completely unattainable within the normal operating limits of the Shoreham diesels. The crankshaft temperature is between 200° - 240° F

under normal operating conditions. Under unusual circumstances the temperature may go as high as 260° F. (Burrell, Wells, Wachob, ff Tr. 23,122 at 21).

109. The shotpeening of the replacement crankshaft fillet areas has resulted in increasing their fatigue endurance limits by a minimum of ten percent (10%) and could conceivably have increased the fatigue endurance limits by as much as twenty percent (20%). (Burrell, ff Tr. 23,122 at 22; Tr. 23,158 Wells).

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DATED: November 5, 1984

LILCO, November 5, 1984

CERTIFICATE OF SERVICE

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

I hereby certify that copies of LILCO's PROPOSED FINDINGS OF FACT were served this date upon the following by first-class mail, postage prepaid, or (as indicated by asterisk) by hand.

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