LILCO, August 14, 1984

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)	)	

TESTIMONY OF ROGER L. McCARTHY, PAUL R. JOHNSTON, EUGENE F. MONTGOMERY AND SIMON K. CHEN ON BEHALF OF LONG ISLAND LIGHTING COMPANY ON SUFFOLK COUNTY'S CONTENTION REGARDING REPLACEMENT CRANKSHAFTS ON DIESEL GENERATORS AT SHOREHAM

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I. INTRODUCTION OF WITNESSES

1. Please state your names, business affiliations and addresses.

A. (McCarthy) My name is Dr. Roger L. McCarthy and I am employed by Failure Analysis Associates as president and chief executive officer. My business address is 2225 East Bayshore Road, Palo Alto, California, 94303.

(Johnston) My name is Dr. Paul R. Johnston. I am employed by Failure Analysis Associates as manager of the structural analysis group. My business address is 2225 East Bayshore Road, Palo Alto, California, 94303.

(Montgomery) My name is Eugene F. Montgomery. I am employed by Long Island Lighting Company as a stress analyst. My business address is Shoreham Nuclear Power Station, Long Island Lighting Company, Wading River, New York.

(Chen) My name is Dr. Simon K. Chen. I am a professional engineer registered in the State of Wisconsin and the owner and president of Power and Energy International, Inc., a private consulting firm. My business address is 555 Lawton Ave., Beloit, Wisconsin, 53511.

2. Please summarize your professional qualifications and your role in evaluating the replacement crankshafts at Shoreham.

A. (McCarthy) I am principal design engineer for FaAA

and hold five degrees, including a Ph.D. in mechanical engineering from M.I.T. My specialty is mechanical design. My resume is Attachment 1.

My role in evaluating the replacement crankshafts at Shoreham has been to personally inspect the broken crankshafts and the replacement crankshafts, to perform the final review of the FaAA reports and to oversee the corporate performance of FaAA's evaluation of the crankshafts.

(Johnston) I obtained my undergraduate degree in Civil Engineering (B.A.I.) in 1976 from Trinity College, Dublin, Ireland. Thereafter, I attended Stanford University where I received a M.S. in Structural Engineering in 1977 and a Ph.D. in Civil Engineering in 1981. I have worked for FaAA since 1978, principally in the analysis of failures in structures and machinery. From 1981 to 1983, I also served as a Consulting Assistant Professor at Stanford University, where I taught graduate courses in finite elements and structural dynamics. I am co-author of the book Finite Elements for Structural Analysis. My resume is Attachment 2.

My role in evaluating the replacement crankshafts at Shoreham has been to evaluate the adequacy of the crankshafts by analysis and by using the results of dynamic tests on the original and replacement crankshafts.

(Montgomery) I received my undergraduate degrees in Mechanical Engineering (B.A., B.S.) in 1973 under a combined 3/2-year program at Queens College in the City University of New York and Columbia University. Thereafter, I attended Columbia University where I received an M.S. in Mechanical Engineering in 1974 and an M.E. (Professional Degree) in Mechanical Engineering in 1981. I have worked for LILCO since 1981, principally in the area of engineering mechanics for safety-related piping, equipment and support structures. From 1980 to 1981, I was a senior engineer in the Piping Stress Analysis Department of Burns & Roe, Inc., Woodbury, N.Y. Prior to that time, I was employed as a senior engineer in the Stress Analysis Department of Ebasco Services, Inc., Jericho, N.Y. from 1978 to 1980. My resume is Attachment 3.

My role in evaluating the replacement crankshafts at Shoreham has been to serve as LILCO's engineering specialist providing technical review and direction to the work performed by LILCO's consultants: Failure Analysis Associates, Stone and Webster Engineering Corporation, and Power and Energy International.

(Chen) I received my undergraduate degree in mechanical engineering (B.S.M.E.) in 1947 from National Chiao Tung University. In 1949 I received a masters degree in mechanical engineering (M.S.M.E.) from the University of Michigan, and in 1952

I received a Ph.D. in mechanical engineering from the University of Wisconsin. I also received an M.B.A. from the University of Chicago in 1964. For the past four and one-half years I have been the owner and president of Power and Energy International, Inc. (PEI), a private consulting firm. Prior to forming PEI, I was president and chief technical officer of the Beloit Power System Division of Louis Allis Litton Industries from 1973 until 1979. From 1971 until 1973 I was vice-president of engineering and applications of the entire Fairbank Morse Power System Division. From 1969 until 1971, I was vice-president and general manager of the large engine division of the Fairbank Morse Power Systems Division of Colt Industry. From 1952 until 1969 I was employed by International Harvester. My first job was project engineer in charge of combustion development. My last job at International Harvester was divisional chief engineer in charge of all engine research and development. My resume is Attachment 4.

My role in evaluating the replacement crankshafts at Shoreham has been to perform a critical review of all analyses and testing of the crankshafts and to conduct an independent analysis of the adequacy of the crankshafts.

3. What issues have you been asked to address in your testimony?

A. (All) We have been asked to address Emergency Diesel

Generator Contention 1(a), admitted by the Board in its July 17, 1984 Memorandum and Order, which is whether:

> 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 Registry 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 FEV.

In summary, this testimony demonstrates that the replacement crankshafts are suitable for unlimited operation in the emergency diesel generators at Shoreham. The structural integrity of the replacement crankshafts has been extensively evaluated by testing, analysis and inspections. There is no requirement that the crankshafts comply with the design standards of the American Bureau of Shipping, Lloyd's Registry of Shipping, the International Association of Classification Societies or FEV's criteria. Therefore, compliance with the design criteria of one or more of the above organizations is not necessary to demonstrate the crankshafts are adequate for their intended service at Shoreham. Furthermore, ABS has approved the torsional critical speed arrangement of the crankshaft.

The crankshafts are required to comply only with the recommendations of the Diesel Engine Manufacturers Association (DEMA). Conventional analytical techniques typically utilized

by the diesel engine industry show that the 13-inch by 12-inch replacement crankshafts comply with DEMA recommendations. Angular displacements of the free end of the crankshaft, stress ranges in the most highly stressed crankpin fillets, and the range of output torque at the flywheel were measured at and above full-rated load. The torsiograph measurements of twist confirm the analyses and show that the crankshafts meet the DEMA recommendations.

In addition, strain gage measurements of maximum bending and torsional stress and calculations of maximum stress by a modal superposition analysis show that the crankshafts have a factor of safety in fatigue of 1.48, without taking into account any benefit of shot peening the crankpin fillets. This factor of safety is more than adequate to assure that the crankshafts will not fail in fatigue during operation. The factor of safety was determined from the measured endurance limit of the original 13-inch by 11-inch crankshafts that cracked in high cycle fatigue. The measured crankshaft response was in close agreement with that predicted by the modal superposition analysis. There is, therefore, more than adequate assurance that the crankshafts are suitable for their intended service.

II. BACKGROUND

4. Please briefly describe the function of the crankshaft in the diesel generators at Shoreham.

A. (All) The crankshaft converts the reciprocating (up and down) 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, i.e., twisting force. The output torque from the crankshaft drives the electrical generator to provide emergency power.

5. Please briefly describe the failure of the original 13-inch by 11-inch crankshafts at Shoreham.

A. (Montgomery) On August 12, 1983, the original 13-inch by 11-inch crankshaft on EDG 102 fractured through the crankpin and rear (generator end) web under cylinder No. 7. Subsequent investigation revealed that the crankshaft on EDG 101 was significantly cracked at the No. 5 and No. 7 crankpins and the crankshaft on EDG 103 was cracked at the No. 6 crankpin.

6. What was the cause of the crankshaft failure?

A. (Johnston, McCarthy) Based upon extensive metallurgical examinations of the fracture surfaces, the cause of the crankshaft failure was determined to be high cycle vibratory fatigue.

7. What caused the crankshafts to fail in high cycle fatigue?

A. (Johnston, McCarthy) The crankshafts failed in high cycle fatigue due to the torsional (or twisting) stresses imposed upon them during operation. Testing and analysis revealed that the crankshafts experienced torsional excursions beyond their fatigue endurance limit, which ultimately led to their failure.

