UNITED STATES OF AMERICA
NUCLEAR REGULATORY COMMISSION

BEFORE THE ATOMIC SAFETY AND LICENSING BOARD

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

JOINT TESTIMONY
of
CARL H. BERLINGER, SPENCER H. BUSH,
ADAM J. HENRIKSEN, WALTER W. LAITY, AND PROFESSOR ARTHUR SARSTEN
on
CONTENTIONS CONCERNING TDI EMERGENCY DIESEL GENERATORS
at the
SHOREHAM NUCLEAR POWER STATION

VOLUME 1

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## INTRODUCTION

Q. Please state your names, your business addresses, and your professional qualifications.

A. (Berlinger) My name is Carl H. Berlinger. I am the NRC Project Group Manager for matters pertaining to Transamerica Delaval, Inc., emergency diesel generators. A summary of my professional qualifications and experience is included as Attachment 1.

A. (Bush) My name is Spencer H. Bush. I am self-employed, under the firm name of Review and Synthesis Associates. A summary of my professional qualifications and experience is included as Attachment 2.

A. (Henriksen) My name is Adam J. Henriksen. I am self-employed, under the firm name of Adam J. Henriksen, Inc. A summary of my professional qualifications and experience is included as Attachment 3.

A. (Laity) My name is Walter W. Laity. I am employed by Battelle Memorial Institute at the Pacific Northwest Laboratory in Richland, Washington. A summary of my professional qualifications and experience is included as Attachment 4.

A. (Sarsten) My name is Arthur Sarsten. I am a member of the faculty of the Norwegian Institute of Technology at Trondheim, Norway. A summary of my professional qualifications and experience is included as Attachment 5.

Q. What is the subject matter of your testimony?

A. (Berlinger) My testimony addresses comments by the NRC staff on the testimony presented by NRC's consultants.

A. (Bush) My testimony addresses metallurgical considerations related to crankshaft fabrication and shotpeening, crack initiation and propagation, and nondestructive examination.

A. (Henriksen) My testimony addresses the technical adequacy of the four components discussed in Suffolk County's contentions, excluding analytical methods for fracture mechanics and stress analysis.

A. (Laity) My testimony addresses the technical assistance that the Pacific Northwest Laboratory is providing to the NRC staff in the review and evaluation of Transamerica Delaval, Inc. (TDI) emergency diesel generators.

A. (Sarsten) My testimony addresses stress analysis of diesel engine components and standards for the design of crankshafts.

Q. How is this testimony organized?

A. (Berlinger, Laity) First, the technical assistance that the Pacific Northwest Laboratory (PNL) is providing to the NRC staff is discussed. This is followed, in turn, by a summary of the testimony presented by the witnesses, and a summary of the premises on which this testimony is based. Suffolk County's contentions admitted by the Atomic Safety and Licensing Board are then addressed.

Role of the Pacific Northwest Laboratory

Q. What is PNL's role with the NRC staff on matters pertaining to TDI diesel engines?

A. (Berlinger, Laity) PNL is providing technical assistance to the NRC staff in reviewing the program established by the TDI Diesel Generator Owners' Group for assessing the adequacy of TDI diesel generators as emergency power sources for safety-related nuclear systems. I (Laity) head the project management team established at PNL for this effort.

PNL's role is to evaluate the technical adequacy of reports and related information submitted to the NRC staff on TDI diesel generators, and to identify any matters that require clarification or elaboration. While PNL's reviewers may perform calculations as appropriate for the review process, it is not the role of PNL to perform independent analyses of the components in question.

PNL has secured the services of several consultants who have extensive experience in the design, testing, operation, and maintenance of medium-speed diesel engines. The PNL project management team also calls upon experts as necessary in areas such as metallurgy, fracture mechanics, stress analysis, nondestructive testing, and heat transfer. These experts provide advice and counsel to PNL and to the NRC staff on the numerous issues that have been raised in regard to the adequacy of TDI diesel generators as emergency power sources for nuclear systems.

In the preparation of this testimony, the witnesses have reviewed the testimonies filed by Suffolk County and by Long Island Lighting Company

(LILCO). The witnesses have also reviewed various relevant documents submitted by the TDI Diesel Generator Owners' Group to the NRC staff, and participated in meetings of the Owners' Group with the Staff. Two of the PNL witnesses (Laity and Henriksen, a PNL consultant) have examined key components of the TDI diesels at the Shoreham Nuclear Power Station during engine disassemblies.

## Summary of Testimony

Q. Please summarize your testimony on the four components in contention.

A. (All) In summary, the information available for our review from LILCO and from the TDI Diesel Generator Owners' Group did not provide an adequate basis for us to reach an unequivocal conclusion regarding the overall adequacy of the Shoreham TDI diesel generators as emergency power sources for nuclear systems. Our reservations pertain to two of the four components in contention: the crankshafts, and the cylinder blocks for the 101 and 102 engines. The following is a brief summary of our position on these components and on the other two components in contention.

### Crankshafts

We have concluded that, at rated engine load, the torsional stresses in the crankshafts exceed the DEMA Standard Practices. Although the crankshafts may still perform satisfactorily, we believe that the information available for our review is not conclusive in this regard. One approach that would resolve our concern about the crankshafts would be to test an engine (either the 101 engine or the 102 engine to also resolve concerns about the cylinder blocks) to $10^7$ cycles (about 740 hours) at rated load, with the engine operated at 110% of rated load for 2 hours out of every 24 hours.

On the basis of information presented in LILCO's testimony, we have concluded that neither the first shotpeening nor the second shotpeening of two of the crankshafts degraded their fatigue resistance. Rather, the second shotpeening may have enhanced the crankshafts' fatigue resistance. However, in our opinion, the effect is not quantifiable from available information.

Cylinder Blocks

Our reservations about the cylinder blocks stem from unresolved questions as to whether or not existing cracks in the camshaft gallery are benign. Pending a more definitive explanation of the origin of these cracks, the stresses in the area where they are located, and the predicted path of crack propagation, we do not have an adequate basis for drawing a conclusion about the suitability of these blocks for nuclear standby service. In our opinion, conclusive information about the behavior of these cracks could be obtained from an engine test as described above for the crankshafts, provided the cracks are characterized as to length, depth, and direction before and after the test, and appropriate strain gage measurements are taken during the test.

Operating experience with the Shoreham engines and with TDI engines at other nuclear power stations suggests that ligament cracks present in the 101 and 102 blocks between the cylinder liner conterbore and the head studs will arrest. This assumes that the material in the cylinder blocks conforms to specifications for ASTM class 40 gray-iron castings. If the ligament cracks arrest, the probability of a crack initiating between studs for adjacent cylinders and propagating into the blocks is, in our opinion, very low because of a limited driving force. However, the blocks should be monitored for this type of cracking with an appropriate nondestructive examination technique. It is difficult to predict the location of crack initiation, which conceivably could start at the threads in the holes for the head studs rather than at the surface of the block. Accordingly, the potential for subsurface cracks should be considered in the selection of the most appropriate NDE technique.

Cylinder Heads

On the basis of known operating experience with TDI heads, we have concluded that problems in service are indicative of manufacturing defects rather than design deficiencies. Subject to nondestructive examination of the firedecks of all cylinder heads at Shoreham, use of heads with no through-wall weld repairs of the firedeck, and surveillance after each time the engine is operated to detect coolant leaks into the cylinders, we have concluded that the heads are suitable for nuclear service through to shutdown for the first refueling.

Piston Skirts

On the basis of operating experience in the R-5 test engine at TDI with piston skirts similar in design to the AE piston skirts installed in the Shoreham engines, and subject to nondestructive examination of all pistons in the area of the stud bosses, we have concluded that the AE piston skirts are suitable for nuclear service through to the shutdown for the first refueling.

Based on the testimony summarized above, the NRC staff believes that these components may be qualified for nuclear standby service at Shoreham if: 1) an engine (either the 101 or 102) is tested at its rated load (either the current FSAR value or a new lower value), 2) the engine block is inspected using nondestructive techniques before and after the test to characterize the cracks in the block, and other key engine components are inspected after the test, 3) the engine block is instrumented during the test with strain gages, 4) the applicant provides additional information to resolve outstanding Staff questions concerning the crankshafts and engine blocks, and 5) the applicant performs limited destructive examinations of the old 103 engine block to resolve outstanding Staff questions concerning cracks in the blocks. The successful completion of these actions is considered to be confirmatory in nature as they are expected to provide a basis for concluding that these components are satisfactory for their intended service.

## Premises on Which This Testimony is Based

Q. Have the witnesses in this testimony identified any premises common to their evaluation of all of the contentions to be addressed?

A. (All) Yes. Our principal premise is that the TDI diesel generators at Shoreham will be reassessed at the time of the first refueling, or after about 1 1/2 years of operation. We anticipate that all phases of the Owners' Group Plan for TDI diesel generators and the plant-specific Design Review and Quality Revalidation of the Shoreham engines will be completed and implemented by the first refueling. In our opinion, it would be more appropriate to decide questions of long-term reliability and operability of the TDI diesels as that time approaches, rather than now.

Other premises on action to be taken before the engines are placed in nuclear standby service are as follows:

- nondestructive examination of all piston skirts, cylinder heads, oil holes in all crankshaft main-bearing journals, and oil holes in the most heavily loaded crankpin journals
- nondestructive examination of the top surface of each engine block to verify that no stud-to-stud cracks are present between adjacent cylinders
- preoperational crankshaft deflection tests under hot and cold conditions.

An additional premise is that, following each time an engine is operated, the engine will be rolled over with the air-start system 4 to 8 hours after shutdown, 24 hours after shutdown, and before each planned start to check for water in the cylinders.

## CRANKSHAFTS

### Contention

a. The replacement crankshafts at Shoreham are not adequately designed for operating at full load (3500 kW) or overload (3900 kW), as required by FSAR Section 8.3.1.1.5, because they do not meet the standards of the American Bureau of Shipping, Lloyd's 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.

Q. Have you reviewed the testimony filed by the County on July 31, 1984, in support of its contentions regarding the crankshafts in these proceedings?

A. Yes.

Q. How are the design rules promulgated by the various classification societies used to assure the adequate design of a diesel engine crankshaft?

A. (Henriksen, Sarsten) A number of organizations provide rules or limits for the design of diesel engines. Some of these organizations are:

- Diesel Engine Manufacturers Association (DEMA)
- American Bureau of Shipping (ABS)
- Lloyd's Registry of Shipping
- International Association of Classification Societies (IACS)
- Det Norske Veritas
- Germanischer Lloyd
- Nippon Kaiji Kyokal (NKK).

All of these organizations with the exception of DEMA have formed design rules as a guideline for the insurability of diesel engines in marine service. The design rules established by each of these organizations represent the experience of the organization on the design/analysis procedures, materials, fabrication techniques, and testing methods that would produce an adequate engine design. Because these rules were formulated by different people under different circumstances, they differ somewhat in approach and detail. The rules are often subject to, or often require, interpretation and discussion with the classification society. These societies provide a mechanism whereby a diesel engine manufacturer who comes up with a design that does not comply strictly with the societies' rules can apply for and receive approval for the design upon submission of stress analyses or other supporting data. These rules may change with time as new design techniques, materials, and fabrication methods are developed.

DEMA is an American trade association of diesel engine manufacturers. In a publication titled Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines, DEMA describes various aspects of the design, operation, and testing of diesel engines. For crankshafts, DEMA provides guidelines only for allowable stresses associated with torsional vibratory conditions. DEMA does not provide any guidance for crankshaft dimensions, material properties, or methods of fabrication.

Q. Should a crankshaft satisfy the rules or design guidelines of several classification societies?

A. (Henriksen, Sarsten) Not necessarily. There is no requirement for this. A designer may choose to follow the design rules of one or more classification societies in accordance with potential market preferences. In the case of the Shoreham engines, the applicable standard is IEEE Std 387-1977, "IEEE Standard Criteria for Diesel-Generator Units Applied as Standby Power

Supplies for Nuclear Power Generating Stations." This standard invokes DEMA Standard Practices as one of the reference standards. No otner rules or standards for the design of diesel engines are invok ᵎ.

Except for the DEMA Standard Practices referenced above, 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. In is not neccessary for good design practice that nuclear standby engines meet any of the rules established by classification societies for marine engines.

Q. Have you reviewed the County's analysis of compliance of the crankshafts with rules of the American Bureau of Shipping, as documented in Exhibit 40 of the County's testimony?

A. (Henriksen, Sarsten) Yes. We do not agree with the County's interpretation of the ABS rules regarding the section modulus of the crank webs, and we do not agree with the County's conclusion that the crankshaft dimensions do not meet the ABS rules. In our opinion, the County's interpretation is not consistent with the interpretation explained by Mr. R. Woytowich during the County's deposition of R. Woytowich, H. C. Blanding and R. A. Guiffra of ABS on July 18, 1984 (pages 129-130 of the transcript). We checked the crank web dimensions of the Shoreham crankshafts on the basis of the interpretation of Mr. Woytowich, and concluded that they do, indeed, meet the ABS requirements at both 3500 kW and 3900 kW. Our evaluation is included as Exhibit 1.

Q. Have you reviewed Dr. Simon Chen's analysis of compliance of the crankshafts with DEMA guidelines for torsional stresses, as documented in Exhibit C18 of LILCO's testimony?

A. (Sarsten) Yes, I have.

Q. What is your assessment of Dr. Chen's analysis?

A. (Sarsten) First, the program employed in Dr. Chen's analysis was limited to the vector sum of only six orders of vibration, but had been expanded to 12 orders by a special subroutine added by him. This accounts for only half of the 24 orders now normally used. Although the 12 orders include the most significant ones, the remaining 12 contribute to the accuracy of the analysis and should be considered.

Second, the harmonic coefficients ($T_n$) employed in the analysis are based upon a table appearing in Lloyd's Registry of Shipping standards ("Guidance Notes on Torsional Vibration Characteristics of Main and Auxiliary Oil Engines" 1976) rather than on values based upon actual cylinder pressure measurements taken on one of the TDI engines at Shoreham. (The latter values were used in an analysis performed by Failure Analysis Associates.) The free-end amplitude of 0.59 degrees calculated by Chen differs from the measured value of 0.693 degrees on a Shoreham engine by 14.9%. With current calculational methods, the calculated and measured values should be in much closer agreement.

Q. Have you reviewed the crankshaft analysis performed by Failure Analysis Associates (Exhibit C17 of LILCO's testimony)?

A. (Sarsten) Yes. FaAA used harmonic coefficients based upon actual measurements referred to in the previous answer. Furthermore, FaAA's computer program employed a modal superposition of an undamped system using a slight modal damping, and combined 24 excitation harmonics. FaAA concluded that the stresses meet DEMA Standard Practices, which limit stresses to less than 5000 psi for any single order of vibration and to less than 7000 psi for the summation of the orders. FaAA's results are much closer to DEMA limits than are Chen's results.

Q. Did you perform an analysis of the torsional stresses for the sum of 24 orders of vibration?

A. (Sarsten) Yes. These results are plotted in Exhibit 2. My analysis is for engine operation at 3500 kW, and employs the same $T_n$ values (TDI Owners' Group harmonic data) used by FaAA. The results are preliminary, and are subject to some slight refinements and checks. However, I anticipate that any changes in my results are unlikely to affect my conclusions to any significant degree.

For Section No. 6 of the crankshaft (i.e., the torsional spring representing the crankshaft elasticity between cylinders 5 and 6), my analysis shows that the stresses for the sum of all orders exceed the DEMA limit of 7000 psi over the entire speed range called for by DEMA, i.e., from 5% below rated speed to 5% above rated speed.

Q. Did you also calculate the stress levels for single orders?

A. (Sarsten) Yes. These results are plotted in Exhibit 3. At the rated speed of 450 rpm, the maximum torsional vibratory stress in the crankshaft occurs for the 4th order. My calculated value for this stress at 450 rpm is approximately 3800 psi. Values of this stress remain below the DEMA limit of 5000 psi throughout the speed range called for by DEMA. The rise of the 5 1/2 order above the 5000-psi limit at 95% of rated speed is not considered important, as the actual stress values so near resonance will depend upon the damping values assumed. Thus, in my view, the crankshaft does meet the DEMA requirements for single orders.

Q. By what means were these stress values computed?

A. (Sarsten) I employed a computer program called COMHOL[a] (acronym for COMplex HOLzer), which calculates the steady-state forced vibration of damped linear systems subjected to periodic forcing functions represented by a Fourier series of harmonics. The shaft and mass damping are represented by complex constants.

Q. How do your results compare with those reported by FaAA?

A. (Sarsten) The stresses that I have calculated for the sum of all orders are somewhat higher than those predicted by FaAA. For example, at the rated speed of 450 rpm, my calculations for the sum of all orders predict a torsional stress of 7096 psi in comparison to the 6626 psi predicted by FaAA. My calculations predict a front-end vibrational amplitude of 0.690 degrees in comparison to 0.662 degrees predicted by FaAA. The Stone & Webster Engineering Corporation measured a front-end amplitude of 0.693 degrees on a TDI engine at full load (as referenced in Exhibit C17 of LILCO's testimony).

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(a) Nervik, N. R., and A. Sarsten. January 1981. User's Manual: Computer Program COMHOL2 for Analysis of Forced Torsional Vibrations of Linear Damped Systems. Department of Marine Technology, The Norwegian Institute of Technology, Trondheim, Norway.

Q. How do your results compare with ABS rules?

A. (Sarsten) In addition to its requirements for crankshaft dimensions, the ABS also requires that the cyclic torsional stresses be held below specific limits that depend upon factors such as engine speed, material, etc. TDI has calculated these values for the Shoreham engines[a] and arrived at 3357 psi for a single order, and 5035 psi for total vibratory stresses as the limits that would be allowed by paragraph 34.47 of the 1984 ABS Rules. According to my calculations, these stress limits are exceeded (3608 and 7096 psi, respectively, corrected for front-end measured amplitude).

Q. Does the method of crankshaft fabrication enter into the evaluation of its adequacy?

A. (Sarsten) Yes, for some of the classification societies (e.g., Det Norske Veritas).

Q. How were the replacement crankshafts fabricated for the Shoreham engines?

A. (Bush) It is our understanding that a forged-slab, hot-twisted fabrication process was employed. PNL (S. Dahlgren) was informed during a telephone conversation with W. Coleman of the TDI Diesel Generator Owners' Group on August 9, 1984, that this process was used.

(a) These values are documented on page 21 of the report enclosed with a letter dated May 3, 1984, from ABS (R. Giuffra) to TDI (R. Yang). The letter and the applicable page of the report are included as Exhibit 4 of this testimony.

Q. How does this hot-twisted fabrication process compare with a closed forging process?

A. (Bush) A closed-forged[a] crankshaft will have isotropic properties, whereas a slab-forged and hot-twisted crankshaft will yield anisotropic mechanical properties. An appropriate heat treatment will improve the properties, but not to the degree possible with closed forging.

A more significant factor is the property gradient across the slab. In a closed forging the maximum mechanical properties exist throughout the overall surface, subject to the degree of machining. A slab-forged and twisted crankshaft will display a definite gradient in mechanical properties from centerline to surface. This means that some areas of the crankshaft will display lower properties in some regions.

The nondestructive examinations, both ultrasonic and magnetic particle, confirm there were no gross slag inclusions near the centerline, which is a positive factor.

Q. FaAA has analyzed the crankshaft safety factor for the replacement crankshaft, and arrived at the conclusion that the crankshaft was adequate. Is this not sufficient proof of adequacy?

A. (Sarsten) The failure of the original crankshaft gave a bench mark for the calculation of the factor of safety of the replacement crankshaft. The result reflects only a single point of reference. I would prefer to assess the adequacy of the crankshaft based upon the large amount of data represented by

(a) We are defining closed forging as using shaped dies to hot form the metal after an initial hot forging breakdown to homogenize the formed ingot.

the appropriate classification societies' rules and their experience in the interpretation of these rules. This should provide a conservative basis for the evaluation.

Q. Is there any way to assess the crankshaft adequacy through testing?

A. (Sarsten) Yes. One could, of course, operate the engine for a sufficient number of cycles. The figure of $10^7$ cycles is often accepted as a sufficient number in such cases. This number of cycles for a four-cycle engine at 450 rpm corresponds to around 740 total running hours. If a subsequent detailed inspection of the crankshaft fails to reveal any deleterious effects, the crankshaft could then be accepted as adequate for the load and conditions at which it had been operated for these $10^7$ cycles.

Q. Can you summarize your conclusions regarding the adequacy of the crankshaft?

A. (Sarsten, Henriksen) Based upon my (Sarsten's) analysis, the crankshafts do not meet DEMA Standard Practices regarding torsional stresses at the rated load of the engine. This does not necessarily imply that the crankshafts are inadequate for their intended service. However, from the information we have reviewed, we do not have a sufficient basis for concluding that the crankshafts are adequate.

The crankshafts do not have to meet the requirements of any or all of the classification societies for this application. On the basis of our review, we believe that they in fact do meet the requirements of ABS with regard to physical dimensions. In my (Sarsten's) opinion, they do not meet the ABS requirements regarding torsional vibration stresses.

It is also our opinion that crankshaft adequacy for a given load and conditions could be established by running a crankshaft under those conditions for $10^7$ cycles.

We believe that nondestructive tests to confirm that the crankshafts are sound should include, in addition to the tests already performed, examinations of all oil holes in main bearing journals and the oil holes in the most heavily loaded crankpin journals.

Contention

b. The shotpeening of the replacement crankshafts was not properly done as set forth by the Franklin Research Institute report, "Evaluation of Diesel Generator Failure at Shoreham Unit 1, April 5, 1984," and the shotpeening may have caused stress nucleations sites. The presence of nucleation sites may not be ascertainable due to the second shotpeening of the crankshafts.

Q. Do you believe that shotpeening of the Shoreham crankshafts was necessary?

A. (Bush) In my opinion, shotpeening of the Shoreham crankshafts probably was not necessary. The fillet radii are quite large, on the order of 0.75 inch, so the stress concentration factors at the fillets should be low. With low stress concentration factors, the probability of crack initiation by fatigue is reduced. Shotpeening of the fillet region is effective in inducing localized compression zones at and slightly below the surface to minimize local tensile or bending stresses at the fillets while undergoing cyclic loading during operation. I consider shotpeening, if done correctly, to be beneficial. Shotpeening has been performed on millions of rotating parts such as camshafts and crankshafts in automobiles, etc., as well as on many millions of springs, and the operational histories have been very good.

Q. Do you consider the original shotpeening adequate?

A. (Bush) No. Surface coverage was inadequate; furthermore, the QA records on the original shotpeening, at least those we have seen, do not yield sufficient information. A definite plus was the report of visual examination and of magnetic particle testing at LILCO that confirm that the as-received surface condition after original shotpeening at TDI was acceptable. A concern with shotpeening is shot breakage and embedment so that the surface contains many indentations. As reported, this was not the case.

Q. Do you consider reshotpeening as deleterious?

A. (Bush) No. There have been experiments on high strength steel, which would be much more susceptible to small indentations serving as stress risers for fatigue crack propagation than would be true with the lower tensile strength material in crankshafts, and there was no perceptible decrease in fatigue resistance after five shotpeening cycles. In fact, Kohls et al. (Exhibit 5) found that the fatigue resistance was enhanced with added cycles through three, and did not deteriorate with five cycles. The surface compressive layer is a major deterrent to the initiation and propagation of fatigue cracks under cyclic fatigue loads.

Q. Can shotpeening lead to a deterioration of properties below the surface, leading to internal crack propagation and ultimate failure?

A. (Bush) I doubt this would occur unless there was a large embedded flaw. This would have been detected by the extensive ultrasonic testing conducted during fabrication. I am aware of embedded flaws in structures other than crankshafts where the surface was in compression, and there has been no evidence of crack growth after 10 years, even with definite cyclic bending stresses occurring in the structure.

Q. Do you consider the reshotpeening to be adequate?

A. (Bush) Yes. According to testimony of D. Cimino (on page 11 of LILCO's testimony of C. Wells, D. Johnson, H. Wachob, C. Seaman, D. Cimino, and N. Burrell concerning Shotpeening of the Replacement Crankshafts), the shotpeening met military specification MIL-S-13165B or exceeded the specification in all critical aspects. The fluorescent penetrant test confirmed the adequacy of the shotpeening.

Q. Do you consider the argument on "nucleation sites" as significant?

A. (Bush) No. My previous comments indicate extensive use of shotpeening of automotive crankshafts, etc., and experimental evidence confirms that repetitive shotpeening is not deleterious.

Q. If the Suffolk County arguments are valid concerning the existence of "nucleation sites", could flaws of metallurgical significance be detected?

A. (Bush) We have not conducted an independent fracture mechanics analysis of the crankshaft; however, fatigue analyses on analogs of crankshafts confirm that a compression zone will minimize propagation of an existing flaw. Extensive work with the ASME XI Code indicates that flaws of significance with regard to fatigue crack propagation would approach ~0.25 inch depth and similar length in the zone influenced by shotpeening. Any flaws of this size would be detectable by magnetic particle testing. I see no reason why such flaws should exist, based on the reported nondestructive examination results.

In conclusion, I do not consider that the second shotpeening degraded the fatigue resistance. In fact, I consider that such fatigue resistance should be somewhat enhanced, but it is not quantifiable. Obviously there is a major caveat. Professor Sarsten's calculations indicate torsional stresses in excess of DEMA Standard Practices. If torsional and/or bending loads are high enough, cracks will initiate and propagate, regardless of fillet design and shotpeening. The ultimate test is to operate the crankshaft to $10^7$ cycles at the proposed power rating to see if cracks initiate.

## CYLINDER BLOCKS

### Contentions

The County contends that the emergency diesel generators (EDGs) are inadequate because:

> Cracks have occurred in the cylinder blocks of all EDGs and a large crack propagated through the front of EDG 103. Cracks have also been observed in the camshaft gallery area of the blocks. The replacement cylinder block for EDG 103 is a new design which is unproven in DSR-48 diesels and has been inadequately tested.

Q. Have you reviewed the testimony filed by the County on July 31, 1984, in support of its contentions regarding the cylinder blocks in these proceedings?

A. (Bush, Henriksen) Yes.

Q. Have you reviewed the testimony filed by LILCO[a] on August 14, 1984, which concludes that:

1. The ligament cracks present in EDG 101 and EDG 102 are benign. Observations of various engines indicate that the cracks will not propagate beyond a depth of 1-1/2 inches. Accordingly, the ligament cracks in EDG 101 and EDG 102 do not and will not impair the ability of the EDGs to perform their intended function.

2. The crack that propagated down the front of the old EDG 103 block and the cracks that developed between the stud holes of adjacent cylinders on the old EDG 103, do not threaten the integrity of EDG 101 or EDG 102. Metallurgical analysis of the existing blocks has established that EDG 101 and EDG 102 do not have the extensive degenerate graphite microstructure that produced markedly inferior fracture fatigue properties in the old EDG 103 block. Further, EDG 103 was subjected to an abnormal load excursion that contributed to further crack extension. A cumulative damage analysis predicts that the EDG 101 and EDG 102 blocks are substantially less likely to develop stud-to-stud cracking and that they will withstand a LOOP/LOCA with sufficient margins, even if they were to initiate stud-to-stud cracking during a LOOP/LOCA.

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(a) Testimony of R. McCarty, C. Rau, C. Wells, H. Wachob, D. Johnson, R. Taylor, C. Seaman, E. Youngling and M. Schuster on Suffolk County Contention Regarding Cylinder Blocks.

3. The cam gallery cracks in the Shoreham EDGs, which were discovered more than 1-1/2 years ago, are not predicted to propagate significantly even after hundreds of hours of engine operation. In addition, there is no reported incident in which cam gallery cracks have caused a sudden engine failure. The cam gallery cracks are, therefore, not predicted to impair the ability of the EDGs to meet their intended function.

4. The replacement block for EDG 103 has been tested adequately. The replacement block is not a new design. It is simply a current production model that incorporates certain product enhancements, each of which has been shown to be beneficial by exhaustive testing in the R-5 engine.

and, further[a], that:

1. The ligament cracks present in EDG 101 and EDG 102 are benign. There is no evidence that the cracks will propagate beyond a depth of 1-1/2 inches. Accordingly, the ligament cracks in EDG 101 and EDG 102 do not and will not impair the ability of the EDGs to perform their intended function.

2. The crack that propagated down the front of the old EDG block and the large cracks that developed between the stud holes of adjacent cylinders on the old EDG 103, do not threaten the integrity of EDG 101 or EDG 102. TDI believes that EDG 103 was subjected to abnormal high stress as a result of an unusual load excursion and that this caused additional extensive cracking in EDG 103.

3. The cam gallery cracks in the Shoreham EDGs were discovered more than 1-1/2 years ago. These cracks have not propagated significantly despite hundreds of hours at full load and overload conditions. It is TDI's opinion that the cam gallery cracks will not propagate significantly and that they will not impair the ability of the EDGs to meet their intended function.

4. The replacement EDG 103 block has been adequately tested. The replacement block is not a new design. It is simply a current production model that incorporates a few product enhancements, each of which has been shown to be beneficial by exhaustive testing in the R-5 engine.

A. (Bush, Henriksen) Yes.

(a) Testimony of C. Mathews, M. Lowrey, and J. Wallace.

Q. Please summarize your conclusions regarding the cylinder blocks.

A. In summary, we conclude that:

- Presently, the information regarding the cracks in the camshaft gallery on the cylinder blocks for EDG 101 and EDG 102 is incomplete. Consequently, no conclusion can be made as to the suitability of these two cylinder blocks for the operation stated.
- The replacement block for EDG 103 is not a new design; it has been proven. Further, if it is certified to be free of stud-to-stud cracks between adjacent cylinders and in the camshaft gallery and if it is inspected for cracks after each operation, it will be suitable for nuclear service for one refueling cycle.

Q. Do you know the material specifications for the cylinder blocks on the Shoreham TDI 101 and 102 engines?

A. (Bush, Henriksen) Yes. Drawing #03-315-03-AC of the cylinder blocks for the Shoreham TDI 101 and 102 engines specify an ASTM-A48-64 class 40, gray-iron casting.

Q. Was the material specification for the original cylinder block on the Shoreham TDI 103 engine also ASTM-A48-64 class 40, gray-iron casting?

A. (Bush, Henriksen) Yes.

Q. What are the material specifications for the replacement cylinder block on the Shoreham TDI 103 engine?

A. (Bush, Henriksen) Drawing #03-315-05-AD of the cylinder block for the Shoreham TDI 103 engine specifies an ASTM-A48-76 class 45B, gray-iron casting.

Q. What is the significant difference between an ASTM-A48-64 class 40, gray-iron casting and an ASTM-A48-76 class 45B, gray-iron casting?

A. (Bush, Henriksen) The tensile and yield strengths of an ASTM-A48-76 class 45B, gray-iron casting are superior to those of an ASTM-A48-64 class 40, gray-iron casting.

Q. Have you reviewed the portion of the FaAA report that deals with the metallurgical analysis performed on cylinder blocks of the Shoreham TDI 101, 102, and 103 engines?

A. (Bush, Henriksen) Yes.

Q. Do you consider the quality of the gray iron in the original cylinder block of the Shoreham TDI 103 engine typical of standard casting practice?

A. (Bush) No. The morphology of the graphite flakes, as evidenced from the photomicrographs presented, was not typical. Such flakes would lead to degraded mechanical properties.

Q. Did you find the quality of graphite in the cylinder blocks from the TDI 101 and 102 engines similar to that in the original block from the 103 engine?

A. (Bush) No. The microstructure of the samples from the cylinder blocks of the 101 and 102 engines is typical for an ASTM class 40, gray-iron casting.

Q. Have you reviewed the portion of the FaAA report that deals with the physical tests that were performed on samples from the cylinder block of the Shoreham TDI 103 engine?

A. (Bush, Henriksen) Yes.

Q. What did you conclude from your review?

A. (Bush, Henriksen) That the results from the physical test confirm the conclusion drawn from the metallurgical analysis. The material in the original cylinder block from the Shoreham TDI 103 engine is substandard as compared to ASTM class 40, gray-iron castings.

Q. Can it be assumed that, since the photomicrographs indicate that the cylinder blocks from engines 101 and 102 indicate typical class 40, gray-iron castings, their physical properties such as tensile and yield stresses are, in fact, typical of class 40, gray-iron castings?

A. (Bush, Henriksen) The assumption may certainly be made that the material in the cylinder blocks for engines 101 and 102 is superior to the material in the original 103 cylinder block. Whether or not the 101 and 102 blocks actually have the physical properties of class 40, gray-iron castings can be confirmed only by actual tests. We have no knowledge that this testing was ever done.

Q. Assuming that the material in the cylinder blocks for engines 101 and 102 conforms to the specifications for ASTM class 40, gray-iron castings, would you consider the ligament cracks presently observed in the blocks between the cylinder liner counterbore and the cylinder head studs as benign?

A. (Bush) The empirical evidence would indicate that these cracks grow to the size cited, then arrest. This empirical evidence is based on repetitive examinations of cracks in both ship and stationary diesels. There is one substantial difference between such diesels and emergency diesels tested periodically. Basically, the first group operates at near steady-state conditions, whereas the emergency diesels will reach peak loads rapidly and operate with variable thermal gradients. Because of this difference, one cannot unequivocally state that the cracks will arrest. A definitive three-dimensional finite element analysis with valid load inputs through the thickness of the block, covering hoop stresses, thermal loads, bolting loads, etc., would confirm whether the crack has arrested because of a rapidly decreasing stress gradient.

Q. If the ligament cracks from cylinder liner to studs could be shown to have been arrested, what, in your opinion, would be the probability of a crack initiating between studs of adjacent cylinders?

A. (Bush) If the liner/stud crack can be shown to have arrested, the probability of a crack initiating between the two studs and then propagating into the block is very low because there is a limited driving force. The initial cracks in the 103 block are believed to be due to the degraded mechanical properties; the very severe overloads because of the load transient are believed to have caused rapid crack growth. In essence, this would correspond to a low-cycle fatigue problem where every cycle drives the crack a substantial distance.

Q. In your opinion, will the ligament cracks presently observed between the counterbore and the studs render the cylinder blocks on engines 101 and 102 unsuitable for nuclear service?

A. (Bush) The nature of the loss of power/loss of coolant accidents is such that demand for high diesel generator-related power is quite short-lived; thereafter, the power demands are much less. Even if the diesel generators were to be derated and it became necessary to meet LOOP/LOCA conditions above the derated rating but no higher than the nameplate rating, the limited duration at higher power should not pose a major problem.

Q. Do you consider checking for cracks between studs of adjacent cylinders after each operation above 50% load as adequate?

A. (Bush, Henriksen) No. As stated earlier, we do not have an adequate basis for concluding that all present cracks are arrested. Therefore, we feel this inspection should be performed after any operation.

Q. Do you consider the suggested eddy-current test as adequate to detect cracks of sufficient size to lead to detorquing of the studs?

A. (Bush) It must be recognized that the eddy-current test with ferritic materials is limited to the "skin" of the metal. All testing of the block surface must be done through the restricted access between cylinder heads. Although eddy-current testing will be difficult, it is not impossible, provided the surface between the two studs is sufficiently smooth (i.e., a machined surface).

The more fundamental issue is the initial locus of crack initiation. The most probable location would be between stud hole and cylinder, which is impossible to examine without disassembly. In my opinion, on the basis of a limited review, the most probable location for cracks to initiate would be at the corner of the counterbore at the start of the threads. Depending on the stress distribution, such a crack could progress down the threads or up to the surface. Based on LILCO testimony for blocks 101, 102, and the original 103 plus blocks for other TDI diesels, cracks exist at the surface and to depths of 1.5 inches. It is possible that the liner/stud cracks might grow down the threads under the start-stop loading typical of emergency diesels. If this occurred, there could be a redistribution of stresses so that cracks may initiate between the studs. We suspect that such cracks would initiate at the corner adjacent to the top thread. However, unless the cracks propagate to the surface, eddy-current testing will be useless. An alternative technique that might work is a zero degree ultrasonic wave commonly used in metals as a depth gage. If the external surface area and geometry are adequate to insert the ultrasonic tranducers, cracks between the studs have the potential of detection. This technique has the advantage of measuring the depth dimension whether the crack reaches the surface or remains subsurface.

Q. Mr. Berlinger, do you agree with the previous response?

A. (Berlinger) Not completely. With regard to the issue of crack initiation sites, limited hard evidence has been submitted by LILCO in their exhibits B-16, B-17, B-18 and B-25. These crack maps indicate that some block cracks which extend down into the block from the block top surface had not been observed to the depth of the stud threads (1 1/2 inches). Conversely, no cracks have been observed at the depth of the threads which did not extend up to the block top surface.

FaAA and LILCO have stated during recent technical discussions that they have used eddy current probes to inspect stud counterbore and thread areas in stud holes in the 101, 102 and old 103 Shoreham blocks. In those cases for which no surface crack indications had been observed, these inspections did not find any subsurface cracks. These measurements/inspections would confirm that cracks which would initiate below the surface would propagate and be evidenced at the block top surface.

The staff believes that it is difficult to predict the locations of crack initiation, and that the potential exists for crack initiation in the block stud area from subsurface initiation sites (e.g., stud threads). However, the evidence from previous inspections of the Shoreham cylinder blocks would indicate that crack initiation would not be subsurface. Therefore, monitoring of the block top surface for stud-to-stud cracks should be done using the most appropriate nondestructive examination technique which should not be limited to consideration of only ultrasonic techniques.

Q. Do you consider the position suggested by LILCO that stud-to-stud cracks to depths of 1.5 inches are acceptable as justified?

A. (Bush) No. The only basis for such a position is believed to be the existence of stud-to-stud cracks in the original 103 block. Cracks of unkown geometry were known to exist prior to the severe overload that drove a crack to a depth exceeding 5 inches. As noted previously, we believe the probability of stud-to-stud cracks is very low, assuming the cast iron is not atypical as was the case with the original 103 block.

The appearance of a stud-to-stud crack in normal quality cast iron would indicate that too little is known concerning the stresses and stress distributions leading to such a crack. A deliberate decision to continue operation without repair of such a crack is not justified because the presence of such a crack indicates that the current analytic techniques do not accurately model crack initiation and growth.

If a well designed three-dimensional finite element analysis using stresses validated by experimental methods were conducted, it might be possible to justify the conscious operation with stud-to-stud cracks. Personally, I doubt it, because of difficulty in establishing local stresses.

Q. Have you had occasion to review the LILCO testimony and exhibits referring to the cracks in the camshaft gallery?

A. (Bush, Henriksen) Yes.

Q. Based on this testimony and relevant exhibits, have you formed an opinion as to why these cracks initiated in the first place?

A. (Bush, Henriksen) No. We believe this point has not been addressed in the testimony or the exhibits.

Q. Have you formed an opinion as to crack growth rate in the camshaft gallery based on FaAA's analysis on this subject?

A. (Bush, Henriksen) No. The FaAA analysis approach probably is correct, provided the input data are correct. However, we have some reservations as to the correctness of the strain gage data supplied by TDI. These data constitute the main basis for the FaAA analysis.

Q. Is your concern regarding the TDI strain gage data related to the fact that the data were obtained from a 6-cylinder rather than an 8-cylinder engine, a slightly larger fuel injection pump, and a little faster rising fuel cam?

A. (Bush, Henriksen) No. Those are minor issues of no consequence.

Q. What is your concern then?

A. (Bush, Henriksen) First, referring to LILCO Exhibit B54, Gage #1 is not located in the area in question; yet the values obtained from Gage #1 are presented in the testimony as the stresses found in the cracked area.

Second, again referring to LILCO Exhibit B54, Gages #2 and 3 appear to be located in the same area. As can be noted in LILCO Exhibit B53, there is a difference of over 50% at 110% load, and over 100% at 100% load in mean stress between the two gages.

Third, and most important, we do not understand how, for the same mode of operation, the stresses can change from tension to compression as a function of engine load. The fuel injection pump is positively loaded every second revolution regardless of load. The vectors in the loading diagram do not change direction as a function of load. Thus, in our opinion, the stresses should not change direction as a function of load.

Q. In your opinion, do the cracks in the cam gallery pose a potentially serious problem?

A. (Bush, Henriksen) Yes. Depending upon the depth of the cracks and the anticipated growth pattern, the cracks may or may not pose future problems. Examination of TDI drawing #03-315-03-AC indicates that cracks may possibly propagate into the cylinder cooling water space, which could result in water entering into the camshaft housing. Lube oil in that housing drains into the engine crankcase. Leakage in this area is unlikely to be noticed during engine operation. Thus, enough water may mix with the lube oil in the crankcase to cause serious damage to bearings, shafting, etc.

Q. In your opinion, do the cracks in the camshaft gallery of the cylinder blocks for engines 101 and 102 render these engines unsuitable for nuclear service for one refueling cycle?

A. (Bush, Henriksen) Yes, until the questions raised regarding the TDI strain gage measurements and the reversal of direction of stresses are answered such that we have a reasonable assurance that the cracks in the cam gallery are benign or grow at such a slow rate that they are of no concern.

Q. Mr. Berlinger, does the staff believe that the concerns, relative to the cracks in the camshaft gallery can be resolved?

A. (Berlinger) Yes, the staff believes if an engine were tested as suggested to resolve the concerns regarding the crankshafts, that data obtained during that testing could provide information regarding the stresses and crack propagation in the cam gallery area.

Assuming that either EDG 101 or 102 was to be tested, if the cam gallery area were thoroughly inspected to characterize the existing cracks by determining the length, depth and direction of existing cracks before and after the suggested $10^7$ cycle test, and, if the crack area were instrumented with strain gages and measurements were taken during these tests, the staff believes that conclusive information about the behavior of the cracks could be obtained which would resolve the existing concerns.

Q. In your opinion, is the replacement cylinder block for EDG 103 of a new design?

A. (Henriksen) No. Drawing #03-315-05-AD indicates that the replacement cylinder block is a modified version of the original cylinder block drawing #03-315-03-AC.

Q. Other than the change in material, which you have stated earlier was an improvement, have you reviewed LILCO's testimony with regard to the other changes to the replacement cylinder block?

A. (Henriksen) Yes.

Q. Do you consider any of these changes or modifications detrimental?

A. (Henriksen) No.

Q. Do you consider any of these changes or modifications beneficial?

A. (Henriksen) Yes. All changes to the replacement cylinder block, as listed in LILCO's testimony, are considered beneficial.

Q. Do you have any remarks regarding any of the changes or modifications?

A. (Henriksen) Yes. LILCO's testimony indicates that the replacement block has a greater cold clearance gap between the cylinder liner and the cylinder block. This change is not reflected in block drawing #03-315-05-AD. However, we understand from a TDI (R. Johnston) letter dated May 4, 1984, to Stone & Webster Engineering Corporation that TDI has recommended this change be made to the cylinder liners. (The TDI letter is Exhibit 6 of this testimony.)