8. What action did LILCO take after the failure of the original crankshafts?

A. (Montgomery) LILCO did a number of things. First, Failure Analysis Associates (FaAA) was hired to determine the cause of the original crankshaft failure FaAA's evaluation of the original crankshafts included: (1) a metallurgical failure analysis; (2) dynamic tests performed on the crankshaft from EDG 101; (3) a review of Transamerica DeLaval Inc.'s (TDI) torsional analysis of the Shoreham crankshafts; (4) a modal superposition analysis of the torsional system; and, (5) the development of a model employing finite element analysis to predict stresses imposed on the crankshafts during operation.

Second, after consulting with FaAA and TDI, LILCO ordered replacement crankshafts from TDI of a different design than the original crankshafts. The original crankshafts had a 13-inch main journal and an 11-inch crankpin. The replacement crankshafts have a 13-inch main journal and an 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. Typical structural dimensions of one throw and fillet details are shown in Exhibit C-1. In addition, the fillet regions of the replacement crankshafts have been shot peened. The average ultimate tensile strength of the original crankshafts was approximately 93,500 psi. The minimum ultimate tensile strength of the new crankshafts is over 100,000 psi. The replacement crankshafts have greater section properties, greater material strength and a more enhanced surface treatment (shot peening) than the original crankshafts.

Third, LILCO embarked on an unprecedented program to test and analyze the replacement crankshafts. This program was designed to ensure that the replacement crankshafts are adequately designed to withstand the stresses they will experience during operation in the Shoreham EDGs. This program included: (1) a detailed multi-modal, multi-frequency torsional dynamic analysis of the crankshaft; (2) finite element structural modeling and stress analysis of a single quarter crank throw geometry; (3) field tests on the EDG 103 replacement crankshaft at various power levels to measure the principal stresses in the fillet region of the crankshafts, torsional vibrations (torsiograph tests), cylinder pressure time diagrams, electrical generator output, and transient conditions due to engine start-up and generator load changes; (4) non-destructive

examination (eddy current tests) of the crankpin fillets on all three crankshafts at cylinder Nos. 5 - 8 after 100 hours of operation at 100% load or greater; and (5) review of the TDI torsional analysis using conventional Holzer and equivalent static equilibrium amplitude techniques.

III. DESIGN REQUIREMENTS

A. The Crankshafts Must Comply with DEMA

9. What were the design requirements for the replacement crankshafts?

A. (Montgomery) The replacement crankshafts were required to meet the recommendations of the Diesel Engine Manufacturers Association (DEMA). Stone & Webster's Specification for Diesel Generator Sets, Spec. No. SH1-89, Revision 2, January 26, 1983 (Spec. SH1-89) required that:

> The diesel engines and auxiliaries shall be designed, engineered, manufactured, and tested in accordance with the latest published applicable sections of the Standards of the Diesel Engine Manufacturers Association (DEMA), at least, but not limited to DEMA "Standard Practices for Low and Medium Speed Stationary Diesel Engines."

The relevant portion of Spec. SH1-89 is attached as Exhibit C-2.

10. Do the replacement crankshafts meet the DEMA recommendations?

A. (All) Yes. As will be discussed in detail later, the crankshafts meet the recommendations of DEMA, both for operation at full load (3500 KW) and at overload (3900 KW).

11. The County contends the replacement crankshafts are inadequately designed for operation at full load (3500 KW) or overload (3900 KW) because they do not meet the requirements of the American Bureau of Shipping (ABS), Lloyd's Registry of Shipping (Lloyd's), or the International Association of Classification Societies (IACS). In addition, under the German criteria used by FEV, the crankshafts are marginal at full load and inadequate at overload. Is there any basis for this contention?

A. (Montgomery) No. There is no licensing requirement, either in the Shoreham FSAR or in any applicable Nuclear Regulatory Commission regulation or guideline, that the replacement crankshafts meet any of these criteria. In fact, the only standby diesel generator design criteria currently referred to in an NRC Regulatory Guide is DEMA.

12. Please explain.

A. (Montgomery) NRC Regulatory Guide 1.9, Revision 2 (December 1979) (Exhibit C-3), addresses the design of standby diesel generator units at nuclear power plants. The Regulatory Guide provides:

> Conformance with the requirements of IEEE Std 387-1977, "IEEE Standard Criteria for Diesel-Generator Units Applied as Standby Power Supplies for Nuclear Power Generating Stations," dated June 17, 1977, is acceptable for meeting the requirements of the principal design criteria and qualification testing of diesel-generator units used as onsite electric power systems for nuclear power plants. . . .

IEEE Std 387-1977 (Exhibit C-4), provides:

> 4.1 Standards. The equipment and accessories of the diesel-generator unit shall conform to the applicable portion of the following standards and the latest revisions thereof, as of the date of approval of this document.

* * *

[5] DEMA, Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines.

Nowhere is there any requirement that the crankshafts meet the criteria established by ABS, Lloyd's, IACS or FEV. As Dr. Carl Berlinger, NRC Lead Engineer for the Assessment of Diesel Engine Reliability/Operability, stated at the July 11, 1984 meeting of the TDI Owners Group:

> NRC does not require the use of Lloyd's and specifically references DEMA, and we would not propose to require that this design be compared to Lloyd's. I don't know whether we really need any additional discussion relative to what standard to use as a basis for licensing or approval of these crankshafts.

The relevant portion of the transcript is attached as Exhibit C-5.

Furthermore, the determination of the fatigue endurance limit of the crankshafts, independent of any code or design requirements, establishes that the replacement crankshafts are adequate for their intended service.

B. The Crankshafts Do Not Have to Comply with ABS, Lloyd's, IACS or the Criteria Used by F.E.V.

13. Notwithstanding that there is no licensing requirement that the crankshafts meet any of these design criteria, is it necessary for the crankshafts to meet the standards of ABS, Lloyd's, IACS or the criteria used by FEV to be considered adequate and reliable for their intended use in the Shoreham EDGs?

A. (Montgomery, Chen) No. The replacement crankshafts have been demonstrated to be adequate and reliable by an extensive program of testing and analysis. This program clearly establishes, apart from any code, that the crankshafts will perform their intended function.

In addition, there is extensive experience with 13-inch by 12-inch crankshafts in DSR-48 engines that establishes the crankshafts are reliable. A table showing the operating history of DSR-48 engines with 13-inch by 12-inch crankshafts is attached as Exhibit C-6. An additional table showing the operating history of each of the Shoreham engines is attached as Exhibit C-7. The crankshafts were all inspected after 100 hours of operation at full load or greater by eddy current inspection. This inspection revealed no relevant indications or crack formations on the crankshafts after more than one million torsional peak stress reversals. The results of the eddy current inspection are attached as Exhibit C-8. Finally, the crankshafts comply with the DEMA recommendations for torsional vibratory stresses.

14. The County contends DEMA is not a design code and that it should not be used to determine the adequacy of the crankshafts. Do you agree?

A. (Chen) I agree that DEMA is not a design code. That is to say, DEMA does not tell an engine manufacturer how to design a crankshaft. However, I do not agree that DEMA does not

provide standards to measure the adequacy of a crankshaft. DEMA provides specific stress limits for crankshafts: 5,000 psi for a single order of vibration and 7,000 psi for the summation of the major orders. Engine manufacturers have used DEMA for years on stationary diesel generator installations to determine whether a crankshaft is adequate for its intended service. In addition, in over thirty (30) years of experience with diesel engines, I have never seen a crankshaft that complied with DEMA fail primarily from torsional fatigue.

15. The County states at page 114 of its testimony that "at a minimum, the crankshafts should be compatible with the rules of all the major classification societies." Do you agree with this statement?

A. (Chen) No. In fact, this statement is absurd. No reasonable person would say that a crankshaft had to comply with the rules of all major societies to be considered adequate. The rules, standards and design methodologies of design 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. Furthermore, certain of the codes explicitly recognize that special consideration should be given to detailed stress analyses and test data if a crankshaft does not comply with literal

code requirements. For example, Section 37.17.1 of the 1983 ABS rules on the diameter of pins and journals (Exhibit C-9) provides:

> Where critical dimensions are proposed which are less than those determined by the above equation, complete supporting data, including detailed stress analysis, are to be submitted for special consideration.