Q. As a design, do you believe the EDG 103 replacement cylinder block inadequately proven?

A. (Henriksen) No. We have compared drawing #03-315-05-AD of the replacement cylinder block with drawing #02-315-05-AW, which depicts the cylinder block for the R-5 prototype test engine. We found that, in the area affected by the changes, with the exception of the dimension regarding the cold gap clearance as mentioned earlier, the two drawings indicate the two cylinder blocks appear to be exactly alike. The R-5 cylinder block has been extensively tested at a load level higher than the EDG 103 will ever experience. Thus, we believe that, provided the R-5 cylinder block did not develop cracks during its extensive testing, as a design the EDG 103 cylinder block has been proven.

Q. Does the fact that the R-5 is a V-engine and the EDG 103 is an inline engine in any way enter into your evaluation when comparing the two cylinder block designs?

A. (Henriksen) Yes. However, for the area of interest there is no difference in cylinder block design between a V-engine and an inline engine.

Q. Have you drawn any final conclusion regarding the EDG 103 replacement cylinder blocks?

A. (Henriksen) Yes. Provided preoperational inspection reveals no cracks between studs from adjacent cylinders or in the camshaft gallery, and provided inspections for cracks are conducted after each operation, the EDG 103 replacement cylinder block is considered suitable for operation through to shutdown for the first refueling.

## CYLINDER HEADS

### Contention

The replacement cylinder heads on the Shoreham EDGs are of inadequate design and manufacturing quality to withstand satisfactorily thermal and mechanical loads during EDG operation, in that:

a. the techniques under which the replacement cylinder heads were produced have not solved the problems which caused the cracking of the original cylinder heads on the Shoreham EDGs;

b. the "barring over" surveillance procedure to which LILCO has committed will not identify all cracks then existing in the replacement cylinder heads (due to symptomatic water leakage);

c. the nature of the cracking problem and stresses exacerbating the cracks are such that there can be no assurance that no new cracks will be formed during cold shutdown of the EDGs;

d. there can be no assurance that cracks in the replacement cylinder heads and concomitant water leakage occurring during cold shutdown of the EDGs (which would not be detected by the barring-over procedure) would not sufficiently impair rapid start-up and operation of the EDGs such that they would not perform their required function;

e. there can be no assurance that cracks in the replacement cylinder heads occurring during operation of the EDGs would not prevent the EDGs from performing their required function;

f. variations in the dimensions of the firedeck (and waterdeck) of the replacement cylinder heads create inadequate cooling, where too thick, and inadequate resistance to mechanical loads, where too thin, and create stress risers at their boundaries;

g. the design of the replacement cylinder head is such that stresses are induced due to non-uniform bolt spacing [and the different lengths of the bolts];

[h. the replacement cylinder head design does not provide for adequate cooling of the exhaust valves;]

i. at least one replacement cylinder head at Shoreham has an indication;

[j. the design of the replacement cylinder heads provides inadequate cooling water for the exhaust side of the head];

k. the replacement cylinder heads at Shoreham were inadequately inspected after operation, because:

1. a liquid penetrant test was done on the exhaust and intake valve seats and firedeck area between the exhaust valves on only nine of 24 cylinder heads, and such tests were done after only 100 hours of full power operation;

2. ultrasonic testing was done on the firedeck areas of only 12 cylinder heads;

3. visual inspections were performed on the valve seat areas of only 32 of the 98 valves, and on only seven firedecks of the 24 cylinder heads for indications of surface damage.

Q. Have you reviewed the testimony filed by the County on July 31, 1984, in support of its contentions regarding the cylinder heads in these proceedings?

A. (Henriksen, Sarsten) Yes.

Q. Are there any portions of the County's contentions that are not addressed in the County's testimony?

A. (Henriksen, Sarsten) Yes. The bracketed portions were not addressed in the County's testimony, as noted on page 61 of that testimony.

Q. Have you reviewed the testimony filed by LILCO on August 14, 1984, which concludes that:

1. There is reasonable assurance that leakages will not occur in the new cylinder heads because of: (i) improved casting techniques, (ii) the application of stress-relief techniques, and (iii) additional and more frequent inspections of the heads.

2. The replacement cylinder heads are adequately designed, since (i) the ranges and dimensions of the firedeck provide for adequate cooling of the firedeck and adequate resistance to mechanical loads; (ii) stress risers are not created at their boundaries; and (iii) non-uniform bolt spacing has no effect on stresses in the cylinder head.

3. The successful operating history of the new heads demonstrates that the new heads should not develop leaks.

4. Even if cylinder head leakage should occur during operation of the engine, it will be detected.

5. Leakage will not initiate after shutdown because leaks of cylinder heads will not develop when the diesel engines are in a standby condition and, in any event, such leakage would be detected by LILCO's barring-over procedure.

6. Even if leakage of the cylinder heads were to develop during standby or go undetected during operation, resultant leakage will not impair the rapid start capability of the diesels.

7. Even in the unlikely event that a new cylinder head were to leak during operation, the leakage will not impair the operation of the diesel engines.

8. None of the replacement cylinder heads at Shoreham has any relevant indications.

9. The replacement cylinder heads were adequately inspected because the heads were subjected to (i) a 100% factory inspection by TDI which was audited by LILCO, (ii) additional pre-operational inspection by the NRC, and (iii) post-operational inspections including liquid penetrant tests on 10 cylinder heads, ultrasonic testing on 13 firedeck areas, and visual inspections on 7 firedecks.

A. (Henriksen, Sarsten) Yes.

Q. Please summarize your conclusions regarding the cylinder heads.

A. (Henriksen, Sarsten) A summary of our conclusions is as follows:

- On the basis of known operating experience with TDI heads, problems in service are indicative of manufacturing defects rather than design deficiencies. Of course, the design of the heads affects the complexity of manufacture, which, in turn, affects the capability of the foundry to produce castings free of unacceptable defects. However, operating experience does not suggest that the design itself is inherently deficient.

- In the absence of further evidence of their reliability, cylinder heads with any through-wall weld repair of the firedeck should not be placed in nuclear standby service if the weld repair is performed from one side only. The coolant side of the firedeck is not readily accessible for weld repair. Without such access, a repair from the combustion side might leave defects on the coolant side that would be difficult to detect, and that might compromise the integrity of the head.

- The following inspections should be completed at Shoreham on all cylinder heads before the engines are placed in nuclear standby service:
  - ultrasonic inspection of the entire firedeck to verify that the minimum thickness requirement (0.400 inch) is met
  - surface inspection (i.e., liquid penetrant or magnetic particle) of the firedeck and the valve seats to verify that they are free of unacceptable surface defects.
- Each time an engine is operated, it should be rolled over with the air-start system to detect for coolant leaks into the cylinders at least 4 hours, but not more than 8 hours, after engine shutdown. A second rollover should be performed in the same manner approximately 24 hours after shutdown. In addition, the engine should be rolled over immediately prior to any planned start.
- Subject to the above comments on weld repairs, inspection, and surveillance, the cylinder heads are considered to be adequate for nuclear service for one refueling cycle.

Adequacy of Design and Manufacture

Q. Have you reviewed the evaluation by Failure Analysis Associates of thermal and pressure stresses in the cylinder heads, which is presented in the FaAA report titled Evaluation of Cylinder Heads of Transamerica Delaval Inc. Series R-4 Diesel Engines?

A. (Henriksen, Sarsten) Yes.

Q. Have you reached any conclusions on the basis of FaAA's evaluation?

A. (Henriksen, Sarsten) No. The flat-plate model used in the FaAA evaluation of thermal and pressure stresses does not, by itself, provide an adequate basis for confirming the design adequacy of the cylinder head, which is much more complex than the model.

Q. Have you any reason to believe that the replacement cylinder head design provides inadequate cooling of the exhaust valves?

A. (Henriksen, Sarsten) No. There is no failure history in evidence to support this claim.

Q. Have you any reason to believe that the replacement cylinder heads provide inadequate cooling water for the exhaust side of the head?

A. (Henriksen, Sarsten) No. There is no failure history in evidence to support this claim.

Q. Have you reviewed the component history of the R-4 cylinder head presented in the FaAA report on this component and in the LILCO testimony?

A. (Henriksen, Sarsten) Yes.

Q. What conclusions have you drawn from that review?

A. (Henriksen, Sarsten) We concur that the changes made in the manufacture of the cylinder heads from those classified as "Group I" (cast prior to October 1978) through "Group III" (cast after September 1980) should improve the reliability of the heads. Operating experience cited in these reports confirms that the changes have made the heads more reliable.

Q. From which group are the heads that are currently installed at Shoreham?

A. (Henriksen, Sarsten) We understand from the LILCO testimony and the FaAA report that all of the heads currently installed are from Group III.

Q. Do you believe that the operating experience is sufficient to conclude that Group III heads will not develop cracks through which coolant could leak into the cylinders?

A. (Henriksen, Sarsten) No. While we agree with LILCO that the Group III heads are superior to heads from Groups I and II, we do not believe that the operating experience with Group III is sufficient to demonstrate that leakage cracks are unlikely to form. Therefore, for precautionary purposes, it is our opinion that the heads should be checked for leakage via the rolling-over procedure described in the summary of our conclusions on this component.

<u>LILCO'S "Barring Over" Surveillance Procedure</u>

Q. Have you reviewed LILCO's procedure for barring-over the engine?

A. (Henriksen, Sarsten) We have reviewed LILCO's Exhibit H24 titled "Emergency Diesel Generator Cylinder Head Leak Detection Test," SP Number

27.307.02, Rev. 2, dated January 18, 1984. This procedure calls for turning the engine over 4 hours and 12 hours after shutdown, using the barring-over device. It also calls for turning the engine over 24 hours after shutdown, using the air-start system.

Q. Do you consider this procedure adequate?

A. (Henriksen, Sarsten) No. Water may not be detected when the engine is rolled with the barring-over device, because of the slow rotational speed. The air-start system rolls the engine much more rapidly, and is mandatory for detection of water leakage because the higher rate of compression will vaporize any water in the cylinder and the vapor will be very noticeable when it is expelled through the indicator cocks. Therefore, we recommend that the surveillance for water leakage be conducted only with the air-start system.

Crack Formation During Cold Shutdown

Q. What is your opinion on the propagation of cracks in a cylinder head and/or the formation of new cracks during cold shutdown?

A. (Henriksen, Sarsten) If a crack, through which water could leak into a cylinder, does not open sufficiently for the water to be detected 24 hours after engine shutdown, it is highly unlikely, in our opinion, that the crack will propagate to the degree that water would leak before the next engine startup. Similarly, we believe it is highly unlikely that a new crack that might leak water would remain undetected after 24 hours and then leak before the next startup. However, we recommend that the engine be rolled over to detect water leakage immediately preceding any planned start, to ensure that no leakage has occurred.

Q. Do you believe that corrosion products in a cylinder head crack could cause the crack to propagate or grow after engine shutdown?

A. (Henriksen, Sarsten) No. We agree with LILCO's testimony on this subject (Volume 1, page 77 of LILCO's testimony "...Regarding Cylinder Heads on Diesel Generators at Shoreham") that corrosion products within a crack will tend to plug the crack, and that no technical foundation exists to suggest that low-strength carbon steels (of the type used in the Shoreham cylinder heads) are susceptible to corrosion product crack wedging.

## Effects of Undetected Leakage on Rapid Start Capability

Q. Could leakage undetected by the barring-over procedure affect rapid start capability?

A. (Henriksen, Sarsten) It is our opinion that, if rolling-over procedures are performed with the air-start system, leakage undetected after 24 hours will not be sufficient to impair rapid start capability. In a test described by LILCO on page 83 of the above-referenced testimony and documented in Exhibit H-26 of that testimony, water in an amount that occupied 98% of the clearance volume of the piston was intentionally placed in a cylinder. According to LILCO, this water did not affect rapid start capability, nor did it adversely affect the head studs or gaskets. This reinforces our opinion that leakage undetected as described above will not impair rapid startup.

Effects of Cracks Occurring During Engine Operation

Q. Could cracks that might occur during engine operation prevent the EDGs from performing their required function?

A. (Henriksen, Sarsten) No. Since the cylinder pressure far exceeds the water jacket pressure, in the event a crack were to develop during engine operation it is unlikely that coolant would enter the cylinder. It is much more likely that the combustion gases would leak into the coolant. This would cause noticeable pulsations in the coolant pressure, which would be noticeable on the applicable gage in the control room but would not lead to engine shutdown nor impair engine performance. In the very unlikely event coolant were to leak into the cylinder during operation, it would be turned to vapor and exit with the exhaust gases. Accordingly, it is highly unlikely that coolant would leak in any amount that would impair lubrication in the cylinder or cause seizure or fracture of the piston.

Effects of Variations in the Dimensions of the Firedeck

Q. Are the maximum firedeck thicknesses measured on the Shoreham cylinder heads large enough to cause inadequate cooling?

A. (Henriksen, Sarsten) It is our understanding that the maximum firedeck thickness measured on the Shoreham cylinder heads is 0.881 inch. Since the thermal resistance of the metal is not the controlling thermal resistance for the combustion gas-to-water-side heat transfer, the reported overthickness of the firedecks will have no significant effect on the amount of cooling.

Q. Are the minimum firedeck thicknesses measured on the Shoreham cylinder heads small enough to create stress risers at their boundaries?

A. (Henriksen, Sarsten) The minimum firedeck thickness reported at Shoreham is 0.460 inch in an area of nominal 0.500-inch thickness. This 8% decrease in thickness is not felt to cause unacceptable reduction in cylinder head strength or stiffness. None of the reported failures for Group II and III cylinder heads indicates that the reduction in strength due to reduced thickness is a concern.

Stresses Induced Due to Nonuniform Bolt Spacing

Q. In your opinion is the nonuniform bolt spacing on the cylinder heads likely to create any serious problems?

A. (Henriksen, Sarsten) No. Nonuniform cylinder head bolt spacing is common practice for most diesel engine manufacturers building both V and inline engines of identical bore and stroke. There is no evidence that the nonuniform bolt spacing has been the cause of any damage of the kind that would necessitate an engine to be shut down.

Indication Found in a Replacement Cylinder Head

Q. Is the 3/8-inch long indication found in cylinder head S/N H-34 in an area of concern?

A. (Henriksen, Sarsten) No. Referring to LILCO Exhibit H15, the 3/8-inch indication found on cylinder head S/N H-34 is located in one of the plates welded onto the side of the head and not in the firedeck. Even if this indication were to propagate through the plate, it could not provide a path for leakage of coolant into the cylinder.

Adequacy of Inspection of Replacement Cylinder Heads After Operation at Shoreham

Q. Did you review the testimony regarding the nondestructive testing performed on the cylinder heads at Shoreham after 100 hours of operation at full load?

A. (Henriksen, Sarsten) Yes.

Q. The County contends that the replacement cylinder heads at Shoreham were inadequately inspected. Do you agree?

A. (Henriksen, Sarsten) Yes, but only because not all of the cylinder heads were inspected.

Q. Do you agree with the testimony of Youngling, Seaman, Kammeyer, and Wells (Vol. 1, page 94 of LILCO's testimony "...Regarding Cylinder Heads on Diesel Generators at Shoreham") that "...results of these inspections provide the required level of assurance that operational stresses will not induce cracking, and support FaAA's conclusions that the cylinder heads at Shoreham are qualified for unlimited operation."?

A. (Henriksen, Sarsten) Not entirely. The 100 hours of operation at full load was surely a step in the right direction. However, adequacy of an unverified design must be established using a data base of many hundreds or thousands of hours. The testing at Shoreham is only one point in that data base.

Q. What then, is the conclusion you can draw from this testing?

A. (Henriksen, Sarsten) The principal conclusion is that there is a high probability that there are currently no cracks that could leak water into the cylinders.

Q. Could cracks develop?

A. (Henriksen, Sarsten) Perhaps, but without the existence of an adequate data base it would be impossible to say definitively. However, if such cracks would occur, they would most certainly be detected by the proposed barring-over procedure.

Q. What further testing should be done, then, to qualify these heads for unrestricted operation?

A. (Henriksen, Sarsten) A statistical sampling inspection program should be established that would build up a data base over several thousand hours of operation. These inspections could be performed on the Shoreham engines and other TDI engines in nuclear service after each fuel cycle or during other maintenance periods.

Q. Have you drawn any final conclusions regarding the cylinder heads on the three EDGs at Shoreham?

A. (Henriksen, Sarsten) Yes. Provided preoperational inspections of the firedecks of all cylinder heads reveal no significant indications, the heads used have no through-wall weld repairs of the firedeck, and proposed surveillance procedures using the air-start system are followed to detect coolant leakage into the cylinders after each time an engine is operated, we conclude that the cylinder heads on the three EDGs at Shoreham are suitable for operation through to the shutdown for the first refueling.

## PISTON SKIRTS

### Contentions

All AE piston skirts in the EDGs were replaced with TDI model AE piston skirts. The replacement AE pistons are of inadequate design and manufacturing quality to satisfactorily withstand operating conditions because:

a. The FaAA report conclusion that cracks may occur but will not propagate improperly depends on a fracture mechanics analysis of an ideal situation which is not valid for the actual conditions which may be experienced by the Shoreham diesels.

b. Excessive side thrust load, which could lead to catastrophic failure, has not been considered adequately, and

c. The analysis does not adequately consider that the tin-plated design of the pistons could lead to scoring causing excessive gas blow-by, and, therefore, causing a failure of proper operation.

Q. Have you reviewed the testimony filed by the County on July 31, 1984, in support of its contentions regarding the pistons in these proceedings?

A. (Henriksen, Sarsten) Yes.

Q. Have you reviewed the testimony filed by LILCO August 14, 1984, on AE piston skirts which concludes that:

1. The FaAA conclusion that cracks may or may not initiate in the AE piston skirts, but if initiated, will not grow, is based on crack initiation and growth analyses considering the important loads and displacements reflected in the actual operating conditions to be experienced by the Shoreham EDGs.

2. Actual operating experience shows no relevant indications in AE piston skirts.

3. The side thrust load on the AE piston skirts is not excessive. Side thrust is not a design or operation problem with the AE piston skirt.

4. The tin-plated design of the AE piston skirt is intended to act as a protective covering for the piston skirt and is not the source of any excessive scuffing that could lead to failure. No known failures of pistons have been caused by tin plating.

A. (Henriksen, Sarsten) Yes.

Q. Please summarize your conclusions regarding the piston skirts.

A. (Henriksen, Sarsten) Our testimony, in summary, is that, provided a 100% inspection proves the piston skirts to be free of defects, the piston skirts will be suitable for operation through to shutdown for the first refueling.

Q. Have you reviewed the FaAA analysis (FaAA-84-2-14 dated May 23, 1984, included as Exhibit 8 of the County's testimony) which concludes that AE piston skirts may or may not develop cracks, but if cracks initiate, they will not propagate?

A. (Henriksen, Sarsten) Yes.

Q. Have your drawn any conclusions from your review of the FaAA analysis with regard to crack initiation in the piston skirts?

A. (Henriksen, Sarsten) No. The area in question is of intricate design, and some of the determining values, although claimed to be conservative, are admittedly assumed. As stated in the conclusions of the FaAA report (page 8-1), the analysis is inconclusive as to whether cracks will initiate or not.

Q. Have you drawn any conclusions from your review of the FaAA analysis with regard to crack growth if cracks are initiated in the piston skirts?

A. (Henriksen, Sarsten) No. Since the analysis of crack growth is based largely on the same input data as was the crack initiation analysis, we have been unable to draw a firm conclusion regarding whether or not cracks that might initiate will grow.

Q. Have you reviewed the operating experience presented by LILCO as relevant to AE piston skirts?

A. (Henriksen, Sarsten) Yes. We have reviewed LILCO's testimony and the FaAA report on this subject.

Q. Have you had occasion to personally see some AE piston skirts that have been in operation?

A. (Henriksen) Yes. I had occasion to see AE piston skirts at both Grand Gulf (16 skirts) on June 4 and June 5, 1984, and Shoreham (8 skirts) on May 23, 1984.

Q. To the best of your knowledge, did any of the piston skirts you have seen show any indication of cracks?

A. (Henriksen) No. At Shoreham the pistons were assembled, so the contested areas could not be viewed. At Grand Gulf, however, I viewed all 16 pistons of the Division I engine, and no indications were evident. These 16 pistons had been inspected earlier, and no indications were reported.

Q. Do you consider all evidence of operating experience with AE piston skirts of equal importance?

A. (Henriksen, Sarsten) All evidence of operating experience is important. However, some is more relevant than others. For instance, experience obtained at a high load level is obviously more relevant to future operation at Shoreham than experience obtained at a relatively low load level.

Q. Is there any piston skirt experience presented that is of special significance for the process of evaluating the performance of the AE piston skirts?

A. (Henriksen, Sarsten) Yes. According to LILCO's testimony on piston skirts (Vol. 1, page 56), two AE skirts from the TDI prototype R-5 engine were inspected by FaAA after approximately 622 hours of operation at 2000 psi maximum cylinder pressure. The inspections revealed no relevant indications.

Q. What, if any, are the differences between the operating parameters of the R-5 engine and the Shoreham engines?

A. (Henriksen, Sarsten) Other than the higher load of the R-5 engine, the only other difference we are aware of is that the R-5 was operated at 514 rpm, while the Shoreham engines are operated at 450 rpm.

Q. What effect does the difference in rpm between the R-5 engine and the Shoreham engines have upon the evaluation of the piston skirts?