In addition, note 3 to Table 34.3 of the 1983 ABS rules concerning Allowable Stress Values for Crankshafts and Tail Shafts Due to a Single Harmonic (Grade 2 Steel) (Exhibit C-10) provides:

> If torsional critical speed arrangements are similar to previous installations proven by service experience, consideration will be given to higher stresses upon submittal of full details.

In sum, the best way to evaluate a crankshaft is through engineering analysis. The County's suggestion that the crankshafts should comply with selected aspects of various codes (i.e., the most conservative part of each code) has no foundation.

16. Is a crankshaft inadequate if it does not comply with ABS, Lloyd's, IACS or the criteria used by FEV?

A. (Chen) No. A crankshaft may be structurally adequate for its intended service and not comply with ABS, Lloyd's, IACS or the FEV criteria. 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 (such as testing or detailed engineering analysis) that the crankshaft is adequate, noncompliance is not significant.

Furthermore, the critical surface temperature and various stress levels of an operating marine engine vary considerably depending upon ship hull design, swells, wind and other sea-ship interactions, as well as the type of fuel used. That is why the marine engine classification rules are more stringent than the rules for stationary land-based engines. A stationary engine, which is perfectly adequate might or might not pass one or more of the marine codes.

17. What is the most accurate way to assess the adequacy of a crankshaft?

(A) (All) The most accurate way to assess crankshaft adequacy is not to rely upon the design criteria of any code. Rather, the most accurate way to assess crankshaft reliabilty is to perform the type of tests and analyses that were performed on the Shoreham crankshafts. This information permits the calculation of actual operating stress states, separate and apart from compliance with the standards of any code.

18. You have just described the most accurate way to

assess the adequacy of a crankshaft. Why are not all crankshafts assessed in this manner?

A. (All) Most crankshafts are not assessed in this manner because the design review normally occurs before the crankshaft is manufactured. This is where design codes are used. It is normally impossible to measure the actual stresses from tests on the crankshaft because the crankshaft does not exist when it is being designed. Because of the uncertainty in predicted loads and response, these design codes are very conservative.

Unfortunately, LILCO had the luxury of having data available from a smaller crankshaft that failed in the same engines. This allowed calculation of the fatigue endurance limit for the replacement crankshafts. This type of data is extremely useful, but it is normally unavailable. In the absence of this detailed information, design codes are relied upon to provide assurance of crankshaft adequacy.

19. Notwithstanding that the crankshaft is not required to meet any of these codes, has the crankshaft been approved by any of these ship classification societies?

A. (Montgomery) Yes. ABS has approved the crankshaft dimensional sizing for diameter of pins and journals and proportions of the crankshaft webs. A copy of the crankshaft drawing certified by ABS is Exhibit C-11. ABS has certified that the material properties of the replacment crankshafts conform to the requirements of ABS grade 4 specifications. A copy

of the material properties certification is Exhibit C-12. Finally, ABS has stated that it would approve the torsional critical speed arrangement of the crankshaft, flywheel and generator at Shoreham for use on an ocean going vessel. A copy of ABS's letter of approval is Exhibit C-13.

20. The County contends ABS's approval is suspect because the information submitted to ABS was deficient in four specific areas: (1) shot peening; (2) maximum firing pressures; (3) strain gage measurement; and (4) operating experience. Please respond to each of these areas.

A. (Montgomery) The County claims the information on shot peening was inaccurate because TDI took credit for a 20% increase in the fatigue limit and there was no discussion of the first shot peening by TDI. As the separate testimony of Messrs. Wells, D. Johnson, Wachob, Seaman, Cimino and Burrell clearly demonstrates, the shot peening does increase the fatigue limit by up to 20%.

21. The County contends that maximum firing pressures as high as 1750 psi have been measured at full load. ABS was informed that the maximum firing pressure at full load was 1700 psi. Please discuss.

A. (Montgomery) The County is simply wrong. The documents relied upon by the County to show that peak firing pressures of 1750 psi have been measured at full load (TDI test logs attached to Suffolk County Exhibit 46) clearly show that the pressures above 1700 psi were measured at 110% of full load. The maximum firing pressure of 1700 psi relied upon by

ABS is correct. A fuller discussion of the inaccuracy of the County's contention concerning maximum firing pressure is contained in the testimony of Messrs. Harris, et al., on pistons.

22. The County contends TDI did not inform ABS that the strain gage test results were only accurate to within ± 5%. Is this significant?

A. (All) There is no significance to the fact that ABS was not informed that the strain gage test results were only accurate to within ± 5%. This is the expected degree of accuracy for field test results of this type.

23. Finally, the County contends TDI did not submit accurate information on the operating experience of the DSR-48 engines. Please discuss.

A. (Montgomery) The operating history submitted for the Shoreham engines was complete and accurate. The information submitted is attached as Exhibit C-6. This clearly shows the number of hours the Shoreham engines have operated at and above 3500 KW. In addition, there was no reason to submit information concerning block cracking since block data is not used in ABS's design rules for crankshafts. ABS was only asked to review the torsional critical speed arrangement. ABS was provided complete and accurate information for the Shoreham engines and approved the crankshafts on that basis.

IV. THE CRANKSHAFTS COMPLY WITH DEMA

24. Do the replacement crankshafts meet the recommendations of DEMA?

A. (Johnston, Chen) Yes, conventional analytical techniques typically utilized by the diesel engine industry show that the replacement crankshafts comply with the recommendations of DEMA.

25. What are the DEMA recommendations for crankshafts?

A. (Johnston, Chen) The DEMA recommendations for allowable crankshaft vibratory stress (Exhibit C-14) state:

> 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.

26. How did you determine that the crankshafts complied with DEMA?

A. (Johnston) In August, 1983, TDI performed a torsional critical speed analysis of the replacement crankshafts. (Exhibit C-15). FaAA reviewed this analysis for compliance with the DEMA allowable stresses. In addition, in January, 1984, Stone & Webster Engineering Corporation, conducted

torsiograph tests on a replacement crankshaft at Shoreham. (Exhibit C-16). FaAA compared the test results with the DEMA allowable stresses. Based upon the review of TDI's torsional analysis and Stone & Webster's torsiograph tests, FaAA conclud-ed the crankshafts complied with DEMA at full load (3500 KW) and overload (3900 KW). FaAA's conclusions are contained in the TDI Owners Group Crankshaft Report. (Exhibit C-17).

(Chen) In addition, I performed independent calculations (Exhibit C-18) to determine whether the crankshafts met the recommendations of DEMA. These calculations employed an inter-nationally known computer program (TORVAP), which is widely used by the diesel engine manufacturers industry to measure nominal crankshaft torsional stresses. On the basis of these independent calculations, I determined that the replacement crankshafts complied with DEMA at full load (3500 KW) and over-load (3900 KW).

27. What is a torsional critical speed analysis?

A. (Johnston, Chen) A torsional critical speed analysis is a method of calculating the torque being transmitted through a crankshaft in a diesel engine at a particular speed and power level. When operating at a particular speed and power level, the torque being transmitted through a crankshaft in a diesel engine varies with time and location. For a four-stroke en-gine, the torsional stress relationship over time repeats

itself every two revolutions of the crankshaft. The maximum torque on the crankshaft at any instant may be much larger than the mean torque required to run the engine at a given speed and power level. This additional torque is caused by a number of factors, including the cylinder firing order (excitation) and the presence of natural torsional modes of vibration of the crankshaft. To determine the maximum torque applied to the crankshaft, it is necessary to conduct a torsional critical speed analysis. Once the maximum torque has been calculated, it is simple to calculate the nominal torsional stresses for comparison to DEMA allowable stresses.

28. How was TDI's torsional critical speed analysis conducted?

A. (Johnston, Chen) TDI calculated the response of the crankshaft at 100% of rated load (3500 KW). The torsional analysis conducted by TDI was of two parts. First, TDI used an analytical technique, known as the Holzer method, to compute the natural frequencies and modes of vibration of the crankshaft system. If you strike a tuning fork, it will tend to vibrate at a particular frequency that is called its natural frequency. Similarly, a twisting force exerted on a crankshaft will induce the shaft to vibrate at certain discrete natural frequencies. The shape or angle of twist as a function of position along the shaft is unique for each natural frequency,

and this is often referred to as a mode shape. The Holzer method permits the manufacturer to calculate the predicted natural frequencies of the various modes of vibration that will result from torsional forces exerted on the crankshaft during operation.