A. (Henriksen, Sarsten) Very little. All other conditions being equal, there will be an increase in inertia due to the increase in speed from 450 rpm to 514 rpm. This will decrease the effective load on the pistons accordingly. However, this slight decrease in effective load does not alter the fact that, for an extended period, the two AE piston skirts from the R-5 engine experienced loads in excess of any which the Shoreham piston skirts will ever experience.

Q. Are the AE piston skirts as installed in the R-5 engine dimensionally the same as the AE piston skirts installed in the Shoreham engines?

A. (Henriksen, Sarsten) No. A study of TDI drawings #03-341-04-AE (R-5 piston skirt) and #03-341-04-AE (Shoreham piston skirt) reveals no differences externally or in the critical areas around the stud holes. However, inside the piston skirt in the area of the wrist pin boss, piston skirt #03-341-04-AE (Shoreham piston skirt) appears strengthened as compared to piston skirt #03-341-04-AE (R-5 piston skirt).

Q. Do you consider the differences between the piston skirts installed in the R-5 engine and those installed in the Shoreham engines to be significant in evaluating the applicability of the operating experience in the R-5 engine?

A. (Henriksen, Sarsten) No. If anything, the piston skirts installed in the Shoreham engines appear to be superior to those installed in the R-5 engine.

Q. Are you familiar with the term "piston skirt side thrust load"?

A. (Henriksen, Sarsten) Yes.

Q. Do you consider the County's contention regarding excessive side thrust to be a matter of concern for pistons installed in the engines at Shoreham?

A. (Henriksen, Sarsten) No. In our experience with medium-speed, high brake mean effective pressure, 4-cycle engines, piston skirt side thrust has never been a problem in piston skirt design.

Q. In your opinion would the AE piston skirt be considered unique in design for diesel engines of this size, speed, and load requirements that would make it vulnerable to excessive side thrust load?

A. (Henriksen, Sarsten) No. Through Ricardo Consulting Engineers, Ltd., Shoreham-by-Sea, England, consultants to PNL, we have available a tabulation (page 5 of Exhibit 7 enclosed with this testimony) accompanied by a sketch (page 7, Exhibit 7), of seven piston skirts, made by different manufacturers. The tabulation includes cylinder bore, data to accurately locate the wrist pin in the piston skirt, maximum firing pressures, and rated BHP/cylinder. The data clearly indicate that there is no drastic difference in design criteria and operating conditions between the AE piston skirts and the other six piston skirts represented in the tabulation. Furthermore, the data indicate that the side thrust load likely to be experienced by the AE piston skirt will be representative of what is demanded of piston skirts in medium-speed, high BMEP diesel engines today.

Q. Are you aware that the AE piston skirts at Shoreham are tin-plated?

A. (Henriksen, Sarsten) Yes.

Q. Do you know why tin plating is applied on piston skirts?

A. (Henriksen, Sarsten) We can see two reasons. One reason would be for conservation purposes, i.e., to prevent iron skirts from rusting during storage and transit, etc. This would likely be a minimal coating and considered sacrificial when running the engine. The second reason would be to assist in the initial break-in period.

Q. Have you any opinion as to how thick the tin plating of the piston skirts should be in order to assist in the initial break-in?

A. (Henriksen, Sarsten) This becomes a matter of judgment. The operating conditions enter into this judgment. Assuming no problems are experienced with lube oil coking on the piston crown and in ring grooves, the fuel the engine will have to burn becomes a major factor in deciding the tin plating thickness. For example, for gas-burning engines, a relatively heavy tin plate coating may be used. On the other end of the spectrum, a heavy fuel-burning engine which will have a fair amount of carbon particles passing by the piston rings down to the crankcase can tolerate only a thin coat of tin plating in order to minimize carbon embedding in the tin plating. We believe a tin plating thickness range of 0.001-inch to 0.0015-inch to be acceptable for piston skirts operating on a good grade number 2 diesel fuel.

Q. Do you know what the thickness of the tin plating is on the AE pistons?

A. (Henriksen, Sarsten) The drawing calls for plating of 0.003 inch on the diameter, which, if properly controlled during the electrolysis procedure, converts to 0.0015 inch on the radius.

Q. On your visit to Shoreham on May 23, 1984, did you have occasion to inspect the AE piston skirt exterior surfaces?

A. (Henriksen) Yes. I found signs of scuffing on most pistons. None of the scuffing, which was judged to be the result of carbon particles embedded in the tin plating, was judged to be serious. There were no signs of distress such as hot spots or discoloration indicating that the skirts had been overloaded.

Q. Did you also have occasion to inspect the cylinder liners for signs of scuffing?

A. (Henriksen) No. The cylinder liners were reportedly at TDI in Oakland, California, where they were to be installed in the new cylinder block.

Q. Have you drawn any final conclusion regarding the AE piston skirt?

A. (Henriksen, Sarsten) We have concluded that, on the basis of presently available information, the AE piston skirts will be suitable for nuclear service for one refueling cycle. This conclusion is based on the conditions that all pistons will be examined by dye penetrant in the area of the stud bosses and that the pistons are as represented by TDI drawing #03-341-04-AE.

ATTACHMENTS

WITNESSES' PROFESSIONAL QUALIFICATIONS

ATTACHMENT 1

Professional Qualifications

Carl H. Berlinger

Division of Licensing
Office of Nuclear Reactor Regulation
United States Nuclear Regulatory Commission

## Education

| | | |
|---|---|---|
| B.S. | Mechanical Engineering, Clarkson College of Technology | 1960 |
| M.S. | Mechanical Engineering, Clarkson College of Technology | 1962 |
| Ph.D | Mechanical Engineering, University of Connecticut | 1971 |

## Current Position

Since January 1984, Dr. Berlinger has been the Group Manager of the TDI Project Group. In this position, he manages the activities of the Project Group Staff and coordinates the efforts of NRR and other offices, interfaces with industry and licensees, and, as appropriate, keeps the ACRS, hearing boards, and the Commission informed regarding the status and resolution of this issue.

## Detailed Experience Record

September 1981 - January 1984 UNITED STATES NUCLEAR REGULATORY COMMISSION

Division of Systems Integration - Core Performance Branch

Branch Chief -

Duties included

1. Management of the activities of a branch engaged in the review, analysis and evaluation of calculational methods used by applicants for the licensing of nuclear power plants in the fuel and core design areas of reactor plant engineering.

2. Responsible for development and application, in conjunction with consultants, of independent calculational methods including complex computer codes for the analysis of fuel and reactor core performance during steady-state, transient, and accident conditions.

3. Participates as a technical specialist on various NRC committees, subcommittees, panels, task force assignments, and on technical, industrial and professional society committees.

4. Represents the Commission in dealings with other governmental departments and agencies, national laboratories, industry and industry organizations in discussion of complex technical matters in the areas of new or proposed reactor systems.

November 1980 - September 1981

UNITED STATES NUCLEAR REGULATORY COMMISSION

Division of Licensing - Systematic Evaluation Program Branch

Section Leader - Systems Engineering

Duties included:

1. Supervised senior technical staff in the Systems Engineering section.

2. Responsible for the analysis, evaluation and safety reviews in the areas of thermal hydraulics, physics, site hazards, and safety analyses aspects of the reactor core, primary and secondary plant systems, electrical and auxiliary systems.

January 1980 - November 1980

UNITED STATES NUCLEAR REGULATORY COMMISSION

Division of Licensing - Operating Experience Evaluation Branch

Branch Chief -

Duties included:

1. Organized newly formed branch; formulated goals and objectives.

2. Established procedures and significance criteria for systematic screening and technical review of domestic and foreign licensee event reports and operating experience reports, respectively.

3. Initiated staff reviews of significant licensee events.

4. Developed licensee event reporting requirements.

5. Managed and participated in the investigation of plant operating problems and identified generic reactor operating problems.

April 1976 - January 1980

UNITED STATES NUCLEAR REGULATORY COMMISSION

Division of Operating Reactors - Reactor Safety Branch

Section Leader -

Duties included:

1. Provided technical supervision and review of senior technical staff in the Reactor Safety Branch.
2. Planned, coordinated and reviewed safety design evaluations of reactor cores, reactor systems, and engineering safety factors, and in accident analysis evaluations.
3. Acted as contract coordinator.
4. Served on the initial on-site response team sent to TMI.
5. Served as the team leader of the on-site response team sent to Oyster Creek following the 1979 plant transient.
6. Served as a reactor systems expert detailed to the Office of the Executive Director.

September 1973 - April 1976

UNITED STATES NUCLEAR REGULATORY COMMISSION (AEC)

Division of Operating Reactors - Reactor Systems Branch

Senior Nuclear Engineer - Reactor Systems Section

Duties included:

1. Served as a senior reactor systems specialist.
2. Responsible for analyzing and evaluating proposed nuclear reactor designs in the areas of thermal hydraulics, nuclear and reactor system performance.
3. Represented the AEC before ACRS, licensee and industry meetings.

4. Responsible for making technical recommendations and formulating technical positions regarding standards, regulatory guides and codes as related to reactor safety.

August 1970 - September 1973

COMBUSTION ENGINEERING CORPORATION

Nuclear Power Division - Accident Analysis Department

Principal Safety Engineer -

Duties included:

1. Responsible for the development of analytical tools for analysis of LMFBR maximum hypothetical accidents.
2. Performed quality assurance of complex computer codes and plant safety analysis (including LOCA and plant transients).
3. Presented testimony before ACRS regarding the San Onofre Units 2 and 3 plants.
4. Developed a transient steam generator/superheater model for the once-through steam generator with integral economizer.

February 1969 - August 1970

UNIVERSITY OF CONNECTICUT

Mechanical Engineering Department

Graduate Teaching Assistant -

Duties included:

1. Taught undergraduate heat transfer course.
2. Designed, procured, constructed and operated all equipment and instrumentation required for Ph.D dissertation.
3. Administered a research budget of $20,000.

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| August 1961 - February 1969 | <u>PRATT AND WHITNEY AIRCRAFT</u> |

<u>Advanced Power Systems</u>

Senior Analytical Engineer -

Duties included:

1. Planning and coordinating research and development of advance engineering products.
2. Analyzed heat transfer, thermodynamic and aerodynamic problems.
3. Supervised the design, manufacture, testing and evaluation of new design concepts.

ATTACHMENT 2
# Professional Qualifications

Spencer H. Bush

Review and Synthesis Associates
630 Cedar
Richland, Washington 99352

## Education

| | | |
|---|---|---|
| B.S. | Metallurgical Engineering, University of Michigan | 1948 |
| B.S. | Chemical Engineering, University of Michigan | 1948 |
| M.S. | Metallurgical Engineering, University of Michigan | 1950 |
| Ph.D. | Metallurgy, University of Michigan | 1953 |

## Employment

| | |
|---|---|
| 1940-42 | Assistant Chemist, Dow Chemical Company |
| 1942-46 | U. S. Army (1944-46: Manhattan Project) |
| 1951-53 | Instructor, Dental Materials, U. of Michigan |
| 1953-54 | Senior Scientist, General Electric Company Hanford Atomic Products Operation (HAPO) |
| 1954-57 | Supervisor, Physical Metallurgy, General Electric HAPO |
| 1957-60 | Supervisor, Fuels Fabrication Development, GE/HAPO |
| 1960-63 | Metallurgical Specialist, General Electric HAPO |
| 1963-65 | Consulting Metallurgist, General Electric HAPO |
| 1965-70 | Consultant to the Director, Battelle-Pacific Northwest Laboratories |
| 1970-83 | Senior Staff Consultant, Battelle-Pacific Northwest Laboratories |
| 1983- | President, Review and Synthesis Associates, Richland, WA |
| 1968- | Affiliate-Adjunct Professor, Metallurgical Engineering--Joint Center for Graduate Study, University of Washington, Washington State University, Oregon State University |
| 1973-74 | Regents Professor, University of California, Berkeley |

## Affiliations (active only)

U.S. Nuclear Regulatory Commission Advisory Committee on Reactor Safeguards (Member 1966-1977, Consultant 1978-)
Executive Committee, Welding Research Council Pressure Vessel Research Committee
Member, ASME Section XI Subcommittee on Nuclear Inservice Inspection
Executive Board, ASME NDE Engineering Subdivision
U. S. Representative, OECD PISC-II Managing Group
Chairman, Washington State Board of Boiler Rules
Sigma Xi
Tau Beta Pi
Phi Kappa Phi

## Society Memberships

Fellow, American Nuclear Society
Fellow, American Society for Metals
Member, American Institute of Mining, Metallurgical and Petroleum Engineers
Fellow, American Society of Mechanical Engineers
Member, National Academy of Engineering

## Awards and Honors

National Academy of Engineering 1970
Regents Professor, University of California, Berkeley 1973-74
ASTM Gillett Lecturer 1975
ASNT Mehl Lecturer 1981
ASME Certificate, Boiler and Pressure Vessel Code
ASME Bernard F. Langer Award 1983

## Licenses

Registered Professional Engineer, Metallurgical Engineering-267 and Nuclear Engineering-292, State of California

Author or co-author of one book, 16 chapters in books, 30 journal articles and numerous other documents and technical papers.

## Summary of Current Areas of Expertise

Consultant on materials and safety with particular emphasis on environmental effects such as stress corrosion and radiation damage as they affect material properties and component design in nuclear reactors. Scientific contributions have been primarily in the physical and mechanical metallurgy of nuclear materials. Specific experimental work has been in temper embrittlement of steels. Work in reactor materials included kinetics studies of oxidation in zirconium alloys, effect of fabrication variables on properties of zirconium alloys, irradiation effects in uranium alloys and reactor structural materials, and stress corrosion. Substantial work has been done in reactor safety, particularly on failure mechanisms in pressurized systems.

A major role has been in the synthesis of available information to develop a coherent picture of the relative roles of materials, fabrication and nondestructive examination on the reliability of nuclear components. Based on such a synthesis of data generated throughout the world, it is possible to suggest changes leading to an improvement in reliability with a comparable improvement in system safety. Consulting on special assignments has become increasingly significant since 1978 for both government and private organizations. Typical activities have been in the areas of component reliability, seismic design of pressure boundary components, seismic fragility values, reactor system reliability under faulted conditions, turbine reliability and valve performance.

ATTACHMENT 3

## Professional Qualifications

Adam J. Henriksen

Adam J. Henriksen, Inc.
Diesel Consultants
7731 N. Fairchild Road
Fox Point, Wisconsin 53217

### Education

Horten High School, Horten, Norway
Graduated in 1934

Royal Norwegian Naval Academy, Engineering Branch
Graduated in 1940

American Management Association (four weeks)
General Management Course 1968-1969

### Service Record

Royal Norwegian Navy
Midshipman Engineer 1937-1940
Engineering Officer (Lieutenant S.G. at time of discharge) 1940-1946

### Societies and Registrations

The American Society of Mechanical Engineers, Member
Registered Professional Engineer in the State of Wisconsin

### Publications

A.S.M.E. Paper Number 60-WA-185, "Supercharging of a Large Two-Cycle, Loop-Scavenged Diesel Engine"

### Experience

| | |
|---|---|
| May 1980 to Date | Consulting Engineer, Diesel Engines |
| March 1975 - May 1980 | Rexnord Inc. Nordberg Machinery Group, Process Machinery Division<br>Milwaukee, Wisconsin |

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| March 1975 - May 1980 | Manager, Service Department<br>Responsible to Division Customer Service Manager for all phases of installing and servicing the Company's product lines of crushers, screens, mills and hoists. Further responsible for all administration of up to 24 authorized repair facilities. |
| November 1953 - March 1975 | Rexnord Inc. Nordberg Machinery Group, Power Machinery Division<br>Milwaukee, Wisconsin |
| September 1966 - March 1975 | Manager, Test and Service Department<br>Responsible to Division General Manager for all phases, inclusive financial and contracting, involved in testing, installing and servicing the company's line of diesel engines and gas turbines. The department consisted of five subsections. |
| September 1965 - September 1966 | Chief Field Engineer<br>Responsible to Manager, Test and Service Department for all field testing, including field R/D work on the company's line of diesel engines. Further responsible for solving problems arising in the field, and for reducing no-charge costs resulting from problems occurring in the field as well as in the factory. |
| February 1964 - September 1965 | Assistant Chief Engineer<br>Responsible to the Chief Engineer for Administrative and Technical leadership of the Engineering Department's R/D and Application groups. Further served as head of a group consisting of shop, service, and engineering personnel for the purpose of solving problems and reduce no-charge costs. |
| May 1963 - February 1964 | Head, Application Engineering<br>Responsible to the Chief Engineering for the Administrative and Technical leadership of the Engineering Department's Application group. This entailed stationary, marine, electrical, and automatic control application engineering. |
| 1961 - 1963 | Head, R/D Department<br>Responsible to the Chief Engineer for the Administrative and Technical leadership of the Engineering Department's R/D group. During this period the group was heavily engaged in R/D work required to upgrade the company's line of four-cycle diesel engines including conducting tests on heavy fuel on these engines. |
| 1955 - 1961 | Senior R/D Engineer<br>Project Engineer in charge of supercharging the company's line of two-cycle diesel, duafuel and spark-fired engines. The commercial rating of the entire product line increased by over thirty percent. |

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| 1953 - 1955 | Marine Project Engineer<br>Marine Project Engineer, planning and drawing in connection with marine installations. Calculating and specifying auxiliary equipment pertaining to above installations. |
| 1952 - 1953 | Yarrows, Ltd., Shipbuilders & Engineers, Victoria, B. C. Canada<br>Position and duties as for above. |
| 1950 - 1952 | Messrs. Zetlitz-Nilsson, Ziegler and Bang, Marine Consulting Engineers, Oslo, Norway<br>Marine Superintendent Engineer, planning of new vessels, examination of building specifications and drawings, charge of supervision of ships in service, examination of engineering reports, etc., prepare detailed specifications for tenders in connection with repairs and class surveys of ships. |
| 1947 - 1950 | Messrs. Harland & Wolff, Ltd., Shipbuilders and Engineers, Glasgow, Scotland<br>Test and Guarantee Engineer, testing marine propulsion and auxiliary diesel engines in the manufacturer's plant, supervising marine machinery installations and sea trials at home and abroad. Guarantee Engineer aboard three vessels for a total of twenty months. |
| 1946 - 1947 | Fred Olsen, Ship Owner, Oslo, Norway<br>First Assistant Engineer aboard S/S EK. |
| 1937 - 1946 | Please refer to service record |
| 1936 - 1937 | Wilhelm Wilhelmsen Lines, Ship Owner, Oslo, Norway<br>Apprenticeship required for entrance to the Royal Norwegian Naval Academy. Shipboard duties. |
| 1934 - 1936 | Horten Naval Yard, Horten, Norway<br>Apprenticeship required for entrance to the Royal Norwegian Naval Academy. Machine Shop practice. |

ATTACHMENT 4

## Professional Qualifications

Walter W. Laity

PNL Project Manager
Diesel Engine Operability/Reliability Project
Battelle, Pacific Northwest Laboratory

### Education

| | | |
|---|---|---|
| B. S. | Mechanical Engineering, University of Washington |
| M. S. | Mechanical Engineering, Oregon State University |
| Ph.D. | Mechanical Engineering, Oregon State University |

### Experience

Dr. Laity joined the staff of Battelle-Northwest in November 1974. His academic background and experience are primarily in the fields of the thermal sciences, transport phenomena, and advanced energy conversion systems.

Dr. Laity served a 5-year tour of duty (1962-1967) as a Naval officer in the headquarters organization of the Naval Nuclear Power Program, where he was involved in the engineering of machinery for Naval nuclear propulsion plants. Machinery for which he was responsible included propulsion and auxiliary turbines, reduction gears, condensers, heat exchangers, propeller shaft bearings, pumps, blowers, air conditioners, and distilling plants. During the last 3 years of that assignment, he was a technical leader for the design, manufacture, testing, and installation of steam plant components of a new design Naval nuclear plant.

Dr. Laity has gained significant additional experience at Battelle as a technical contributor, project manager, and manager of an R&D section of 38 people. His attention has been focused on fundamental and applications-oriented research in the fluid and thermal sciences, and the application of these disciplines to the evaluation and development of energy systems for both well-established and new technologies.

### Professional Registration

Registered Professional Engineer, Oregon, No. 7440.

### Professional Affiliations

American Society of Mechanical Engineers
Accreditation Board for Engineering and Technology (ASME Visitor)
Sigma Xi

ATTACHMENT 5

## Professional Qualifications

Arthur Sarsten

Professor of Internal Combustion Engines
The Norwegian Institute of Technology (NTH)
7034 Trondheim, Norway

at

Division of Combustion Engines and
Marine Engineering, Marine Technology Center
Department of Marine Technology
Hakon Hakonsons gt34
N-7000 Trondheim, Norway

### Practical Training

1942 - 1945 Apprentice, A/S Wichmann, Rubbestadneset, Norway. Machine shop work in engine factory in various lathes, drill presses, shaping etc. One year in diesel engine assembly work.

### Education

- 1939 N.Y. Public Schools + 1 Year High School
1940 - 1945 Voss off. Landsgymnas, Voss, Norway
1949 - 1953 The Norwegian Institute of Technology, Trondheim, Norway. B.Sc. in Mechanical Engineering, diploma thesis in I.C. Engines.

1958 - 1960 Renesselaer Polytechnic Institute, Troy, N.Y. Post graduate work evenings, later full time. M.Sc. in ME 1960.

1960 - 1963 R.P.I., Troy, N.Y. full time. Thesis in field of nonlinear vibrations D.Sc. 1963.