TDI used the Holzer method to calculate the system's first three natural frequencies, which are shown in Exhibit C-19. In a four stroke engine such as the Shoreham diesel generators, operation at the fourth order critical speed produces the maximum stresses. The fourth order critical speed calculated by TDI is 581 rpm. The Shoreham engines operate at 450 rpm, which is significantly below the fourth order critical speed.

29. What is the second step of the analysis?

A. (Johnston, Chen) The second step in a torsional critical speed analysis is to determine the dynamic torsional response of the crankshaft due to gas pressure and reciprocating inertia loading for each order. The first order is a harmonic which repeats once per revolution of the crankshaft. For a four-stroke engine, harmonics of the order 0.5, 1.0, 1.5, 2.0, 2.5. . . exist. TDI performs this calculation separately for each order of vibration up to 12. For each order, the applied torque and nominal torsional stress at a cylinder due to gas pressure and reciprocating inertia is calculated.

30. What was the result of TDI's analysis and how did the result compare to DEMA allowables?

A. (Johnston) TDI calculated the response for the first three modes and plotted the results for only the first mode, since higher modes produce much smaller stresses. The nominal shear stresses for the significant orders are shown in Exhibit C-20. The largest single order stress at rated load and speed is for the fourth order. This stress, 2980 psi, is well below the 5000 psi allowed by DEMA. Due to the analytical technique TDI employed, TDI did not calculate the torsional stresses created by the summation of the major orders of vibration for purposes of comparison with the DEMA allowable of 7000 psi.

31. Given that TDI only calculated single order stresses, what further action was taken to assure that the crankshafts complied with DEMA?

A. (Johnston) Stone & Webster performed torsiograph tests on the replacement crankshaft in EDG 103 in January, 1984 at various power levels. (Exhibit C-16). The torsiograph tests measured the total torsional vibrations resulting from all orders. These torsional vibrations were converted into stresses for comparison with DEMA.

32. How is a torsiograph test performed?

A. (Johnston, Chen) A torsiograph test is performed by placing a seismic instrument (a device for measuring angular displacement due to vibration) on the end of a crankshaft and recording the angular displacement due to vibration under different engine operating conditions.

The test is usually performed in two stages. The first stage is without load and is used to determine the location of critical speeds, or natural frequencies, of the crankshaft. This is done by varying the speed of the engine and recording the vibratory response. As the frequency of vibration for any order approaches a natural frequency of the shaft, the amplitude of vibrations will increase and reach a peak at the natural frequency. If you know the engine speed where this peak vibration occurs, it is simple to calculate the natural frequency. Critical speeds may also be determined while operating at a fixed speed and observing the frequency content of the response.

33. How did the natural frequency measured by Stone & Webster compare to the natural frequency computed by TDI?

A. (Johnston) The frequency content of the torsional vibration signal at 450 rpm showed a resonance at 38.6 Hz. This value is in excellent agreement with TDI's computed value of 38.7 Hz. This comparison demonstrates that the mass elastic properties used in TDI's analysis for representation of the crankshaft are correct.

34. What is the second stage of the torsiograph test?

A. (Johnston, Chen) The second stage is to determine nominal stresses in the crankshaft under various load conditions. This test is performed at rated speed of 450 rpm with

variable load. The purpose of this test is to confirm the forced vibration calculations.

The torsiograph provides the angular displacement response (the angle of twist) 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.

35. How were the nominal stresses determined from the torsional vibrations measured by Stone & Webster?

A. (Johnston) Stone & Webster tabulated the single order and peak-to-peak torsional vibration response for both 3500 KW (100% of rated load) and for 3800 KW (109% of rated load). FaAA factored these values to obtain nominal shear stresses, which are shown in Exhibit C-21. The results at 100% load show that the largest single order (the fourth order) has a stress of 3108 psi, which is well below the DEMA allowable of 5000 psi. The total stress of 6626 psi is also below the DEMA allowable of 7000 psi.

At 3800 KW the stresses of 3242 psi for a single order and 6875 psi for combined response are also lower than 5000 psi and 7000 psi respectively. At 3900 KW the corresponding stresses are 3287 psi and 6958 psi, by linear extrapolation. The measured response at 3500 KW is in close agreement with that calculated by TDI.

36. Did FaAA calculate the stresses at 95% and 105% of rated speed?

A. (Johnston) Yes, we calculated the fourth order and total stresses at 95% and 105% of rated speed. On the basis of our calculations, we conclude that the stresses at those speeds satisfy the DEMA allowables.

37. What conclusions did FaAA draw from the stresses calculated from the torsiograph test data and the stresses calculated analytically by TDI?

A. (McCarthy, Johnston) The design calculations on the 13-inch by 12-inch crankshafts performed by TDI are appropriate and show that the crankshaft stresses are below DEMA recommendations for a single order. Combined stress was not calculated by this method, but was determined by torsiograph testing. The Stone & Webster torsiograph test results show that the 13-inch by 12-inch crankshaft stresses are below the DEMA recommended levels for both single order and combined orders for both 3500 KW (100% rated load) and 3800 KW. A linear extrapolation to 3900 KW also shows compliance. In addition, no harmful torsional vibratory stresses occur within 5% above and 5% below rated speed.

38. Dr. Chen, do your calculations also show that the replacement crankshafts comply with DEMA?

A. (Chen) Yes.

39. Please describe your calculations.

A. (Chen) I calculated the natural frequencies, as well as the torsional stresses of the engine generator system using the TORVAP R and TORVAP C computer programs. I calculated the response for single orders and combined orders. I also calculated the torsional vibration at the free end of the crankshaft. The calculations I performed are typical of the calculations performed by the diesel engine industry to check the adequacy of a crankshaft to withstand torsional stress.

40. What were the results of your natural frequency calculations?

A. (Chen) The natural frequency calculations are essentially identical to the natural frequency calculations of TDI and FaAA. The results are shown in the following table:

Mode	TDI	FaAA	PEI
1st	2323.2	2323.8	2323.3
2nd	5575.5	5576.4	5575.2
3rd	7000.3	7002.0	7000.4

41. What were the results of your free end amplitude calculations?

A. (Chen) The results of the free end amplitude calculations are in close agreement to the values calculated by FaAA and measured by Stone & Webster. The results for the fourth order and the combined response are shown in Exhibit C-22.

42. What were the results of your single order nominal stress calculations?

A. (Chen) The maximum torsional stresses are caused by the fourth order. I calculated the fourth order stresses for all modes. This contrasts to TDI's calculation, which only allows the calculation of fourth order stresses for single modes. I calculated these stresses at full load, overload, 95% of rated load and 105% of rated load. The fourth order stresses are as follows:

Fourth Order Stresses

RPM	KW	PSI
450	3500	3455
450	3900	3740
427.5	3500	3071
472.5	3500	4010

43. What was the result of your sum of orders response and nominal stress calculation?

A. (Chen) The sum of orders stresses at full load, overload, 95% and 105% of rated load are as follows:

Sum of Orders Stresses

RPM	KW	PSI
450	3500	5101
450	3900	5401
427.5	3500	6232

472.5	3500	5673

44. Do the crankshafts comply with DEMA at overload conditions?

A. (Chen) Yes. At 3900 KW the fourth order stress is 3740 psi and the sum of orders stress is 5401 psi. These figure are well within the DEMA allowables. It should be noted that DEMA does not require stress calculations at overload conditions. Nonetheless, the replacement crankshafts are within the DEMA stress limits at overload.

45. Dr. Chen, have you ever seen crankshafts that have failed from torsional stress?

A. (Chen) Yes. I have seen quite a few crankshafts that have failed from torsional stress.

46. Are you aware of any crankshafts that comply with DEMA that have failed primarily due to torsional stress.

A. (Chen) No. In more than thirty (30) years of experience in the diesel engine industry, I do not know of any situations in which a crankshaft that met DEMA recommendations has failed primarily from torsional fatigue. I was chairman of the DEMA Technical Committee from 1971 through 1973 and I can state with confidence that a crankshaft that complies with DEMA is reliable for its intended service.

V. THE FATIGUE ANALYSIS AND FIELD TESTING OF THE CRANKSHAFTS SHOW THAT THE CRANKSHAFTS WILL NOT FAIL DURING OPERATION

47. What is the purpose of a fatigue analysis?

A. (McCarthy, Johnston) The purpose of a fatigue analysis is to determine the useful life of a given component (in this case a crankshaft) for its specified service loads. FaAA performed a fatigue analysis which enabled us to conclude that the crankshafts have unlimited life for their intended service.

48. Why did FaAA perform a fatigue analysis of the crankshafts?

A. (McCarthy, Johnston) Although the crankshafts meet the nominal stress recommendations of DEMA for operation at 3500 KW and 3900 KW, the stresses for combined orders calculated from the torsiograph measurements are close to the recommended allowable of 7000 psi. (The stresses for single orders are considerably lower than the recommended allowable of 5000 psi.) While the DEMA limits are believed to contain an intrinsic safety margin, a fatigue analysis was performed to determine the true safety margin of the crankshafts and to provide an additional measure of assurance, independent of design criteria specified by <u>any</u> code, that the crankshafts are adequately designed to perform their intended function in the Shoreham EDGs.

49. How was the fatigue analysis conducted?

A. (Johnston, McCarthy) To conduct a fatigue analysis FaAA had to determine the maximum stresses the crankshafts would see in service, as well as the endurance limit for the crankshaft material. FaAA performed a two part analysis to determine the maximum stresses. First, a dynamic torsional analysis of the crankshaft was performed 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. Torsional and gas pressure loading cases were considered in the finite element model to evaluate the effects of twisting and bending loads. These analyses permitted FaAA to determine the maximum stresses. These stresses were also obtained from a dynamic strain gage test on the replacement crankshaft.

The fatigue endurance limit was established for the replacement crankshaft 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. The endurance limit was compared with values provided in the literature and found to be acceptable. The factor of safety against fatigue failure was computed from the test data gathered from the original and

replacement crankshafts. The factor of safety is large enough to provide confidence in the reliability of the crankshafts.

50. Let us discuss separately each part of the fatique analysis. What is the purpose of a dynamic torsional analysis?

A. (Johnston) 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 computes the phase relationship between the various orders and modes, which permits this summation. The dynamic torsional analysis represents a more accurate calculation of the stresses actually experienced by the crankshaft during operation than conventional analytical techniques. (Technical details of the dynamic torsional model are contained in Section 3.1 of Exhibit C-17).

51. What did you do with the total torque calculated from the dynamic torsional analysis?

A. (Johnston) The total torque was used as input data to the finite element model to determine the actual maximum state of stress in the crankshaft.

52. What was the purpose of constructing a finite element model of a one quarter crank throw?

A. (Johnston) The nominal crankshaft stress values calculated from the dynamic model (i.e. total torque) are

considerably less than the actual maximum stresses in the crankshaft. Those 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 response of the crankshaft; and, (2) connecting rod forces applied to the crankpin and reacted at bearing supports. A finite element model of a one quarter crank throw, considering stresses due to torsional loading and stresses due to gas pressure loading, was used to compute the actual maximum value and location of stresses in the crankpin fillet area. The strain gages used during dynamic testing were placed at the location of maximum stress calculated by the finite element model. (Technical details concerning the finite element model are contained in Section 3.2 of Exhibit C-17).

53. Please describe the dynamic testing.

A. (Johnston) Stone & Webster conducted dynamic tests on the replacement crankshaft on EDG 103 in January, 1984. Instrumentation for the measurement and recording of significant dynamic data included the following:

1. Cylinder firing pressure of cylinder Nos. 5 and 7 was measured;

2. Dynamic torque in the crankshaft between the engine casing and the flywheel was measured by a strain gage torque bridge;

3. Crankpins Nos. 5 and 7 were instrumented with three element strain rosettes to measure crankpin fillet dynamic strains.

These tests were performed under a variety of loads and transient conditions to investigate the dynamic response of the crankshaft.

54. How were the results of these tests used in FaAA's analysis?

A. (Johnston) First, the cylinder firing pressure measured by Stone & Webster was utilized to obtain the gas pressure loading for input to the dynamic torsional analysis. The total torque produced by this loading was calculated and corresponds closely to the torque measured by Stone & Webster near the flywheel. (Exhibit C-23). Second, the dynamic strains measured by Stone & Webster in the crankpin fillets of crankpin Nos. 5 and 7 were used to compute the maximum stresses, which were used to calculate the factor of safety. These stresses are within the range predicted by FaAA's finite element analyses. (Exhibit C-24).

55. Are the results of Stone & Webster's dynamic torsional testing confirmed by the analytical models used by FaAA?

A. (Johnston, McCarthy) Yes. The results of FaAA's

analytical models agree with the dynamic strain gage tests. Dynamic testing of the crankshaft, in this regard, is considered to be an essential element of the design review program because it is only through carefully conducted measurement that the actual engine dynamics and local component stresses are confirmed.

56. After measuring the maximum stresses in the fillet area, what was the next step in your analysis.

A. (Johnston) The next step in the analysis was to compare the measured stresses with the fatigue endurance limit of the replacement crankshafts. The results of the finite element analysis were used to determine the maximum principal stress range in the fillet area, which was then compared to the fatigue endurance limit of the replacement crankshaft.

57. How was the fatigue endurance limit of the replacement crankshaft established?

A. (Johnston) 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.

58. How was the endurance limit established for the original crankshafts?

A. (Johnston) The original 13-inch by 11-inch crankshaft on EDG 101 was instrumented with strain gages in the fillet location of Crankpin No. 5. This fillet had previously experienced a fatigue crack during performance testing. After the test, the three-dimensional finite element model of a quarter section of a crank throw showed that the strain gages 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, it was determined that the endurance limit for the original crankshafts was 36.5 ksi.

59. What is the fatigue endurance limit for the replacement crankshafts?

A. (Johnston) The fatigue endurance limit 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.

60. Having obtained the fatigue endurance limit for the replacement crankshafts, were you able to calculate the factor of safety against fatigue failure?

A. (Johnston) Yes. The factor of safety against fatigue failure was calculated by plotting the maximum principal stress range measured in the crankpin fillet area on a Goodman diagram, constructed using the fatigue endurance limit and the ultimate tensile strength values for the replacement crankshafts. (Exhibit C-25). The factor of safety against fatigue failure is 1.48, without taking into account any beneficial effect of shot peening the fillet regions.

61. Does a factor of safety of 1.48 provide sufficient assurance that the replacement crankshafts are adequate for their intended service in the Shoreham EDGs?

A. (McCarthy) Yes.

62. What is the basis for your opinion that a factor of safety of 1.48 is sufficient for the replacement crankshafts?

A. (McCarthy) To explain that I must first explain what a factor of safety is. With that understanding, the acceptability of a factor of 1.48 will become apparent.

63. What is a factor of safety?

A. (McCarthy) 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, i.e. effects or things we don't know. 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.

64. What is the difference between a factor of safety in endurance limit, yield strength, and in ultimate strength?

A. (McCarthy) 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.

65. What types of uncertainties is the factor an allowance or compensation for?

A. (McCarthy) Uncertainties as to service load, material properties, stress concentration factors, lifetime, etc., which obviously are directly related to the amount of testing, analysis, and understanding a designer has of a particular part and its service environment.

66. What is an acceptable allowance for this uncertainity, or, in other words, what is an acceptable factor of safety?

A. (McCarthy) This is totally determined by the degree of uncertainity 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. Portions from several of the most widely used Mechanical Engineering design references are attached as Exhibit C-26. 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.

67. How well is the design of the replacement crankshafts understood?

A. (McCarthy) To put it simply, extremely well. We have the benefit of the information 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, far less information to work with when assessing a design. This results in uncertainities in the design being reduced substantially.

68. What does this understanding of the crankshaft design mean in terms of an acceptable factor of safety.

A. (McCarthy) 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 this particular part, it would be my opinion that our degree of understanding would certainly permit 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. Therefore the factor of 1.48 is quite acceptable.

VI. CONCLUSION

69. Please summarize your conclusions.

A. (All) Our conclusions are as follows:

1) The maximum stresses that will be exerted on the crankshaft during operation, both for a single order and for combined orders, are below the DEMA allowables. These stresses were calculated using analytical techniques commonly used by the diesel engine industry, and by the use of torsiograph test data.