### Memberships

Society of Automotive Engineers
American Society of Mechanical Engineers
The Institute of Marine Engineers
The Royal Norwegian Society of Sciences and Letters
The Norwegian Academy of Technical Sciences

### Experience

1954 - 1959 Wichmann Motorfabrikk A/L, Rubbestadneset, Norway (Manufacturer of two-stroke marine diesel engines up to ca.

| | |
|---|---|
| | 2500 bhp.) Position would correspond to project engineer for AC type (280 x 420 mm). Design, calculation and follow-up to production stage of this type of loop-scavenged engine and hydraulic c.p. propeller units. Supervision of 1-2 detail draftsmen. |
| 1958 - 1960 | ALCO Products Inc., then at Schenectady, N.Y. Calculation of stress and vibrations in engine components. Cam design and dynamics. R&D work accumulator fuel injection. |
| 1963 - 1964 | Gebr. SULZER, Winterthur, Switzerland. Mainly 2-stroke diesel engines. Design calculator rotating through various departments. Design of cams and related computer programming, FORTRAN II for IBM 1620. Balancing and torsional vibration calculation, some test bed work. |
| 1964 - 1978 | Professor of Internal Combustion Engines, The Norwegian Institute of Technology, Trondheim, Norway, and head, Division of I.C. Engines (Institutt for forbrenningsmotorer) staff ca. 20. Also research and consultant work, mainly for foreign engine firms. Engaged in computer work FORTRAN IV, UNIVAC 1107-1108. We have been active in engine dynamics, valve dynamics, torsional vibrations, thermal loading problems, use of finite element technique for temperature and stress field calculations, sale of TESTRAN FEM-package to various engine and component firms. Lab does radioactive wear tests, bearing work, consumer tests and research on outboard engines. Headed Norwegian Large Bore Research Project 1965 - 1968 ($200 000,-) for research on thermal damage on certain crosshead engines. Awarded (with 3 co-authors) The Herbert Ackroyd Stuart Award 1968'9 from The Institute of Marine Engineers for paper reporting results of this research. |
| 1971 - 1973 | Dean, Department of Mechanical Engineering, Norwegian Institute of Technology. 14 Divisions, ca. 600-700 students. |
| 1974 | Prof. invité, Département de génie mécanique, Université de Sherbrooke, Canada. |
| 1978 - present | Professor of Internal Combustion Engines, Division of Combustion Engines and Marine Engineering, at the new Marine Technology Center. Staff approx. 40. Head of Division 1978 - 1980, (rotates). |
| 1983 - 1984 | Visiting professor at Lawrence Berkeley Laboratory, One Cyclotron Road, Berkeley, CA 94720. |

## Partial List of Relevant Publications


Sarsten, A. "A Computer Programme for Damped Torsional Vibrations Using a Complex Holzer Tabulation", European Shipbuilding No. 6. 1962. Vol. XI, p. 138-146.

Sarsten, A., Valland, H. "Computer-aided Design of Valve Cams." Int. Comb. Engines Conf., Bucharest 1967, Paper 11-19, p. 761-786.

Fiskaa, G., Iversen P., Sarsten, A. "Computer calculation of stresses in axi-symmetric thermally loaded components." Inst. of Mech. Engineers Symposium Computers in I.C. Engine Design, Manchester. April 1968, Proc. 1967-68, Vol. 182, Part 3L, p. 152-168.

Sarsten, A., Hansen, A, Langballe, M., Martens, O. "Thermal Loading and Operating Conditions for Large Marine Diesel Engines." IMAS69 Conference, London, Sect. 4, p. 38-49. Given Herbert Ackroyd Stuart Award 1968'9 by The Institute of Marine Engineers.

Hansen, A., Rasmussen, M., Sarsten, A. "Thermal Loading of Diesel Engine Components and Its Prediction." Paper A30, 9th Intern'l Congress on Combustion Engines (CIMAC) Stockholm, Sweden 1971, 25 pp.

Wacker, E., Strecker, E., Sarsten, A., Haaland, E. "Finite Element-Programme zur Berechnung von Brennraum-Bauteilen", Motortechnische Zeitschrift (MTZ) 32. Nr. 8, Aug. 1971, p. 267-279.

Sarsten, A., Holth, T., Øvbrebo, A. "A Method for Direct Solution of Steady-State Forced Vibration of Linear Systems." ASME paper 13-DGP-12, presented at Diesel and Gas Engine Power Conf. Washington, D.C. April 1973.

Sarsten, A. "Massekrefter og massemomenter ved stempelmaskiner. (Inertia forces and moments in piston engines). 176 pp., Tapir Forlag, Trondheim, 1968.

Sarsten, A. "A Direct Method for Calculating the Steady-State Vibration of Marine Shafting Systems." Report IF/R15, Div. of I.C. Engines, NTH, Trondheim, 1974.

Valland, H., Sarsten, A. "Application of the direct solution method to engine vibration problems," Norwegian MARITIME RESEARCH No. 1, 1980, Vol. 8, pp. 39-50.

Sarsten, A. "A reduction method for calculation of the forced vibration of large, free systems with multiple branch points." Report IFMM 81, Div. of Comb. Eng. and Mar. Engrg., 35 pp.

Chen, T., Sarsten, A. "Combustion simulation of medium speed diesel engines and result analysis." 2nd Congress of IMAEM, Trieste, Sept. 21-26, 1981.

Einang, P. M., Koren, S., Kvamsdal, R., Hansen, T. and Sarsten, A. "High-Pressure, Digitally-Controlled Injection of Gaseous Fuel in a Diesel Engine, with Special Reference to Boil-Off from LNG Tankers," Paper, CIMAC '83 Conference, Paris, June 1983.

RELATED CORRESPONDENCE

DOCKETED
USNRC

'84 SEP -5 P2:34

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) | ) | |

EXHIBITS
for

JOINT TESTIMONY
of
CARL H. BERLINGER, SPENCER H. BUSH,
ADAM J. HENRIKSEN, WALTER W. LAITY, AND PROFESSOR ARTHUR SARSTEN
on
CONTENTIONS CONCERNING TDI EMERGENCY DIESEL GENERATORS
at the
SHOREHAM NUCLEAR POWER STATION

VOLUME 2

EXHIBIT 1

# LILCO CRANKSHAFTS

A ABS Requirements: (Par 34.17.1 dia. of pins & journals)

$$d = c\sqrt[3]{\frac{M + (M^2 + 4T^2)^{1/2}}{f}}$$

where:

$d$ = crankpin diameter, in

$c$ = 1.0 (for more than 6-cyl. engines)

$D$ = cyl. bore, 17 in

$P$ = max. firing pressure

a) 1700 psi @ 4890 bhp

b) 1800 psi @ 5380 bhp

$L$ = span between bearings (inner edge to inner edge of main bearings), 17.93 in

$H$ = horsepower at rated speed

a) 4890 bhp (100%)

b) 5380 bhp (110%)

$R$ = rated speed, 450 rpm

$f$ = grade 4 forging, 2,310

and: $M = 0.131\, PD^2L$

$T = 63{,}000\, H/R$

Required crankpin diameter :-

a) 100% load or 4890 bhp : d= 10.84"

b) 110% load or 5380 bhp : d= 11.103"

): The 12" crankshaft is acceptable both at 100% & 110% load, as far as crankpin diameter is concerned.

B Paragraph 34.17.4 Solid Crankshaft Webs

$$wt^2 \geq 0.35\, d^3$$

where $w$ = effective width of web, 21"
* $t$ = thickness of web 4.965"

* Note: Proportions are such that pins & journals overlap. Furthermore, the pin fillet is undercut with a re-entry into the web. Interpreting the correct method to determine t for such a case, Waytowich of ABS replies to a question concerning this on p 129, line 21 to p 130, line 6 of their testimony:

" I believe that our normal practice would be to measure that dimension from the boundary of the actual crankshaft material, at one fillet to that at its opposite fillet, rather than constructing the arbitrary lines of a face of the web and going between them.

Essentially it makes sense to count only the metal that is actually there."

Employing this interpretation, a figure of 4.965" has been determined (See full scale drawing)

): $wt^2 \geq .35\, d^3$

a) 100% load; $d \geq 10.84$ in

$$wt^2 \geq .35 \cdot 10.84^3$$

$$21 \times 4.965^2 \geq .35 \times 10.84^3$$

$$518 \geq 445.81$$ Satisfied at diameter required for 100% load

b) 110% load; $d \geq 11.03$ in

$$wt^2 \geq .35 \cdot 11.03^3$$

$$21 \times 4.965^2 \geq .35 \cdot 11.03^3$$

$$518 \geq 479.02$$ Satisfied at diameter required for 110% load

At the limit $wt^2 = .35 d^3 \Rightarrow d = 11.39''$

The crankshaft will meet ABS requirements up to a $d = 11.39''$, given by formula in section 34.17.1.

i.e. for power and firing pressures in excess of 110% load.

Arthur Sauston

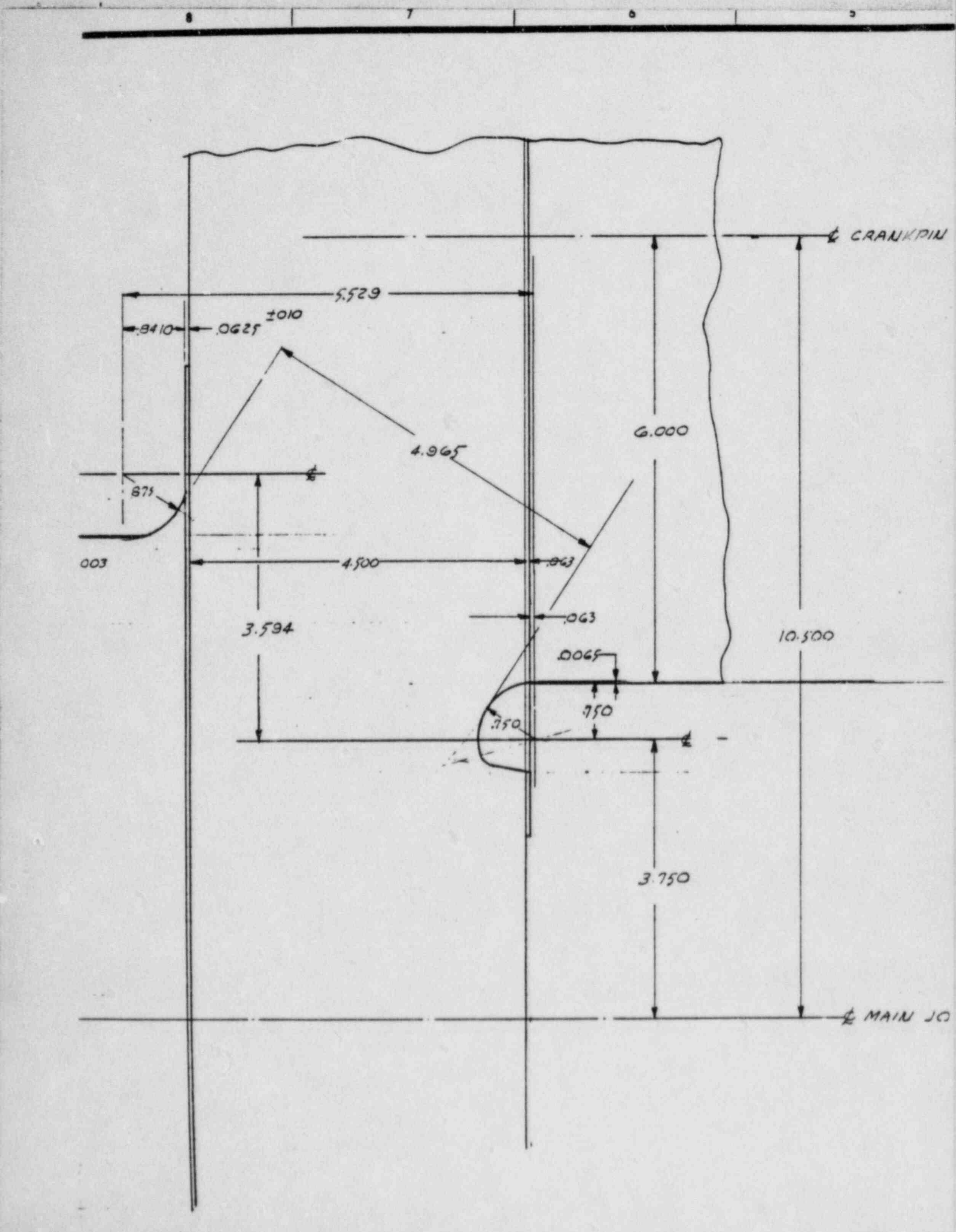
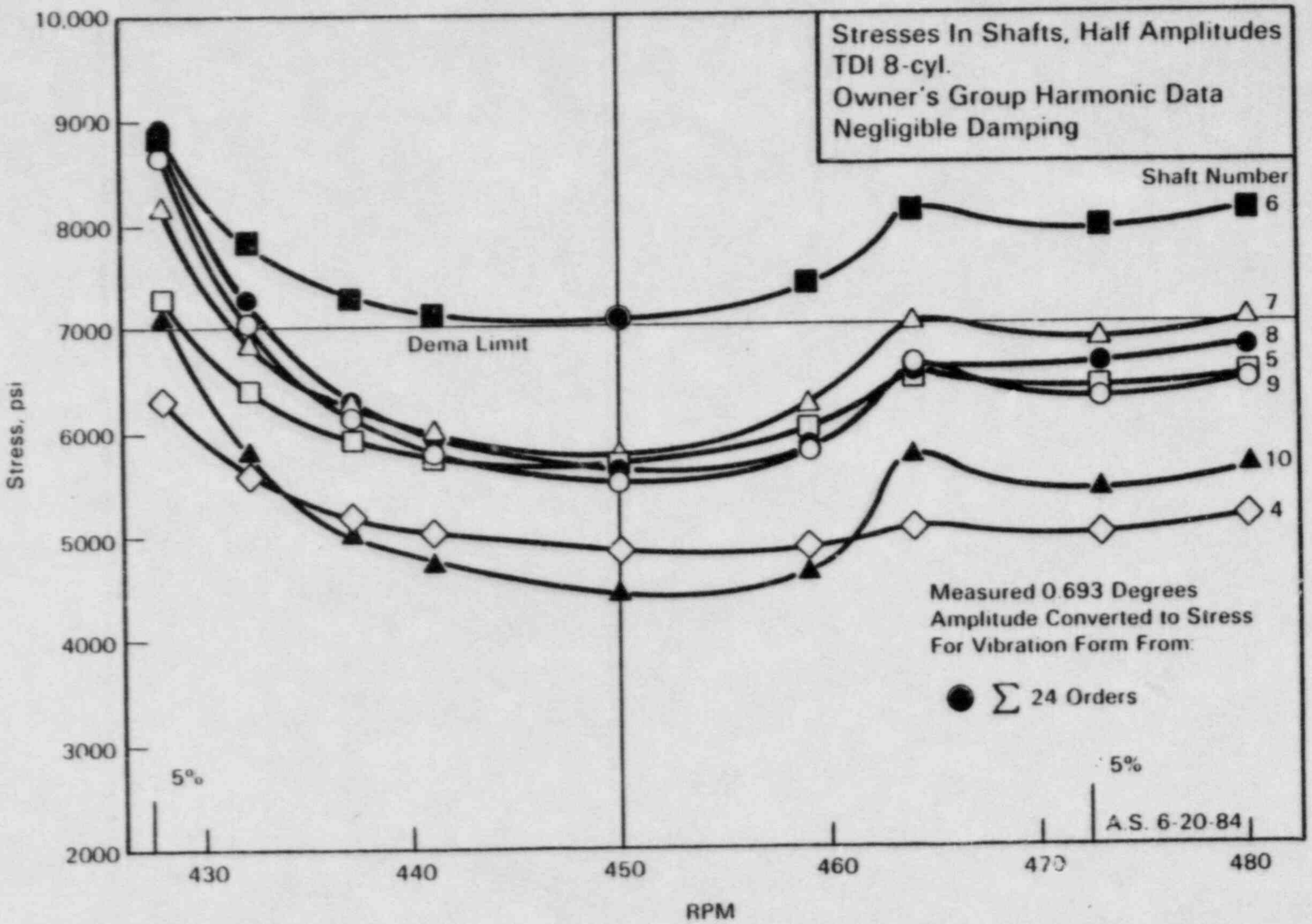

8
7
6
5
℄ CRANKPIN
5.529
.8410
.0625 ±.010
4.965
6.000
.875
℄
.003
4.500
.063
.063
3.594
10.500
.0065
.750
.750
℄
3.750
℄ MAIN JO

EXHIBIT 2


Stresses In Shafts, Half Amplitudes
TDI 8-cyl.
Owner's Group Harmonic Data
Negligible Damping
Shaft Number
6
7
8
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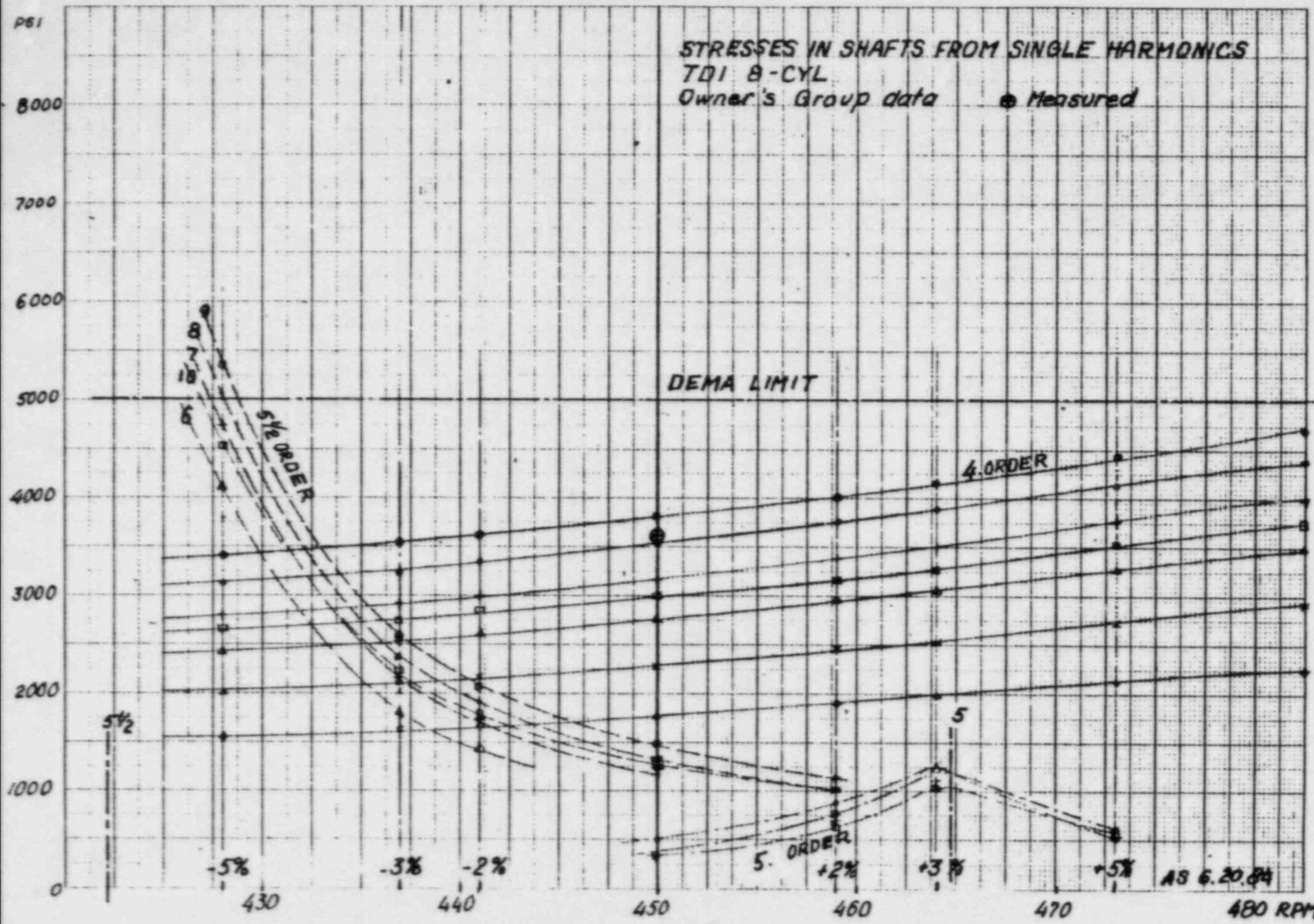
9
10
4
Dema Limit
Measured 0.693 Degrees
Amplitude Converted to Stress
For Vibration Form From:
∑ 24 Orders
5%
5%
A.S. 6-20-84
Stress, psi
10,000
9000
8000
7000
6000
5000
4000
3000
2000
430
440
450
460
470
480
RPM

EXHIBIT 3


STRESSES IN SHAFTS FROM SINGLE HARMONICS
TDI 8-CYL
Owner's Group data
Measured
DEMA LIMIT
PSI
8000
7000
6000
5000
4000
3000
2000
1000
0
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440
450
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5½
5
5½ ORDER
4. ORDER
5. ORDER
9
8
7
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6
5
4
AS 6.20.84

EXHIBIT 4

*American Bureau of Shipping*

*Sixty-five Broadway*

*New York, N.Y. 10006*

Refer to RTW:ml

3 May 1984

File No. T8-3

Transamerica Delaval DSR-48 Diesel Engine/Generator for Long Island Lighting Company Shoreham Plant Report on Crankshaft Torsional Stresses.

Transamerica Delaval Inc.
Engine & Compressor Division
550 85th Avenue
P. O. Box 2161
Oakland, CA 94621

Attention: Mr. Roland T. M. Yang
Manager Applied Mechanics.

Gentlemen:

We have your letter of 3 April 1984 submitting copies of the above subject report for our review, and with regard thereto have to advise as follows:

We note from the submitted report that the torsional vibration stress in the crankshaft for the first mode 5½ order critical speed (422 RPM) was expected to approach or exceed that permitted by the Rules for the submitted crankshaft material.

We further note from the submitted report that tests were conducted to determine the actual stresses in the crankshaft, and that these tests indicated a substantial margin of safety against fatigue failure due to torsional vibration.

Based on the submitted test data, and on submitted service experience with similar engines having similar torsional critical speed arrangements, we advise that we would have no objection to the submitted torsional critical speed arrangement for use on diesel generator sets on an ocean going vessel, insofar as our classification requirements for marine service are concerned.

Three (3) copyies of the subject report, stamped to indicate our review, are being returned.

Very truly yours,

AMERICAN BUREAU OF SHIPPING

W. M. HANNAN
Vice President

by: *Robert A. Giuffra*
Robert A. Giuffra
Principal Surveyor - Machinery

G. E. T. S. O. A. R. F. R. T. Y. M. H. L. C. R. C.
RECEIVED
TICKLER MAY 07 1984 UPDATE
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TO FILE: ; SEE ME

cc: LILCO. (E. Montgomery)
Accounting Dept. w/enclosure
Legal Dept. (M. Adams)
Subject File 460

TELEPHONE 212-440-0300 CABLE ADDRESS RECORD TWX 710-581-3089 TELEX ITT 421966 RCA 232099 WUI 620353

REPORT

ON

CRANKSHAFT TORSIONAL STRESSES

TRANSAMERICA DELAVAL MODEL DSR-48

Serial No. 74010/12

for

LONG ISLAND LIGHTING COMPANY

REVIEWED IN
ACCORDANCE WITH
ABS LETTER DATED
3 MAY 1984
AMERICAN BUREAU
OF SHIPPING
NEW YORK

Roland Yang.
April 4. 1984
Transamerica Delaval I
Oakland, CA.

TABLE OF CONTENTS

## ALLOWABLE TORSIONAL STRESS CALCULATION.

**Based on Para. 34.47 of 1984 ABS Rules.**

$$S = \left(\frac{U + 23180}{18}\right) C_k \; C_d \; C_r$$

where U = Minimum Tensile Strength of Shaft Material 100000 PSI

$C_k$ is .55 for propeller shafts and crankshafts

$C_d$ is size factor, $.35 + 0.487 / \sqrt[5]{12} = .6463$

$C_r$ is speed ratio factor, 1.38 for 90% to 105% rated RPM.

$$S = \left(\frac{100000 + 23180}{18}\right) ( .55 )( .6463 )( 1.38 )$$

=3357 PSI due to single order

Total Allowable Stress = 150% of 3357 = 5035 PSI

## ALLOWABLE TORSIONAL STRESS CALCULATION.

**Based on Table 34.3 of 1982 ABS Rules.**

| Engine Speed | .3 x 450 RPM = 135 RPM | .8 x 450 RPM = 360 RPM | .95 to 1.0 x 450RPM 427.5 to 450 | 1.05 x 450RPM 472.5 RPM |
|---|---|---|---|---|
| Grade 2, 60000 psi | 5689 psi | 3556 psi | 2134 psi | 3556 psi |
| Grade 4, 100000 psi | 8217 psi | 5136 psi | 3082 psi | 5136 psi |

Stress limit multiplier $= \frac{2}{3}\left(\frac{100000 - 60000}{60000}\right) + 1 = 1.4444$ for adjustment from 60000 psi to 100000 psi material.

EXHIBIT 5

*J. B. Kohls,*[1] *J. T. Cammett,*[1] *and A. W. Gunderson*[2]

# Effects of Multiple Shot-Peening/Cadmium-Plating Cycles on High-Strength Steel

**REFERENCE:** Kohls, J. B., Cammett J. T., and Gunderson, A. W., **"Effects of Multiple Shot-Peening/Cadmium-Plating Cycles on High-Strength Steel,"** *Residual Stress Effects in Fatigue, ASTM STP 776,* American Society for Testing and Materials, 1982, pp. 158-171.

**ABSTRACT:** A study was made of the effects of multiple shot-peening and cadmium plating operations on high-strength AISI 4340 steel used in aircraft landing-gear applications. No detrimental effects were observed on surface microstructure and tensile properties or on fatigue and unnotched stress corrosion resistance in high-humidity air. An apparent degradation in stress corrosion life of fatigue precracked specimens was observed after four and five peening and plating operations.

**KEY WORDS:** shot-peening, cadmium plating, fatigue, stress corrosion, tensile, high-strength steel

High-strength steels are used widely for load-bearing components in aircraft landing gear. Typically, such components are shot-peened after machining, then are plated with cadmium and chromium followed by painting, all to enhance resistance to fatigue and corrosion. Overhaul rework procedures for such components include stripping platings, inspecting for cracks, build-up and remachining of worn areas, followed by shot peening and plating as for the original finishing sequence. Landing-gear components typically are subjected to several such overhaul procedures during their service life.

The objective of this program was to establish the effects of the original and overhaul rework peening and plating cycles on fatigue and stress corrosion resistance of high-strength AISI 4340 steel which is commonly employed in aircraft landing-gear components. Experimental evaluations involved metallography and tension testing in addition to fatigue and stress corrosion testing in high-humidity environments. The remaining sections of this paper are devoted to descriptions of material and specimen preparation, test procedures, results obtained, and interpretation thereof.

[1] Metcut Research Associates Inc., Cincinnati, Ohio 45209.
[2] U. S. Air Force, AFWAL/MLLX, Wright-Patterson Air Force Base, Ohio 45433.

## Procedure

### *Material and Specimen Preparation*

The material employed in this work was vacuum-melted AISI 4340 steel per requirements of MIL-S-8844. This material, heat-treated nominally to a 1790 to 1830 MPa ultimate strength level, was used in landing gear of many earlier aircraft. The material was procured in the form of forgings 25 by 108 by 1829 mm. Each forging was cut into eight specimen blanks approximately 12 by 102 by 460 mm. Specimens were rough machined about 4 mm oversize prior to heat treatment. The geometries of tension, fatigue, and stress corrosion specimens are shown in Fig. 1. Following rough machining, all specimens were heat-treated.

The heat treatment consisted of oil quenching from 1085 K and tempering at 480 K. The resulting hardness was 52 to 54 Rc. The average results from tension tests were 2070 MPa ultimate tensile strength, 1397 MPa 0.2 percent yield strength, 51 percent reduction of area, and 12.4 percent elongation (25 mm gage length). After heat treatment, the specimens were finish machined. The final 0.5 mm of material was removed from all surfaces by a controlled low-stress grinding procedure [*1*].[3] This introduces low-level compressive stresses at the surface and within about 0.1 mm beneath the surface. Further, this grinding procedure does not produce any overtempering or re-transformation of the martensitic surface microstructure. After finish grinding, the edges of the specimen gage sections were radiused to about 1 mm and hand polished through 600 grit SiC paper to a surface roughness of about 0.2 $\mu$m AA.

### *Shot-Peening*

Following heat treatment and machining, specimens other than those tested in the baseline condition (no shot-peening or cadmium plating) were shot-peened per MIL-S-13165B. Specimens were clamped in a vertical position and rotated at 10 to 15 rpm. Six nozzles were used to propel the shot simultaneously at the specimen. These nozzles oscillated during peening to ensure consistent overall coverage of the surface. After peening for 3 min, each specimen was flipped end for end and then peened for an additional 3 min. Peening was performed with hardened size 230 steel shot. Coverage was 200 percent. The resulting Almen strip intensity was 6A to 8A.

### *Cadmium Plating*

Cadmium plating was performed per MIL-C-8837, Type II. The procedure involves vacuum deposition of cadmium followed by a supplementary chromate treatment to form a protective oxide film. Specimens were cleaned in a solvent and were lightly dry-blasted prior to insertion in the vacuum chamber to ensure

[3] The italic numbers in brackets refer to the list of references appended to this paper.

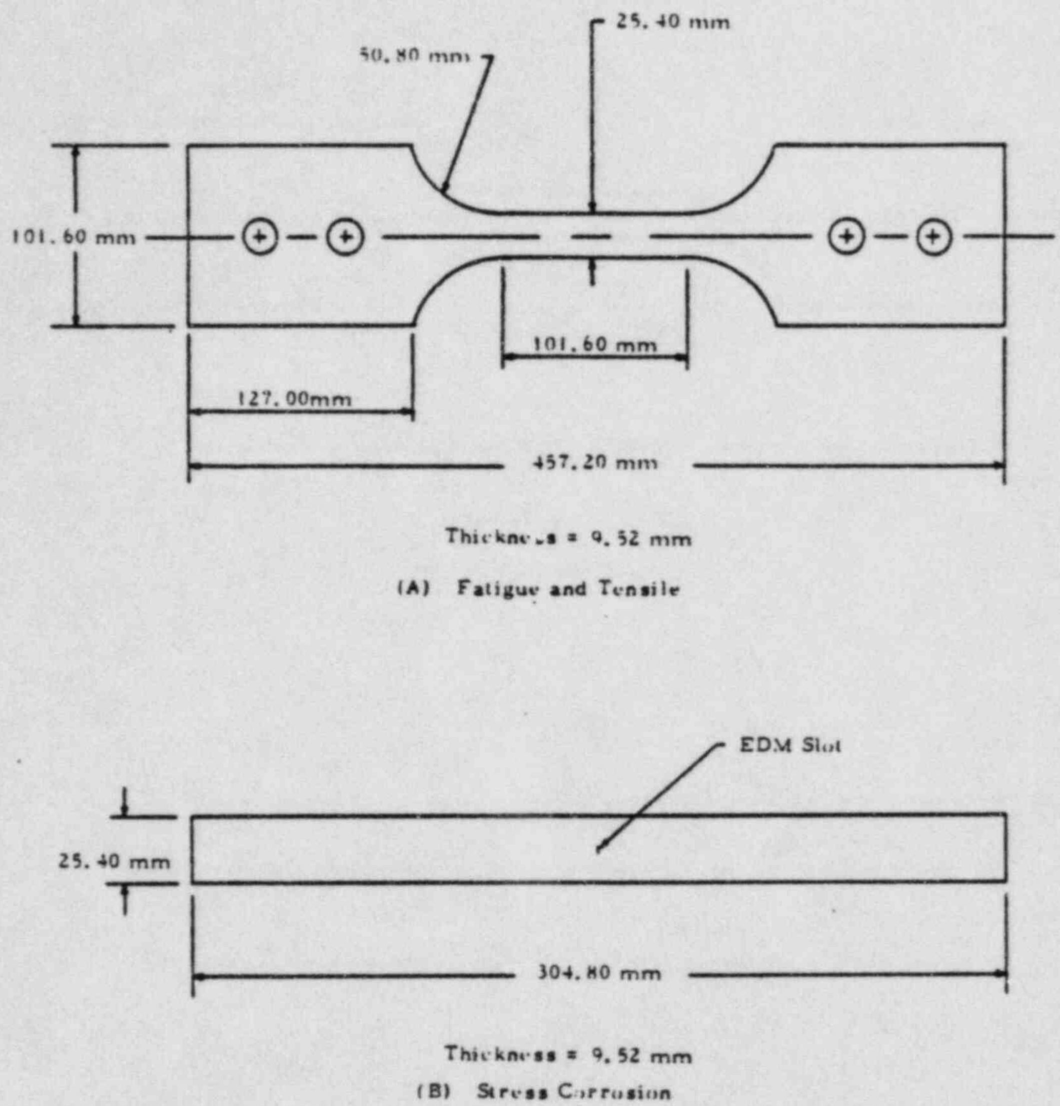



FIG. 1—*Specimen geometries.*

cleanliness of surfaces. The blasting did not roughen the surface beyond the finishes specified in Fig. 1. The plating on specimens selected for multiple shot-peening and plating cycles was stripped between each cycle.

### *Tension and Fatigue Testing*

Tension and fatigue tests were performed on a servocontrolled closed-loop hydraulic universal test machine. The load cell and all support equipment were calibrated immediately before and after this program using secondary standards whose calibrations were traceable to the National Bureau of Standards. The loading grips and associated fixtures were aligned using a strain gaged specimen of the same geometry as the test specimen.

Tension tests were performed per ASTM Methods of Tension Testing of Metallic Materials (E 8) in ambient air at about 293 K and 50 percent relative humidity.

The strain rate for all tests was 0.005 $min^{-1}$ to failure. Strain measurement was performed via an LVDT extensometer attached to the specimen gage section over a 25 mm gage length.

Fatigue tests were conducted under constant load amplitude conditions at stress ratio $R = 0.1$ and $-0.3$ in a high-humidity air environment. The environment was maintained by bubbling compressed air slowly through a column of water and then passing the air into a plastic jacket surrounding the specimen gage section. All testing was performed at a frequency of 2 to 4 Hz using a sinusoidal load-time waveform. Tests were terminated after $10^6$ cycles if fracture had not occurred beforehand.

### *Stress Corrosion Testing*

Stress corrosion testing was performed per ASTM Practice for Preparation and Use of Bent-Beam Stress-Corrosion Test Specimens (G 39) with the exception that tests were conducted under constant load rather than constant displacement in four-point bending. Testing was conducted in deadweight-loaded test frames, commonly used for creep and stress rupture testing. The frames were outfitted with four-point bend fixturing specially designed for this program. The constant bending moment test section of each specimen was the central 75 mm of its 300 mm length.

The test environment was 293 K air at 80 to 100 percent relative humidity produced by slowly bubbling compressed air through a water reservoir and then passing it into a plastic bag surrounding the specimen test section. Both unnotched and fatigue precracked specimens were tested. The fatigue precracked specimen had been manufactured with 1.2 mm wide by 0.6 mm deep electrically discharge machined (EDM) notch in the geometric center of one surface. These specimens were fatigue precracked before any shot-peening or plating cycles. Fatigue precracking was performed in ambient air under three-point bend loading at a frequency of 30 Hz and a stress ratio $R$ of about 0.1. Fatigue cracks were initiated at a calculated maximum surface stress of 100 ksi and were permitted to grow until the total surface notch plus crack length reached 2.5 mm.

## Results and Discussion

### *Residual Stresses*

No residual stress measurements were included in the scope of this work. In previous work, however, Metcut Research Associates performed residual stress measurements on quenched and tempered AISI 4340 (50 Rc) [*1*]. Residual stress results from that work, characterizing surface and subsurface residual stresses parallel to the grinding direction, are shown in Fig. 2. Please note that this figure, reproduced from Ref *1*, is in customary English units rather than the SI units used otherwise throughout this paper. As can be seen, the gentle grinding produced relatively low compressive stresses to a depth of less than 0.05 mm (0.002 in.), while the shot-peening produced relatively large compressive stresses to a depth

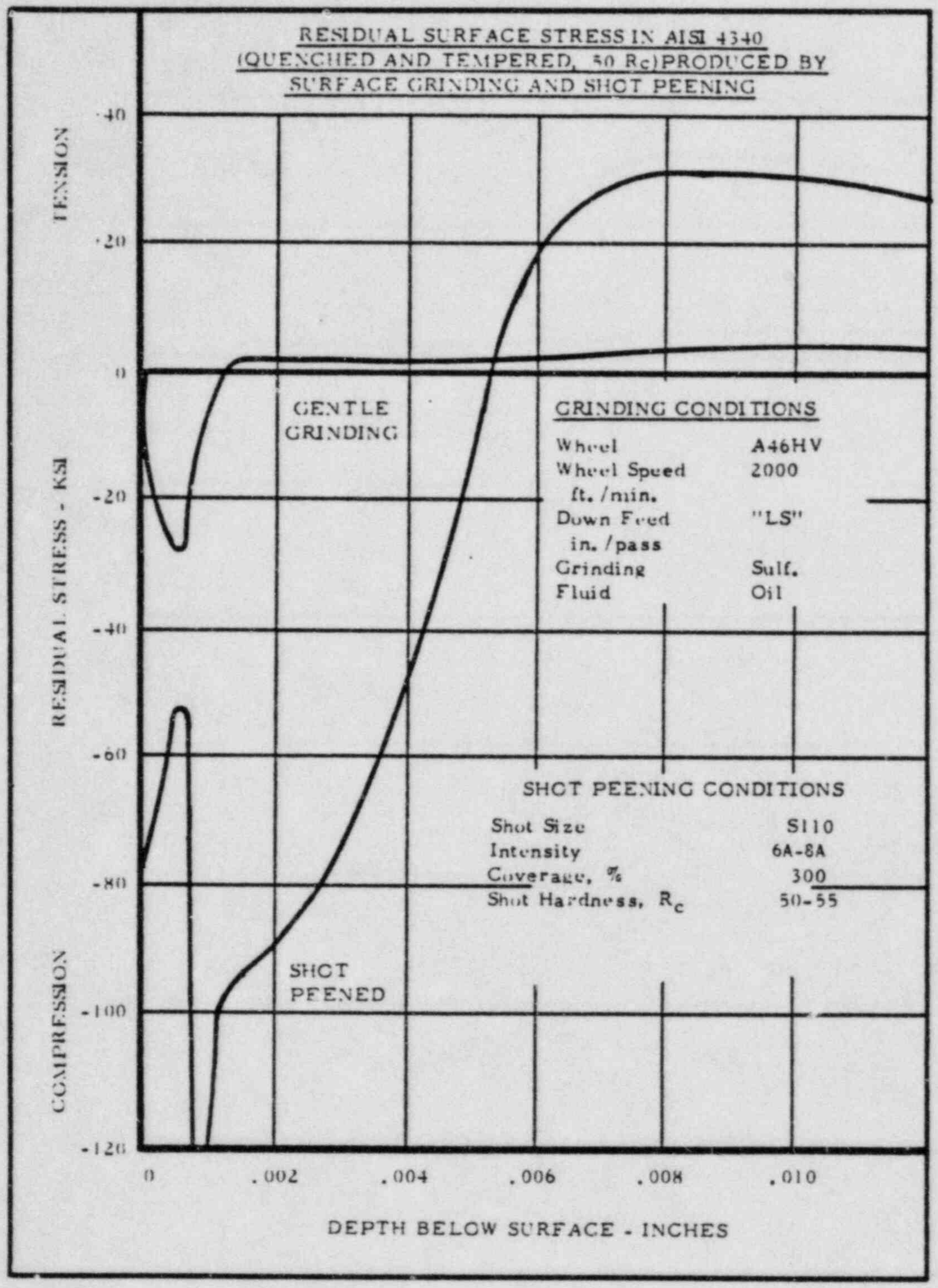



FIG. 2—*Residual stress data for AISI 4340 steel, 50 Rc (1 ksi = 6.9 MPa; 1 in. = 25.4 mm)* [1].

in excess of 0.1 mm. It is believed that the residual stress data shown in Fig. 2 are representative of residual stresses created in the AISI 4340 steel employed in the current study, since the same grinding and shot-peening parameters were used.

*Tension Test Results*

Tension test results from baseline specimens (as-heat-treated and gently ground) and from specimens subjected to from one to five shot-peening and plating cycles are summarized in Fig. 3. As can be seen, no degradation of tensile strength, yield strength, or elongation occurred as a result of shot-peening and plating cycles.