2) There is no requirement that the crankshafts comply with the design criteria of ABS, Lloyd's, IACS or FEV. Noncompliance with the design criteria of any of these organizations does not mean the replacement crankshafts are inadequately designed for their intended service at Shoreham. Notwithstanding the fact that compliance is not required, ABS has approved the replacement crankshafts.

3) FaAA performed a fatigue analysis of the replacement crankshafts. This analysis establishes that the crankshafts have a factor of safety in fatigue of 1.48, without benefit of shot peening. This is more than adequate to provide reasonable assurance the crankshafts will not fail in fatigue during operation.

4) There is no basis for the County's contention that the replacement crankshafts are inadequately designed for operation at full load (3500 KW) or overload (3900 KW). Both conventional and highly sophisticated analyses, as well as extensive testing, establish that the crankshafts are adequately designed for the service they will see in the Shoreham emergency diesel generators.

Attachment 1

Failure Analysis Associates

ROGER L. McCARTHY

Specialized Professional Competence

Mechanical, machine, and mechanism design. Dynamic mechanical system design, analysis modeling, control (including dedicated computer control), and failure analysis. Custom product design. Human factors engineering and testing; design analysis of man/machine interface. Design analysis research. Risk analysis; quantification of hazards posed by design and construction of mechanical components, products, or system failure in the industrial and transportation environments. Design analysis through large scale accident data analysis and evaluation, including vehicle design and collision performance. Evaluation of mechanical/electrical design-related explosion hazard; heat transfer design. Reinforced polymer composite design analysis, including tires. Patent analysis relating to mechanical design.

Background and Professional Honors

A.B. (Philosophy), University of Michigan, with High Distinction
B.S.E. (Mechanical Engineering), University of Michigan, summa cum laude
S.M. (Mechanical Engineering), Massachusetts Institute of Technology
Mech.E. (Mechanical Engineering), Massachusetts Institute of Technology
Ph.D. (Mechanical Engineering), Massachusetts Institute of Technology

President,
Failure Analysis Associates
Principal Design Engineer
Failure Analysis Associates
Program Manager, Special Machinery Group,
Foster-Miller Associates, Inc.
Project Engineer, Machine Design and Development Engineering, Engineering Development Division, Proctor & Gamble Company, Inc.

Registered Professional Mechanical Engineer, California, #M20040
Registered Professional Mechanical Engineer, Arizona, #13684
Phi Beta Kappa, Sigma Xi, James B. Angell Scholar
National Science Foundation Fellow
Outstanding Undergraduate in Mechanical Engineering, University of Michigan

Member, American Society of Metals, American Society of Mechanical Engineers, Society of Automotive Engineers, American Welding Society, National Safety Council, American Society for Testing and Materials
Member, American Society of Safety Engineers
Member, Human Factors Society, System Safety Society, National Society of Professional Engineers
Member, American Society of Heating, Refrigeration, and Air-Conditioning Engineers
Member, National Fire Prevention Association

Selected Publications

"School Bus Wheel Rim Safety — Multipiece vs. Single Piece," National School Bus Report, Springfield, Virginia (December 1982) (with G. E. McCarthy).
"Warnings on Consumer Products: Objective Criteria For Their Use," 26th Annual Meeting of the Human Factors Society, Seattle, Washington (October 25-29, 1982) (with J. N. Robinson, J. P. Finnegan and R. K. Taylor).
"Average Operator Inaction Characteristics with Lever Controls — Study of the Column Mounted Gear Selector Lever," 26th Annual Meeting of the Human Factors Society, Seattle, Washington (October 25-29, 1982) (with J. P. Finnegan, G. F. Fowler and S. B. Brown).
"Catastrophic Events: Actual Risk versus Societal Impact," 1982 Proceedings, Annual Reliability and Maintainability Symposium, Los Angeles, California (January 26-28, 1982) (with J. P. Finnegan and R. K. Taylor).

"Product Recall Decision Making: Valid Product Safety Indicators," Proceedings of the Fourth International System Safety Conference, San Francisco, California (July 9-13, 1979). Published by Professional Engineer Magazine (March 1981).
"Large Vehicle Wheel Servicing: Reduction of Risk Through Implementation of An CSHA Standard Governing Multipiece and Single Piece Rims: Phase IV," Published by the National Wheel and Rim Association (March 1981) (with J. P. Finnegan).
"Program to Improve Down Hole Drilling Motors: Task 2, Lip Seal Design," Failure Analysis Associates Report FAA-81-7-6 to Sandia National Laboratories (October 1980) (with V. Pedotto).
"A Safety and Fracture Mechanics Analysis of the Pneumatic Tire: A Perspective on the Firestone 500 Radial Tire," Presented at the International Conference on Reliability, Stress Analysis and Failure Prevention, of the American Society of Mechanical Engineers, San Francisco, California (August 18-21, 1980) (with W. G. Knauss).
"Multipiece and Single Piece Rims: The Risk Associated with Their Unique Design Characteristics: Phase III," Published by the National Wheel and Rim Association (June 1980) (with J. P. Finnegan).
"An Engineering Safety Analysis of the Steel Belted Radial Tire," Society of Automotive Engineers Paper #800840 (June 9-13, 1980).
"A Simple Technique to Improve the Allocation of Safety Inspection Resources," Proceedings of the Fourth International System Safety Conference, San Francisco, California (July 9-13, 1979) (with P. M. Besuner).
"An Engineering Analysis of the Risk Associated with Multipiece Wheels," National Highway Traffic Safety Administration, ANPR Docket No. 71-19, Number 7 (June 1979) (with J. P. Finnegan).
"Planar Thermic Elements for Thermal Control Systems," Journal of Dynamic Systems, Measurement and Control, Vol. 99, Series G, No. 1 (March 1977) (with B. S. Buckley).

Attachment 2

Failure Analysis Associates

PAUL R. JOHNSTON

Specialized Professional Competence

Static and dynamic analysis of structures; response spectrum and time history analysis of structures; earthquake engineering; probabilistic methods in structural analysis, decision analysis; the finite element method, non-linear stress analysis; analysis of PWR steam generator tube denting phenomena; soil-structure interaction; geotechnical engineering, elasto-plastic constitutive relations for soils, consolidation, tunnelling in soil or rock; design of steel and reinforced concrete structures, automated design.

Background and Professional Honors

B.A., B.A.I. (Civil Engineering), Trinity College, Dublin University, Ireland (First Class Honours, Foundation Scholar)
M.S. (Structural Engineering), Stanford University
Ph.D. (Geotechnical Engineering), Stanford University (John A. Blume Fellowship)
Structural Engineer, Failure Analysis Associates
Consulting Assistant Professor, Department of Civil Engineering, Stanford University
Researcher, Geotechnical Group, Department of Civil Engineering, Stanford University
Geotechnical Engineer, Jo Crosby and Associates
Member, American Society of Civil Engineers
Member, Institute of Engineers of Ireland

Selected Publications

"Probabilistic Environmental Model for Solid Rocket Motor Life Prediction," NWC-TP-6305 (August 1981) (with G. Derbalian, J. Thomas and G. Brooks).
"Northeast Utilities Tube Plugging Criteria," FAA-81-8-12 (August 1981) (with J. Thomas, G. Derbalian, H. Wachob and S. Rau).
"Finite Element Consolidation Analysis of Tunnel Behavior in Clay," Ph.D. Thesis, Stanford University (June 1981).
"Structural Analysis of PWR Steam Generator Egg Crates," FAA-80-7-3 (June 1980) (with J. Thomas, S. Rau and G. Derbalian).
"Structural Analysis of Millstone Unit No. 2, Steam Generator Tubes and Support Plate," FAA-79-06-03 (June 1979) (with J. Thomas, G. V. Ranjan and G. Brooks).
"Steam Generator Support Plate Analysis for Indian Point Unit 2," FAA-79-01-3 (January 1979) (with J. Thomas, G. Derbalian, G. V. Ranjan and R. Cipolla).
"Quasi-Material Properties for Millstone Unit 2 Steam Generator Support Plate Analysis," FAA-78-12-3 (December 1978) (with J. Thomas and G. V. Ranjan).