*Fatigue Test Results*

Fatig·e testing was performed axially at maximum stress levels of 1170 and 1380 M. a at stress ratios $R$ of 0.1 and −0.3. Results representing each combination of stress level and stress ratio are presented in Fig. 4. It is evident that the

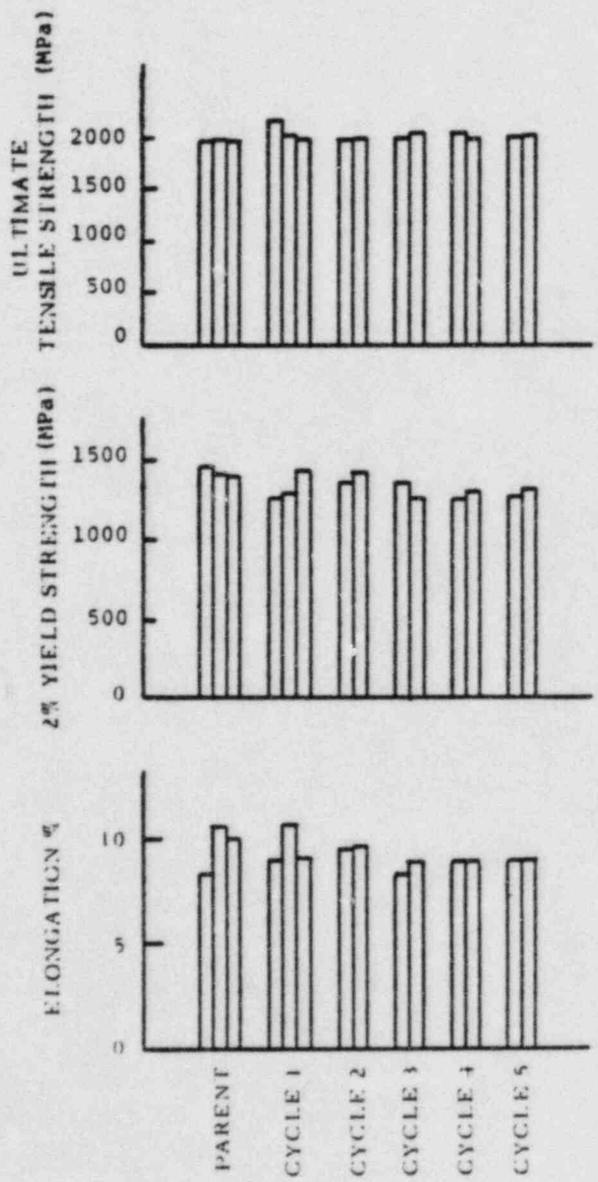



FIG. 3—*Tension test results.*

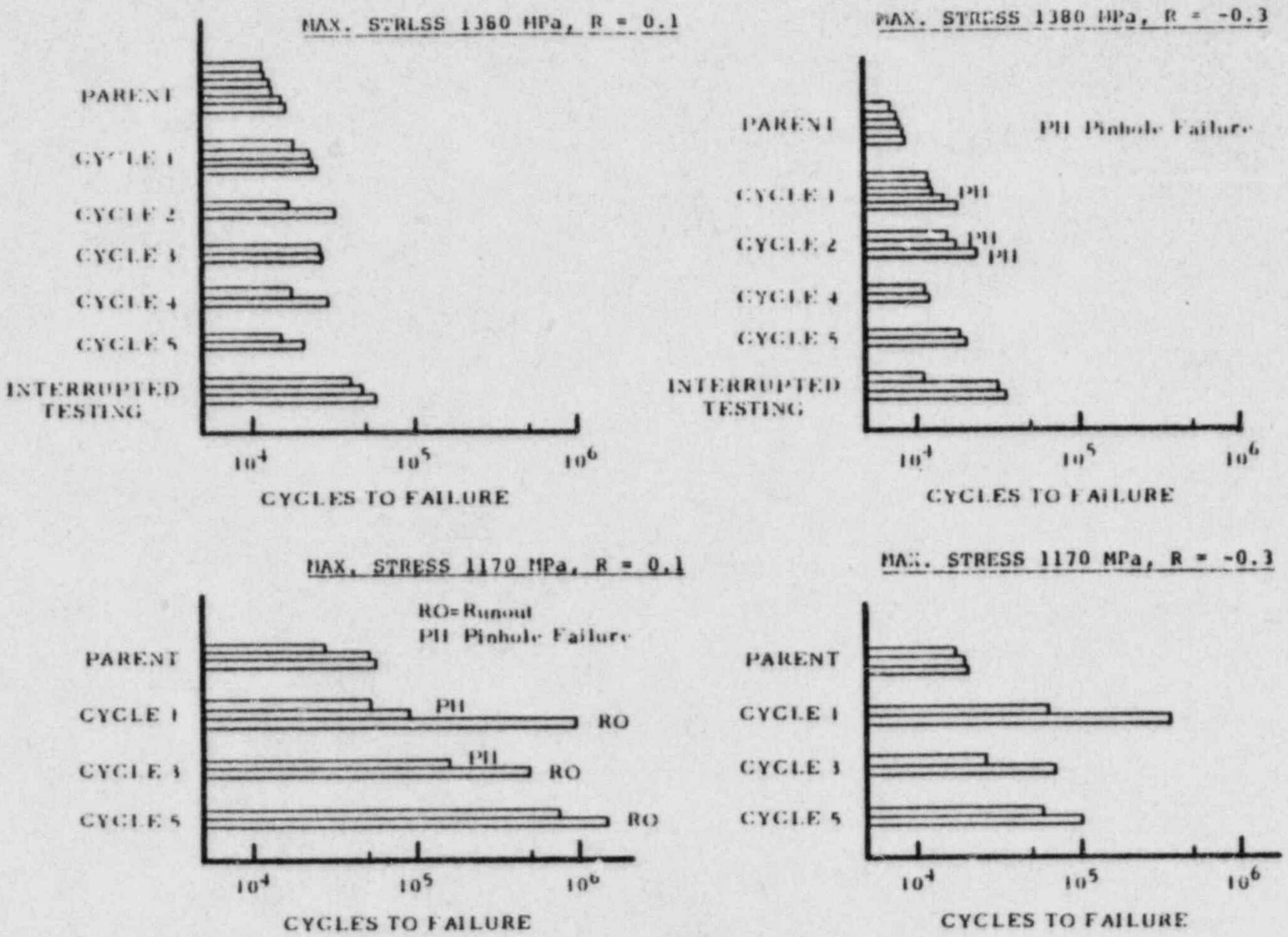



FIG. 4 — *Fatigue test results.*

average fatigue lives of specimens subjected to one to five shot-peening plus plating cycles exceeded the average lives of all baseline specimens tested at the same stress level and stress ratio. This effect, however, was greater for specimens tested at the lower stress level (1170 MPa) than for specimens tested at the higher stress level (1380 MPa).

The greater fatigue life after shot-peening is consistent with the residual stress patterns presumed to be in the specimens, since previous work by Metcut has shown a strong correlation between peak residual stress and fatigue strength in AISI 4340 steel [2]. It is believed further that the effect of shot-peening is less pronounced for the higher testing stress level (1380 MPa) because this is close to the magnitude, though opposite in sense, of shot-peening residual stresses presumed to be in the surface and subsurface layers.

It is also evident from the results in Fig. 4 that fatigue lives of specimens subjected to from three to five shot-peening and plating cycles were generally lower than lives of specimens subjected to one or two such cycles. Determination of the reason for this was beyond the scope of this investigation. It is believed, however, that the observed behavior resulted either from an over-peening effect or from hydrogen accumulation with repeated stripping, peening, and plating operations. It is re-emphasized, however, that fatigue lives of shot-peened and plated specimens generally exceeded those of baseline specimens regardless of the number of shot-peening and plating cycles.

Also shown in Fig. 4 are fatigue results from "interrupted testing" wherein specimens were cycled in fatigue between successive shot-peening and plating cycles. The number of fatigue cycles applied after each shot-peening and plating cycle was one fourth the average fatigue life of specimens tested at the same stress level and stress ratio to failure after just one shot-peening and plating treatment. After three such increments of fatigue cycling and four cycles of shot-peening and plating, the specimens were tested to failure. It is evident that the lives of specimens thus treated exceeded those of all baseline specimens and generally exceeded those of specimens subjected to from one to five shot-peening and plating cycles without intermittent fatigue cycling.

### *Stress Corrosion*

A total of 24 stress corrosion tests were performed, 14 on smooth specimens and 10 on fatigue precracked specimens. All multiple shot-peening and plating cycles were performed on individual specimens prior to stress corrosion testing. All precracking of notched specimens was performed prior to shot-peening and plating cycles.

Initially, the maximum bending stress level for testing was chosen to be equal to the 0.2 percent offset yield stress (1415 MPa) for the material. This level subsequently was increased to 1655 MPa when no specimen failures were observed at the lower stress level. Therefore the surface stress level as reported here is a pseudo-elastic stress level calculated per simple beam theory rather than an actual stress level. Specimens were held at load in the moist air environment for at least 200 h or until fracture, whichever occurred first.

Stress corrosion results for smooth specimens are presented in Table 1. These results are inconclusive with respect to the influence of shot-peening and plating on stress corrosion resistance, since no stress corrosion failures occurred. Visual examination of specimens after testing revealed neither any cracking nor any general corrosion on the specimens.

Stress corrosion results from notched and fatigue precracked specimens are presented in Table 2. It is evident that lives of specimens subjected to four or five shot-peening and plating cycles were lower than for baseline specimens or those subjected to a lesser number of such cycles. As was mentioned previously in discussion of fatigue results, it is believed that this behavior resulted either from an over-peening effect or from hydrogen accumulation during successive stripping, peening, and plating operations. The extent of fatigue precracking in specimens so prepared greatly exceeded the depth to which any shot-peening would have influence. Therefore the belief is favored that hydrogen accumulation was responsible for the observed behavior.

### *Metallography*

The metallographic specimens prepared for this program were oriented parallel and perpendicular to the machining lay. The specimens were mounted in epoxy material embedded with aluminum oxide pellets for optimum edge retention. They were polished by conventional means and examined in the unetched and etched conditions at magnifications of up to approximately ×1000. The etchant used was a 2 percent Nital solution.

Baseline 4340 samples and five groups of samples with varying number of shot-peening and plating cycles were examined. Surface structural features are briefly described and characterized by photomicrographs shown in Fig. 5. Traces of a thin white layer were observed on the surfaces of the peened samples. These

TABLE 1—*Stress corrosion results—smooth specimens.*

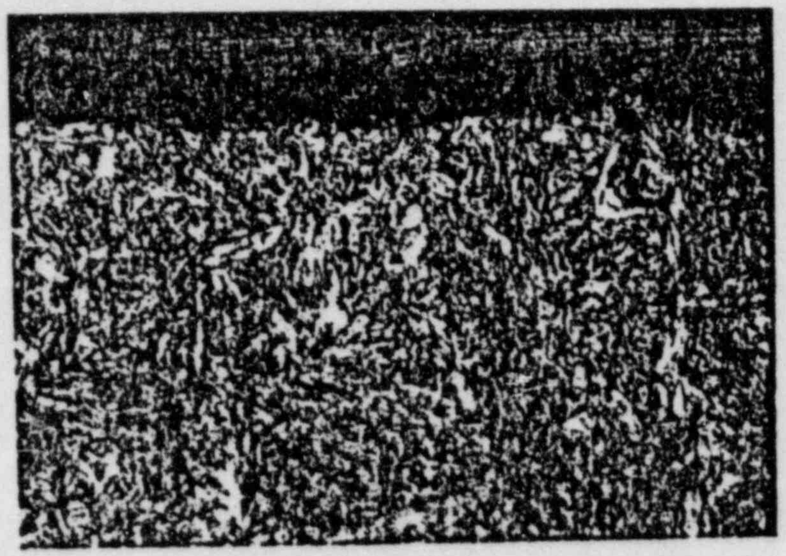
| Specimen Number | No. of Shot-Peening and Plating Cycles | Nominal (Pseudo-Elastic) Surface Stress, MPa | Test Duration, h | Result[a] |
|---|---|---|---|---|
| 11 | none | 1415 | 258 | N |
| 12 | none | 1415 | 257 | N |
| 13 | none | 1415 | 279 | N |
| 14 | none | 1415 | 279 | N |
| 16 | 1 | 1415 | 259 | N |
| 23 | 1 | 1415 | 259 | N |
| 18 | 2 | 1655 | 214 | N |
| 21 | 2 | 1655 | 209 | N |
| 19 | 3 | 1655 | 209 | N |
| 24 | 3 | 1655 | 213 | N |
| 17 | 4 | 1655 | 215 | N |
| 22 | 4 | 1655 | 215 | N |
| 15 | 5 | 1655 | 200 | N |
| 20 | 5 | 1655 | 200 | N |

[a]N = No cracking observed; test terminated.

TABLE 2—*Stress corrosion results—fatigue precracked specimens.*[a]

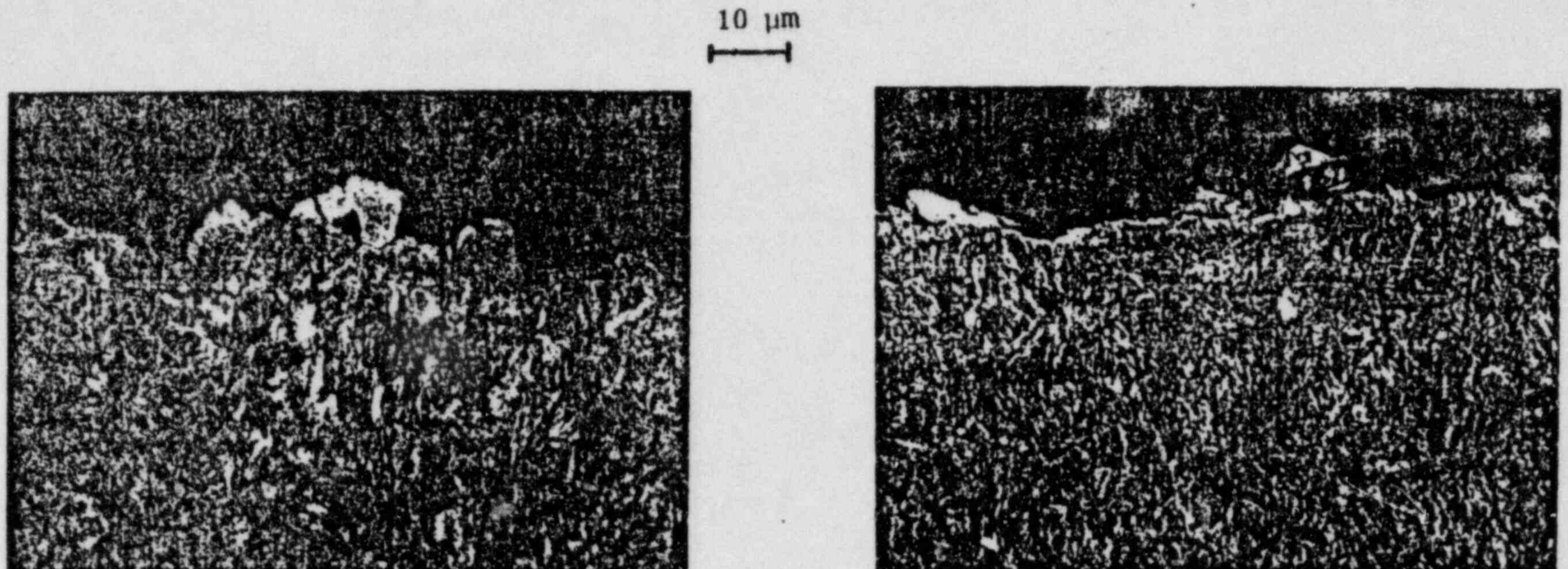
| Specimen Number | No. of Shot Peening and Plating Cycles | Nominal (Pseudo-Elastic) Surface Stress, MPa | Nominal Surface Stress Intensity Factor,[b] $MPa \cdot m^{1/2}$ | Test Duration, h | Result[c] |
|---|---|---|---|---|---|
| 9 | none | 1415 | 46 | 266 | N |
| 10 | none | 1415 | 46 | 266 | N |
| 9[d] | none | 1550 | 50 | 216 | N |
| 10[d] | none | 1655 | 54 | 214 | F |
| 7 | 1 | 1655 | 54 | 362 | N |
| 8 | 1 | 1655 | 54 | 350 | F |
| 6 | 2 | 1655 | 54 | 213 | N |
| 3 | 3 | 1655 | 54 | 233 | N |
| 5 | 3 | 1655 | 54 | 204 | F |
| 1 | 4 | 1655 | 54 | 42 | F |
| 2 | 5 | 1655 | 54 | 97 | F |
| 4 | 5 | 1655 | 54 | 2.2 | F |

[a]Precracked nominal crack length = 2.5 mm.
[b]Calculated per A. F. Grandt, Jr., and G. M. Sinclair, *Stress Analysis and Growth of Cracks, ASTM STP 513*, American Society for Testing and Materials, 1972, pp. 37-58.
[c]N = No crack extension observed (precracked specimens); test terminated. F = specimen fractured.
[d]Retest of a specimen from a terminated test at a lower stress.



Baseline: As-Ground

After One Shot Peening/Plating Cycle




After Three Shot Peening/Plating Cycles

After Five Shot Peening/Plating Cycles

FIG. 5—*Metallographic sections through AISI 4340 steel specimen surfaces; all sections parallel to grinding direction.*

white or light etching layers and stringers may be attributable to a high degree of surface plastic deformation. The thin layers probably represent highly deformed material rather than untempered martensite, which has a similar appearance.

In addition to the preceding general characterization of surface features, a metallographic study was performed on several failed test specimens in an attempt to ascertain whether or not the observed white layer influenced the failure process. The specimens selected for this study represented parent or baseline material and extremes in test life for various fatigue and stress corrosion test conditions.

Before proceeding with metallographic examination of the test specimens, a test blank and the two baseline specimens were macro-etched to investigate whether or not any significant grinding burn had occurred. This was done in order to resolve the issue of whether the presence of a white layer could be traceable to machining in the manufacture of the specimens. The three specimens were etched by a multi-step procedure widely used in industry, which consisted of a dilute solution of 4 percent nitric acid in water and a solution of 2.5 percent hydrochloric acid in acetone. One of the parent specimens was also etched with a 2 percent Nital solution. None of these etching techniques revealed the presence of grinding burn on the specimens.

The test specimens were first examined on a binocular microscope at magnifications of up to approximately ×40 in order to locate failure origins. Examination of the fatigue specimens revealed that failure origins were located either at one of the corners of the specimen or on the sides of the specimen. Failures in the stress corrosion specimens initiated from the pre-existing fatigue crack that was introduced at the bottom of the EDM notch.

Metallographic sections were made approximately through the center of each failure initiation site and examined in the unetched and etched conditions at magnifications up to approximately ×1200. Observations indicated that the white layer was not associated exclusively with the initiative area of specimens exhibiting the lowest fatigue lives. Fatigue initiation was apparently also influenced by other forms of surface degradation, such as microcrack and slivers, and by specimen geometry (that is, the corner areas).

## Conclusions

Specific conclusions from experimental results were as follows:

1. Shot-peening/cadmium-plating cycles up to five in number had no influence on tensile properties relative to those from as-heat-treated material.

2. Fatigue resistance in high humidity air at stress ratios $R$ of 0.1 and $-0.3$ was enhanced by shot-peening/cadmium-plating cycles up to five in number. The increase was most noticeable after one to three such cycles.

3. Stress corrosion results from unnotched specimens in high-humidity air were inconclusive since both as-heat-treated and shot-peened/cadmium plated specimens survived 200-h exposure at up to a 1650 MPa elastic surface stress level without cracking.

4. Fatigue precracked stress corrosion specimens subjected to four and five shot-peening/cadmium-plating cycles exhibited shorter lives than as-heat-treated specimens and specimens subjected to fewer shot-peening/cadmium-plating cycles. All specimens were fatigue precracked to a surface crack length of about 2.5 mm after heat treating, prior to any shot-peening/plating cycles. Stress corrosion testing of precracked specimens was performed in 293 K, 80 to 100 percent relative humidity air at a pseudo-elastic surface stress level of 1650 MPa.

5. No microstructural changes of significance relative to mechanical properties were observed to result from shot-peening/cadmium-plating cycles. White stringers observed metallographically at the surface tended to increase in prominence with increasing cycles. These stringers were believed to be an etching phenomenon related to plastic deformation in the peened surface layers.

## *Acknowledgments*

Sponsorship of this work by the Air Force Wright Aeronautical Laboratories/Material Laboratory, AFWAL/MLSA, under Contract F33615-78-C-5201 is gratefully acknowledged. One of the co-authors, A. W. Gunderson, of the Materials Integrity Branch, Systems Support Division, served as project monitor. Also acknowledged are the contributions of various Metcut Research Associates personnel, in addition to the co-authors, who were instrumental in performance of the experimental work: W. J. Stross, Tensile and Fatigue Testing; L. R. Gatto, Metallography; and T . E. Arnold, Stress Corrosion Testing.

## References

[1] Koster, W. P. et al., "Surface Integrity of Machined Structural Components," AFML-TR-70-11, March 1970.

[2] Koster, W. P. et al., "Surface Integrity of Machines Materials." AFML-TR-74-60, April 1974.

EXHIBIT 6

XEROX TELECOPIER 495;22- 8-84; 9:01AM ; → 5093752718;# 1

P. MARTIN

500 80th Avenue
P.O. Box 2161
Oakland, California 94621
(415) 577-7400

TDI #1 ✓

TELECOPY

TO: WALTER LAITY - RUSH

May 4, 1984 FROM: BRUCE GERMANO

Stone and Webster Engineering Corporation
P.O. Box 604
N. Country Road
Wading River, N.Y. 11782

Attention: Mr. Ralph Jaquinto

Subject: Long Island Lighting Company
Shoreham Nuclear Power Station
Transamerica Delaval Engines S/N 74010/12

Reference: Your letter dated May 1, 1984

Gentlemen:

Transamerica Delaval has revised the cylinder liner, T.D.I. P/N 03-315-02-OE, design dimensions to increase clearances between the liner and block at the upper collars and reduce liner proudness. These revisions can be incorporated at Shoreham in engine S/N 74012 by reworking the cylinder liners in Oakland when they are returned for hydrostatic testing of the new cylinder block. Should Long Island Lighting Company elect to have the liners reworked, the following would be performed:

Reduce upper collar diameter from 19.501/19.499 to 19.496/19.494

Reduce lower collar diameter from 18.997/18.995 to 18.992/18.990

Reduce upper collar height from 1.507/1.505 to 1.5015/1.5005

Reestablish gasket groove depth to .101/.099

IN ACCORDANCE WITH M. LOWERY VIA TALCON 5/8 GM

Please review this information and notify our Parts Department of your decision so that scheduling of rework can be initiated.

If this office can be of any further assistance, please do not hesitate in contacting us.

Very truly yours,

ATTACHMENT TO E & D.CR # F-44463B
PAGE 3 OF 4

Robert Johnston
Supervisor, Customer Service Engineers

RJ/tg

EXHIBIT 7

# Facsimile Message

RICARDO
Fax No. 079 17 64124

From GEORGE MURRAY / C GRAY
Date 10 . 8 . 84

To BATTELLE NORTH WEST LAB.


Fax No. 509 375 3641

For The Attention of DR. DAVID DINGEE

Subject SUFFOLK COUNTY SUBMISSIONS (CONT)

PISTON TIN PLATING

TIN PLATING OR 'TINNING' CAN BE APPLIED TO PISTON CAST IRON SKIRTS FOR CONSERVATION PURPOSES. I.E. TO AVOID RUSTING DURING STORAGE & TRANSIT. THIS MAY BE A MINIMAL COATING AND WOULD BE CONSIDERED SACRIFICIAL WHEN RUNNING IN THE ENGINE.

A SECOND USE IS TO ASSIST IN THE INITIAL RUNNING-IN PERIOD WHERE A POTENTIAL SCUFFING SITUATION EXISTS. THIS SHOULD HAVE BEEN DETERMINED ON THE MANUFACTURERS TEST BEDS DURING PROTOTYPE TESTING.

THE THICKNESS OF THE COATING WOULD BE 1 - 2 μm (·00004" - 00008") ONLY.

THICKER COATINGS HAVE BEEN USED (UP TO ·002 - ·003") THICK ON 2 STROKE HEAVY FUEL ENGINES BUT THE DANGER ARISES WHEN DEBRIS IS PICKED UP IN THE COATING AND DAMAGES THE LINER. ANOTHER PROBLEM IS THAT THE TIN CAN BALL-UP.

WE DON'T BELIEVE TIN PLATING SHOULD BE USED ON THE SKIRT TO COVER UP DEFICIENCIES IN THE HOT PROFILE TO IMPROVE NORMAL RUNNING CONDITIONS. THE DIFFERENCE IN SHAPE BETWEEN

# Facsimile Message

RICARDO
Fax No. 079 17 64124

From

Date

To

Page 2 Of

Fax No.

For The Attention of

Subject SUFFOLK COUNTY SUBMISSIONS (CONT.)

COLD AND HOT SKIRTS WOULD BE FAR GREATER THAN CAN BE TAKEN UP BY THE TIN PLATE.

TIN PLATING MAY BE AN AID IN THE COLD STARTUP PROCEDURE TO ASSIST IN THE POOR LUBRICATION AREAS. A PARTIAL SEIZURE AND HOT PISTON MAY BE AVOIDED IF THE TIN WEEPS AND ASSISTS LUBRICATION.

ONE PISTON MANUFACTURER ADVISES AGAINST TIN PLATING, STATING THAT FEW ENGINE BUILDERS ARE SPECIFYING TIN PLATING.

THE PREFERRED ALTERNATIVE IS GRAPHITE COATINGS.

A GRAPH IS ENCLOSED . PAGE WHICH SHOWS PISTON SCUFFING SHOULD NOT OCCUR ON THIS ENGINE AT THIS RATING AND SPEEDUNDER NORMAL CONDITIONS

RICARDO
CONSULTING ENGINEERS

P. 3.

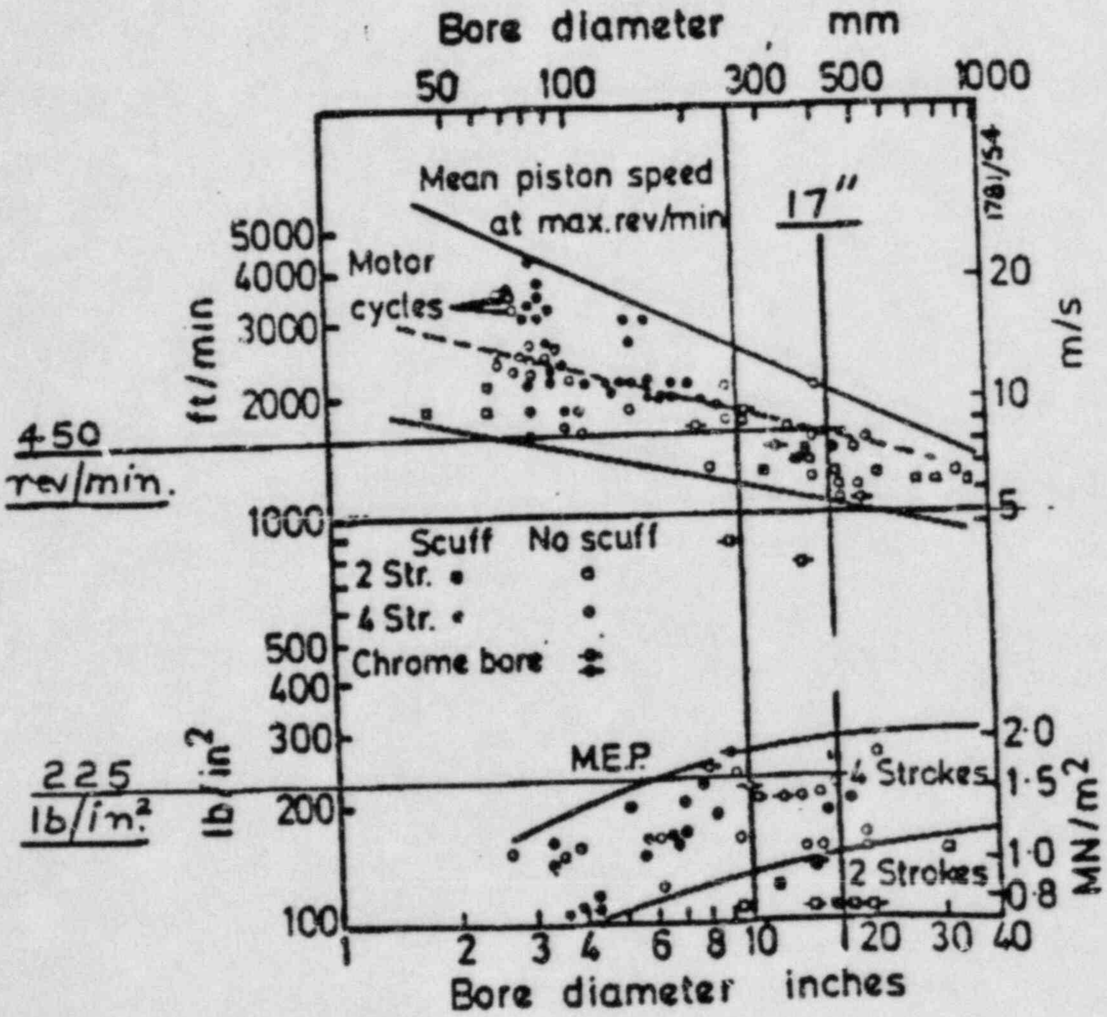



Fig 4 Incidence of Scuffing

(Reference identified in Ricardo telex number 9183 dated August 20, 1984, which is enclosed as an attachment to this facsimile message)

# Facsimile Message

RICARDO

Fax No. 079 17 64124

From C. GRAY

Date

To

Page 4 Of

Fax No.

For The Attention of

Subject SUFFOLK COUNTY SUBMISSIONS (CONT.)

CRANKSHAFT SAFETY FACTOR ACCORDING TO PROPOSED CIMAC RULES

BASED ON DRG. No. 03-310-05-AC
MAX. CYLINDER PRESSURE 1700 PSI
BMEP 206 PSI
GENERATOR OUTPUT 3200 kW
SPEED 450 rev/min
RECIPROCATING MASS 364 kg
ROTATING MASS 202 kg
NOMINAL TORSIONAL (SHEAR) STRESS 6224 PSI
UTS OF STEEL 695 $N/mm^2$

THE CALCULATED SAFETY FACTOR IS 0.91. THE PROPOSED RULES SPECIFY THAT IT SHOULD NOT BE LESS THAN 1.1, THUS THE CRANKSHAFT DOES NOT COMPLY WITH THE RULES AT THIS OPERATING CONDITION.

IF THE BENDING STRESSES ALONE ARE CONSIDERED THE SAFETY FACTOR IS 1.09 AND THIS IS STILL JUST BELOW THE MINIMUM ALLOWABLE VALUE, SHOWING THAT THE BENDING STRESS IS THE MAIN FACTOR AFFECTING THE SAFETY FACTOR.

# Facsimile Message

RICARDO
Fax No. 079 17 64124

From ______________________ Date ______________

To ______________________ Page 5 Of __

Fax No. ______________________

For The Attention of ______________________

Subject SUFFOLK COUNTY SUBMISSIONS (CONT)

PISTON SIDE THRUST

WE DO NOT, FOR INITIAL DESIGN USE PISTON THRUST PRESSURE AS A CRITERIA. IT IS MORE NORMAL FOR US TO PROPORTION THE PISTON FOR ITS DUTY, APPLICATIONS, FUELS & SPEED.

FOR THIS TYPE OF DUTY WE WOULD LOOK FOR LONG LIFE. LONG PISTONS ASSIST IN THIS AIM BY MINIMISING TILTING & THEREFORE REDUCE EDGE LOADING AT TOP AND BOTTOM OF SKIRT. LOW SPEEDS TEND TO ALLOW PISTONS TO BE LONGER, - LESS INERTIA AT LOW SPEED.

TO COMPARE THE TDI PISTON SIDE PRESSURE WITH COMPARABLE ENGINES WOULD REQUIRE ACCESS TO FULL TECHNICAL DATA INCLUDING PRESSURE DIAGRAMS. WE HAVE DRAWN UP A CHART ON WHICH ARE COMPARED PISTONS OF SEVERAL ENGINES AGAINST THEIR OPERATING CONDITIONS.

COMPARING PISTON. 1 (TDI ENGINE) TO PISTON. 7, THE PISTON PIN IS PROPORTIONATELY HIGHER IN PISTON 7. WHIC IS RELIABLE.

COMPARING PISTON 1 TO PISTONS 3 & 4 & ADDING R/B + S/B, THE SKIRTS ARE SIMILAR OR SHORTER. PISTON 3 HAS A VERY HIGH FIRING PRESSURE (A RELATIVELY NEW ENGINE)

# Facsimile Message

RICARDO
Fax No. 079 17 64124

From ___

Date ___

To ___

Page 6 Of ___

Fax No. ___

For The Attention of ___

Subject SUFFOLK COUNTY SUCMISSIONS (CONT)

| | CYL. BORE | L/B | C/B | R/B | S/B | $\alpha^\circ/B^\circ$ | MAX. FIRING. P. lb/in² | RATED BHP/CYL. |
|---|---|---|---|---|---|---|---|---|
| T.D.I. R4 → 1. | 432 | 1·36 | ·76 | ·36 | ·64 | 35/48 | 1750 | 677 |
| 2. | 420 | 1·56 | ·96 | ·45 | ·57 | 35/48 | 1710 | 70[illegible] |
| 3. | 400 | 1·47 | ·89 | ·47 | ·53 | 43/48 | 2030 | 750 |
| 4. | 381 | 1·34 | ·81 | ·45 | ·52 | 42/46 | 1620 | 660 |
| 5. | 400 | 1·57 | ·93 | ·87 | ·65 | 45/52 | 1650 | 650 |
| 6. | 410 | 1·59 | ·93 | ·49 | ·64 | 45/52 | 1810 | 765 |
| 7. | 438 | 1·6 | ·81 | ·37 | ·75 | 37/53 | 1600 | 530 |

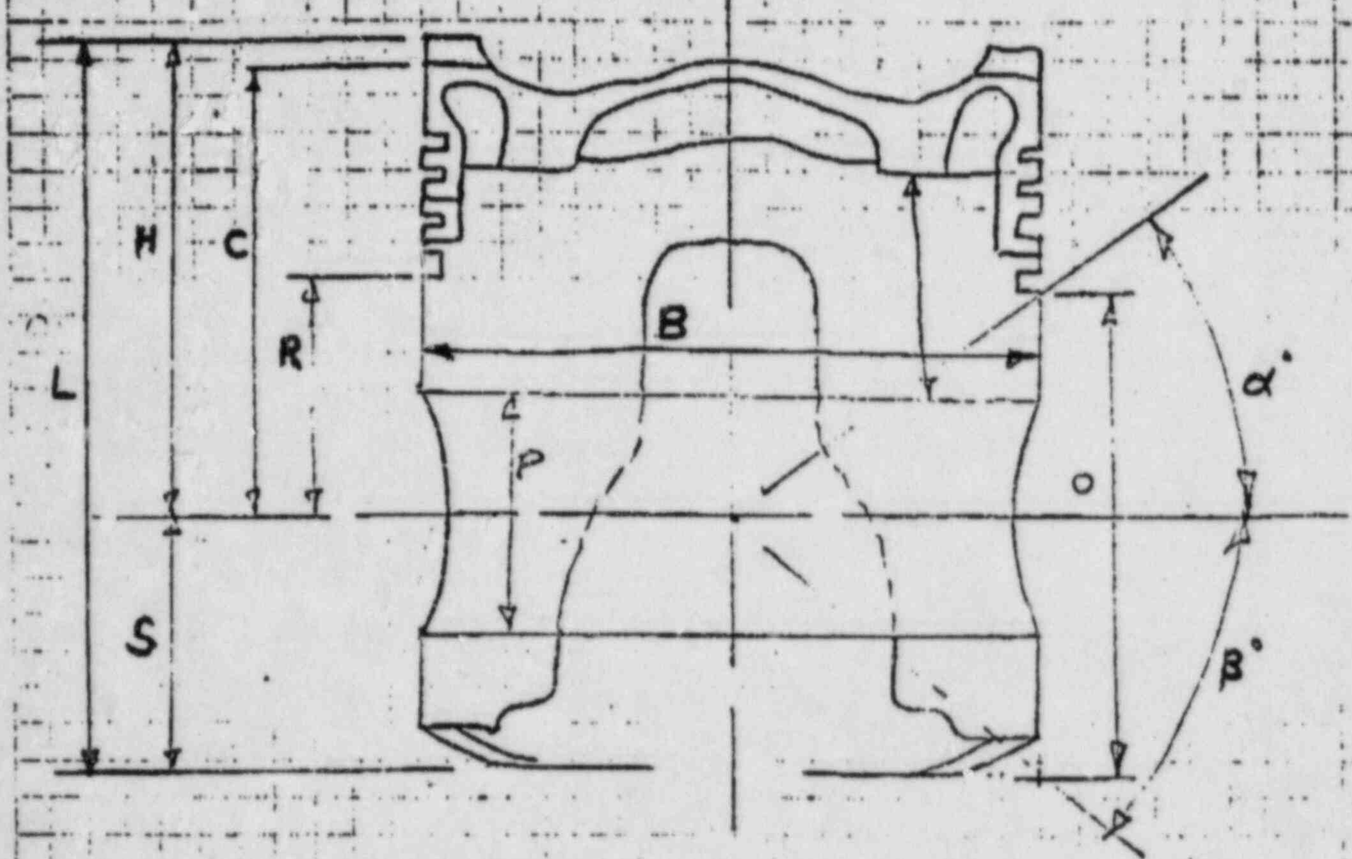
(Engines corresponding to numbers are identified in Ricardo telex number 9183 dated August 20, 1984, which is enclosed as an attachment to this facsimile message)

RICARDO CALCULATION

REF No

PROJECT SHEET OF P 7

SECTION



# Facsimile Message

RICARDO
Fax No. 079 17 64124

From

Date

To

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Fax No.

For The Attention of

Subject SUFFOLK COUNTY SUBMISSIONS (CONT)

AND THE SIDE PRESSURE WILL BE COMPARATIVELY HIGHER.

PISTON. 3. REFLECTS THE TREND OF NEWER ENGINES WHICH ARE NOT EXCESSIVE IN LENGTH AND ARE DESIGNED FOR MUCH HIGHER PRESSURES (UP TO 2500 LB/IN$^2$)

THE SKIRT LENGTHS ON PISTON 1 & 2 ARE SIMILAR, $\alpha°$ & $\beta°$ ARE ALSO SIMILAR AND FIRING PRESSURE IS SIMILAR. PISTON 2 IS NOT KNOWN TO BE UN-RELIABLE.

WE CAN SEE NO FEATURE OF PISTON. 1. (TDI ENGINE) THAT INDICATES THAT FAILURE SHOULD OCCUR FROM PISTON PROPORTIONS, SKIRT LENGTHS OR HEIGHT OF PISTON PIN.

THERE ARE PISTONS ON OTHER ENGINES WITH MORE EXTREME PROPORTIONS THAN THESE WHICH HAVE LONG PRODUCTION RUNS.

THRUST PRESSURES WILL UNDOUBTABLY INCREASE WITH TIME. THE FIGURE QUOTED IN 1949 BY WALSHAW 85 LB/IN$^2$ PROBABLY RELATES TO NORMALLY ASPIRATED ENGINES OR WITH MILD TURBO CHARGING. CYLINDER PRESSURE TODAY ARE DOUBLED FROM THAT PERIOD. WITH SIMILAR

# Facsimile Message

RICARDO
Fax No. 079 17 64124

From

Date

To

Page 2 Of

Fax No.

For The Attention of

Subject SUFFOLK COUNTY SUBMISSION (CONT)

PROPORTIONED ENGINES SIDE THRUSTS MAY ALSO BE DOUBLED, 123 LB/IN$^2$ IS 50% INCREASE OVER 1949 FIGURE. THESE MAXIMUM PRESSURES ARE NOT CONSTANT, BEING THE COMBINATION OF INERTIA AND FIRING PRESSURES AND CON ROD ANGLE.

HIGH PRESSURES ARE IMPORTANT ONLY WHEN THE MOMENTARY LOCAL LUBRICATING CONDITIONS CANNOT BEAR IT. LUB.OIL FILM CONDITIONS NEED EXAMINATION TO PURSUE THIS.

THE HIGH PRESSURE OCCURS WHEN THE PISTON HAS A REASONABLY HIGH VELOCITY & A HYDRODYNAMIC OIL FILM IS GENERATED.

'BANANERING' OF THE PISTON SKIRT SHOULD ALREADY BE REFLECTED IN THE PROFILE OF THE SKIRT SELECTED BY TDI IN THE ENGINE TESTS FOR TYPE TESTING. POSSIBILITY OF SEIZURE FROM HOT PISTON SKIRT SHAPE SHOULD HAVE BEEN EXAMINED IN THE SAME TESTS.

THE DIFFERENCES IN CYL. PRESSURES DISCUSSED 1670 – 1800 LB/IN$^2$ IS LITTLE MORE THAN INDIVIDUAL CYLINDER PRESSURE SCATTER.

# Facsimile Message

RICARDO
Fax No. 079 17 64124

From

Date

To

Page 10 Of 10

Fax No.

For The Attention of

Subject SUFFOLK COUNTY SUBMISSIONS (CONT)

A GREATER CYLINDER PRESSURE MAY BE OBTAINED BY DETONATION ON COLD ENGINE START UP. WITH A FAST COLD START REQUIRED, THE CONDITIONS OF COLD STARTING AIR, COLD ENGINE AND EXCESS FUEL MAY CAUSE DETONATION IN THE PERIOD THE TURBOCHARGER IS ACCELATING TO FULL BOOST, CAUSING CYLINDER PRESSURES TO RISE CIRCA 25% OVER NORMAL. THIS MAY CAUSE PISTON DAMAGE

HAVE START UP PRESSURES BEEN MEASURED? DOES THE ENGINE DETONATE?

WE WILL CONTINUE ASSESSMENT.

G.E. [signature]

87383 RICSHM G

TELEX OFFICE
AUG 20 1984

TELEX NUMBER 9183 20TH AUGUST 1984

FOR THE ATTENTION OF MR.W.W.LAITY

REFERENCES
A) - GRAPH ON PAGE 3 OF MY FACSIMILE MESSAGE IS FROM-PISTON RING SCUFFING, A BROAD SURVEY OF PROBLEMS AND PRACTICE. M.J.NEALE - INSTITUTION OF MECHANICAL ENGINEER PROCEEDINGS 19708- 71. VOLUME 158.2/71.

WE USE THIS GRAPH FOR REFERENCE PURPOSES IN ASSESSMENTS AND TO ASSIST IN FUNDAMENTAL DESIGN DECISIONS ON NEW ENGINES. IT IS VERY DIFFICULT TO RELATE ENGINE RELIABILITY TO ENGINE SPEED ESPECIALLY AS ENGINE DEVELOPMENT VARIES CONSIDERABLY BETWEEN MANUFACTURERS. THE SURVEY DOES SHOW HIGH PISTON SPEEDS TO BE MORE PRONE TO SCUFFING.

B) ALL DATA IN TABLE ON PAGE 6 CAN BE TAKEN FROM PUBLISHED DATA AND MEASUREMENT. SMALL VARIATIONS MAY OCCUR DETAIL DIMENSIONS AND MAX. FIRING PRESSURE DUE TO LATEST SOURCE VARIANCES.

ENGINES ARE:-
1.TRANSAMERICA DELAVAL R4.
2. G.M.T. 420
3. MAN 40/45
4. MIRRLEEES 'K' MAJOR
5. PIELSTICK PC2
6. SWD TM410
7. NOT DEFINED.

REGARDS
GEORGE MURRAY/RICARDO CONSULTING ENGINEERS +
BATTELLE RCLD

87383 RICSHM G

UNITED STATES OF AMERICA
NUCLEAR REGULATORY COMMISSION

BEFORE THE ATOMIC SAFETY AND LICENSING BOARD

DOCKETED
USNRC

'84 SEP -5 P2:35

OFFICE OF SECRETARY
DOCKETING & SERVICE
BRANCH

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

## CERTIFICATE OF SERVICE

I hereby certify that copies of "JOINT TESTIMONY OF CARL H. BERLINGER, SPENCER H. BUSH, ADAM J. HENRIKSEN, WALTER W. LAITY, AND PROFESSOR ARTHUR SARSTEN ON CONTENTIONS CONCERNING TDI EMERGENCY DIESEL GENERATORS AT THE SHOREHAM NUCLEAR POWER STAITON" (2 volumes) in the above-captioned proceeding have been served on the following by deposit in the United States mail, first class, or, as indicated by an asterisk, through deposit in the Nuclear Regulatory Commission's internal mail system, or, as indicated by a double asterisk, by hand delivery, this 30th day of August, 1984:

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U.S. Nuclear Regulatory Commission
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2300 - 6th Street, N.W.
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