Attachment 3

EUGENE F. MONTGOMERY
18 Fourth Place
Syosset, New York 11791

Telephone - Home: 516/921-0866
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EXPERIENCE SUMMARY:

Over eight years of progressively increasing responsibility in the performance and management of engineering mechanics activities on nuclear power plant piping systems and equipment for electric utility and consulting engineering firms.

EDUCATION:

Columbia University School of Engineering and Applied Sciences, New York, New York

Bachelor of Science, Mechanical Engineering	- May 1973
Master of Science, Mechanical Engineering	- October 1974
Mechanical Engineer (Professional Degree)	- January 1981

Queens College, City University of New York, Queens, New York

Bachelor of Arts, Physics	- May 1973

EXPERIENCE: (See Attachment for Details)

1981 to Present — Stress Analyst, Nuclear Engineering Department
Long Island Lighting Company
175 East Old Country Road
Hicksville, NY 11801

Shoreham Nuclear Power Station - Unit No. 1
Mark II BWR/4 Capacity 819 Mw Net

Responsible Owner's representative for the engineering, coordination, review and approval of stress related activities performed in support of Shoreham licensing, start-up and system turnover.

1980 to 1981 — Senior Engineer, Stress Anlaysis Engineering Department
Burns and Roe, Incorporated
185 Crossways Park Drive
Woodbury, NY 11797

Washington Nuclear Project (Hanford) Unit No. 2
Mark II BWR/5 Capacity 1100 Mw Net

Lead Engineer for various engineering evaluations related to fatigue analysis and high frequency effects of Mark II Suppression Pool loads on containment piping, equipment and support structures.

EXPERIENCE (Cont'd.)

1978 to 1980 — Senior Engineer, Stress Analysis Engineering Department
Ebasco Services, Incorporated
2 World Trade Center
New York, NY 10048

Laguna Verde Units No. 1 and 2
Mark II BWR/6 Capacity 600 Mw Net

Stress Engineer responsible for the design, analysis and checking of major ASME III Code Class 2, 3 and USAS B31.1 nuclear power piping systems.

1977 to 1978 — Engineer 'A', Stress Analysis Engineering Department
Burns and Roe, Incorporated
185 Crossways Park Drive
Woodbury, NY 11797

Washington Nuclear Project (Hanford) Unit No. 2
Mark II BWR/5 Capacity 1100 Mw Net

Stress Engineer responsible for the combined application of finite element methods (ANSYS), piping flexibility analysis (ADLPIPE) and Fortran IV computer programming to achieve the optimum design of nuclear power piping systems and their supports (normal/pipe-rupture) according to project specifications.

PROFESSIONAL SOCIETY MEMBERSHIP:

Associate Member - American Society of Mechanical Engineers

Associate Member - New York State Society of Professional Engineers

Member - Tau Beta Pi (National Engineering Honor Society)

REFERENCES: Will be furnished on request.

DETAILS OF EXPERIENCE LISTING

From 3/81 to Present

Stress Analyst, Nuclear Engineering Department
Long Island Lighting Company
175 East Old Country Road
Hicksville, NY 11801

Shoreham Nuclear Power Station Unit No. 1
Mark II BWR/4 Capacity 819 Mw Net

Responsible Owner's representative for the engineering, coordination, review and approval of stress-related activities performed in support of Shoreham licensing, start-up and system turnover. Major assignments included the following:

- In responsible charge of engineering review and approval of calculations performed by project consultants (Stone & Webster, Inc., General Electric) for seismic qualification and hydrodynamic re-evaluation of all safety-related equipment subject to IEEE-344, 1975 and the latest NRC criteria. Represented client interests at NRC-Equipment Qualification Branch technical audits of detailed dynamics analyses and test reports. Interfaced and coordinated between NRC and consultants to obtain acceptable resolutions on outstanding technical concerns.

- Member of Motor Operator Test Group addressing issues on vibration aging and mechanical fatigue of Limitorque motor operators. Participated in formulation of procedures and test specifications used to qualify the equipment to long-duration, high frequency loads.

- Initiated and coordinated stress-engineering software development for the Nuclear Engineering Department. Conducted evaluations to assemble an applications package consisting of essential structural and piping codes.

- Lead Engineer for the Independent Design Review of the safety-related portions of the ECCS Core Spray System piping, supports, equipment and structures. Developed program plan and description, reviewed technical proposals. Coordinated audit open items/findings resolutions between Independent Design Reviewer (Teledyne Engineering Services) and project consultants.

- Project Engineer for the As-Built Piping Reconciliation Program responsible for monitoring and minimizing the impact of field modifications due to calculation close-out and reviews.

- LILCO Engineering Specialist for the Transamerica Delaval (TDI) Recovery Program. Reviewed diagnostic calculations on failure of engine crankshaft and analyses of replacement crankshaft design. Developed "tracking System" for nuclear/non-nuclear diesel engine failure experience for use in the TDI Owner's Design Review/Quality Revalidation effort.

Special Training

LILCO sponsored departmental training lectures. Covered topics included:

- 10 CFR 50 Appendix B Quality Assurance Requirements
- BWR Systems Familiarization Course
- General Employee Training (GET) (for access to vital plant areas)
- Shoreham Emergency Preparedness Training
- English Language Institute Study Course
- Technical Specialist QA Auditor Training

From 4/80 to 3/81

Senior Engineer, Stress Analysis Engineering Department
Burns and Roe, Incorporated
185 Crossways Park Drive
Woodbury, N.Y. 11797

Washington Public Power Supply System

Washington Nuclear Project (Hanford) Unit No. 2
Mark II BWR/5 Capacity 1100 Mw Net

In responsible charge of engineering evaluations in the following areas:

- Lead Engineer for the fatigue analysis of MSRV lines and downcomers subjected to extended duration LOCA-related hydrodynamic loads. Supervised engineering personnel in lower classifications.
- Member of Mark II SRSS/LCAC (Square-Root-Sum-Square and Load Combination Acceptance Criteria) Subcommittee addressing issues on MSRV and downcomer fatigue analysis, essential piping functional capability, SRSS Newmark-Kennedy Criteria and high frequency content of Mark II loads.
- Lead Engineer for analysis of drywell ECCS (Emergency Core Cooling Systems) for Annulus Pressurization faulted loading conditions. Assisted and trained other stress analysts in performing calculations on conformance with project design specifications and ASME code.

Conceptual Engineering

- Developed an analytical approach for determining the optimum support configuration restraining large, eccentric motor-operator valves. Guidelines in the form of simplified computational procedures and tables were prepared. (Published paper titled, "Optimum Rigid Support Spacing for Eccentric Operator Valves," June 1981.)

From 5/78 to 4/80 — Senior Engineer, Stress Analysis Engineering Department
Ebasco Services Incorporated
2 World Trade Center
New York, N.Y. 10048

Stress Engineer responsible for the design, analysis, and checking of major ASME Code Class 2, 3 and USAS B31.1 nuclear power piping systems.

Comision Federal de Electricidad

Laguna Verde Units No. 1 and 2
Mark II BWR/6 Capacity 600 Mw Net

- o Responsible for thermal, pressure, deadweight and seismic design, analysis and checking of safety-related systems according to ASME Boiler and Pressure Vessel Code, Section III and USAS B31.1 using the proprietary pipe flexibility code PIPESTRESS 2010.
- o Developed initial support location, selection and sizing (or modified line routing, when necessary) on the following BWR systems: reactor water cleanup (RWCU), reactor core isolation cooling (RCIC), high pressure core spray (HPCS), low pressure core spray (LPCS), residual heat removal (RHR), standby liquid control (SLC), and numerous other Reactor and Control Building systems.
- o Prepared, checked and reviewed system stress analysis reports. Interfaced equipment allowable nozzle loads, pipe support loads, and postulated pipe stress break locations with other disciplines.

Houston Lighting and Power Company

Allens Creek Nuclear Generating Station
Mark III BWR Capacity 1200 Mw Net

- o Performed investigative study to determine the structural response of proposed Main Steam and Reactor Feedwater seismic interface/pipe rupture restraint system outside primary containment. An in-house dynamic-plastic finite element code, PLAST 2267, used for analysis.

Conceptual Engineering

- o Responsible for deriving maximum seismic support spans based upon a frequency design criteria. Nondimensional charts and tables developed for supports around right angle elbows, large radius bends, and parallel offset configurations. Prepared summary report for inclusion in project Pipe Stress Analysis Guidelines.

Special Training

Ebasco Services, Inc. sponsored departmental training lecture series. Covered topics included:

- o Code Stress Basis
- o Quality Assurance
- o Stress Analysis of Fossil Plant Piping
- o Pipe Rupture Interface with Stress Analysis
- o Thermal Stress Analysis According to B31.1
- o Seismic Charts Analysis
- o Vibration Theory and Problems in Piping

From 2/77 to 4/78	Engineer 'A', Stress Analysis Engineering Department Burns and Roe, Incorporated 185 Crossways Park Drive Woodbury, N.Y. 11797

Stress Engineer responsible for the combined application of finite element methods (ANSYS), piping flexibility analysis (ADLPIPE) and Fortran computer programming to achieve the optimum design of nuclear power piping systems and their component supports according to the applicable portions of ASME Boiler and Pressure Vessel Code, Section III.

Washington Public Power Supply System

Washington Nuclear Project (hanford) Unit No. 2
Mark II BWR/5 Capacity 1100 Mw Net

- o Responsible for the pipe rupture analysis of Main Steam high energy line breaks outside primary containment. Non-linear, elasto-plastic, dynamic finite element analysis (ANSYS) used to determine whip restraint gap size, maximum support member forces/moments, plastic piping response, penetration nozzle reactions, MSIV end loads and deformations. Prepared and reviewed final stress analysis report.

- o Responsible for the engineering, design and analysis of major wetwell piping and components subjected to direct hydrodynamic Mark II submerged structure loads. Time history and response spectra techniques (ADLPIPE) used to locate supports and evaluate piping response on MSRV lines, downcomers and miscellaneous wetwell penetrations under normal/upset/emergency/faulted hydrodynamic loading conditions.

- Coordinated application of DFFR (GE Dynamic Forcing Function Report) and DAR (Design Assessment Report) for developing force vs. time curves due to SRV discharge, Chugging, Condensation Oscillation, Pool Swell and Fallback input to pipe stress analysis. Developed Fortran programs for data file manipulation.

- Performed detailed analysis of MSRV X-Quencher device and its associated support structure under direct and indirect structural loads. Verified member sizes and anchor bolt-down adequacy. Prepared final stress report.

Jersey Central Power and Light

Three Mile Island Unit No. 2
PWR Capacity 880 Mw Net

- Responsible for verifying the design adequacy of Reactor Pressure Vessel and Main Steam Generator base plate shear pin bolt design under longitudinal and circumferential hot/cold leg coolant line breaks. The dynamic finite element codes STARDYNE and ANSYS were used in conjunction with an empirically developed collapse moment equation. Prepared final stress report.

Conceptual Engineering

Prepared Fortran software necessary to interface company developed piping graphics package with ADLPIPE, a conventional pipe flexibility code. Linkage permitted free thermal execution of designers' proposed routing while simultaneously plotting the layout on orthographic or isometric view.

Special Training

- "Practical Seismic Design of Structures" administed by Structures Group, Metropolitan Section ASCE.

- "Advanced Topics and New Developments in Finite Element Methods" administered by MARC Analysis Research Corporation.

Attachment 4

PEI CONSULTANTS

Biographical Data On

Dr. Simon K. Chen, PE

March 16, 1983

Position President

Home 325 Racine Street, Delavan, WI 53115
Home Phone: 414-728-6994

Education

B.S., M.E.	1947	National Chiao-Tung University
M.S., M.E.	1959	University of Michigan
Ph.D., M.E.	1952	University of Wisconsin
M.B.A.	1964	University of Chicago, Executive Program

Work Experience

President, Power and Energy International, Inc. — 1979 - present
Technical consulting and product development

President, Beloit Power Systems, Inc. — 1973 - 1979
Manufacturers of engine and turbine driven alternators, up to 15,000KW, rotary positive screw gas compressor, power plant controls, and gen-sets.

V.P., Engineering and Application, Fairbanks-Morse Power Systems Colt Industries — 1969 - 1973
Developer of O.P. Blower series line with increased rating, O.P. sparked gas engine, manufacturer of SEMT-PC-2 for marines, stationary and nuclear standby applications, developer of 38A-20 engine, producer of large irrigation pump, rotary compressor, alternators and motors.

Divisional Chief Engineer, Diesel Engine R&D, International Harvester Company — 1965 - 1969
Developer and manufacturers of vehicular diesels and spark-gas engines for construction equipment, farm equipment, medium-duty truck, and industrial applications.

Chief Project Research Engineer, Engineering Research, IH — 1956 - 1965
Corporate research on alternate power plant, engine combustion, advanced power train concept, advanced vehicle analysis, and corporate product planning.

Project Engineer, IH, Melrose Park — 1952 - 1956
In charge of combustion research on diesel and stratified charge engine.

Technical Society Membership List and Honors

SAE, ASME, SNAME, EGSMA, CIE, Who's Who in the World, Who's Who in Finance and Industry, Engineers of Distinction by Engineers Joint Council in 1973, SAE Arch T. Colwell Merit Award in 1966, University of Wisconsin Alumni Distinguished Service Award, 1973, Chinese Institute of Engineer's Achievement Award in 1976, Director and Technical Chairman of Diesel Engine Manufacturing Association, 1971-73, Member Compressed Air and Gas Institute, 1973-79, SAE Fellow-1983, Registered Professional Engineer - State of Wisconsin.

Power and Energy International Inc. P.O. 1064 555 Lawton Ave. Beloit, WI 53511 608/382-7071

PEI CONSULTANTS

Publications

Dr. Simon K. Chen

January 16, 1984

- "Compression and End Gas Temperatures from Iodine Absorption Spectra," Co-author, SAE, 1954.
- "Development of a Single Cylinder Compression Ignition Research Engine," Co-author, SAE 650733, 1965.
- "Development and Evaluation of the Simulation of the Compression-Ignition Engine," Co-author, SAE 650451, 1965.
- "Engine Development Criteria and Techniques," Modern Engineering and Technology Seminar, Taiwan, Republic of China, July 1974.
- "Engine Cycle Analysis and Combustion Problems," Modern Engineering and Technology Seminar, Taiwan, Republic of China, July 1974.
- "Diesel Application," Modern Engineering and Technology Seminar, Taiwan, Republic of China, July 1974.
- "Highlights of the Energy Session," Energy Quarterly, Republic of China, January 1975.
- "A Collection of Abridged Management Papers," Modern Engineering and Technology Seminar, Taiwan, Republic of China, July 1976.
- "Marketing in a Competitive Market," Modern Engineering and Technology Seminar, Taiwan, Republic of China, July 1976.
- "Management Philosophy and High Technology Development," Energy Quarterly, Taiwan, Republic of China, January 1978.
- "Vibration Analysis for a Sound Generator-Set Design," Electrical Generating Systems Marketing Association, Chicago, IL, September 26-27, 1978.
- "Waste Heat Recovery Cycle Analysis and Systems for Diesel and Gas Turbine Engines," 13th CIMAC Conference, Vienna, Austria, May 7-10, 1979.
- "Small Industrial Diesel Planning," September 16, 1980.
- "An International Perspective of Taiwan's Automotive Industry," Society of Automotive Engineers, SAE-ROC Technical Meeting, Tawian, Republic of China, November 23-25, 1981.
- "The Development of ROC Machine Tool Industry and the Impact of Automation," Industrial Technology Research Institute, Taiwan, Republic of China, September 1981.
- "Japan's Robot and Robotics Development," March 11, 1982.
- "Techno-Economic Recommendations to Fight Recession Accelerated by Energy Shock," May 5, 1982.
- "US Robots and Robotics," August 1983.
- "A Review of Engine Advanced Cycle and Rankine Bottoming Cycle and Their Loss Evaluations," Co-authored, SAE 830124, 1983.
- "Flexible Manufacturing Systems Applications," Modern Engineering and Technology Seminar, Singapore, November 1983.
- "The Impact of Automation on Newly Industrialized Countries," Modern Engineering and Technology Seminar, Singapore, November 1983.