TECHNICAL EVALUATION REPORT

EVALUATION OF DIESEL GENERATOR FAILURE
AT SHOREHAM UNIT 1

FINAL REPORT, FAILURE CAUSE EVALUATION

NRC DOCKET NO. 50-322

NRC TAC NO. --

NRC CONTRACT NO. NRC-03-81-130

FRC PROJECT C5506

FRC ASSIGNMENT 20

FRC TASK 426

Prepared by

Franklin Research Center
20th and Race Streets
Philadelphia, PA 19103

Prepared for

Nuclear Regulatory Commission
Washington, D.C. 20555

Author: R. C. Herrick

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Lead NRC Engineer: R. J. Giardina

April 6, 1984

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U.S. Nuclear Regulatory Commission
Washington, D.C. 20555

Attention: Mr. M. Carrington (MS-540)
Project Officer

Subject: FRC Project C5506
NRC Contract NRC-03-81-130
FRC Assignment 20
FRC Task No. 426
Title: Evaluation of Failure Cause, Diesel Generator Failure
Shoreham Unit 1

Dear Mr. Carrington:

The attached report presents FRC's technical review of the investigation of the diesel generator failure at Shoreham Unit 1.

Submittal of this report completes FRC's efforts on Task 426 of this assignment.

Very truly yours,

S. Pandey

S. Pandey
Project Manger

SP/SA/cm
Enclosure

cc: R. J. Giardina (1 copy)
R. Caruso (1 copy)
C. Berlinger (1 copy)

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Prepared by:	Reviewed by:	Approved by:
R. C. Herrick		
Principal Author	Project Manager	Department Director (Acting)
Date: 4/3/84	Date: 4/3/84	Date: 4/3/84

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FOREWORD

This Technical Evaluation Report was prepared by Franklin Research Center under a contract with the U.S. Nuclear Regulatory Commission (Office of Nuclear Reactor Regulation, Division of Operating Reactors) for technical assistance in support of NRC operating reactor licensing actions. This report constitutes the final report of the two-phase effort.

1. INTRODUCTION

In August 1983, a crankshaft of one of three emergency diesel generators (DG 101, 102, and 103), manufactured by Transamerica Delaval, Inc. (TDI), and installed at the Shoreham Nuclear Power Station, owned by the Long Island Lighting Company (LILCO), fractured during plant preoperational diesel generator tests. Inspection revealed severe cracking in the crankshafts of the other two diesel generators.

During the failure investigation that followed, Failure Analysis Associates (FaAA) and Stone and Webster Engineering Corporation (SWEC) were engaged to carry out intensive analytical and experimental investigations. Early inspection and evaluation indicated that of the two remaining diesel generators (DG 101 and DG 103), sufficient crankshaft operational life would be available from diesel generator DG 101 for an instrumented operational test program if the cracks in the crankshaft were ground out. An operational test program was planned, and operational tests were completed on September 28, 1983. In the meantime, the two diesel generators that could not be operated were disassembled for detailed inspection and rebuilding. Sections of the fractured crankshaft from diesel generator DG 102 were taken to FaAA laboratories for metallurgical examination of the fracture.

The Nuclear Regulatory Commission (NRC) requested that Franklin Research Center (FRC) provide an independent technical review of the failure investigation performed by the Licensee and thereby provide a technical basis for the NRC's licensing actions regarding these failures.

Phase I included the following:

a. attend an onsite inspection and review of the Shoreham diesel crankshaft failure, and review operation and maintenance history provided by the Licensee

b. analyze the data and information obtained in the onsite visit and prepare an interim report providing initial findings and conclusions regarding the events leading to crankshaft failures

c. provide in the above any conclusions about other mechanical problems the Licensee has had with these diesel generators.

Phase II included:

a. review and evaluate submittals and data provided by the Licensee on the causes of these failures

b. provide technical assistance to the NRC lead engineer in evaluation of applicable data on diesel generators provided by the staff.

Accordingly, FRC participated in the onsite inspection, reviewed test procedures and diesel generator operating history, reviewed the crankshaft design analysis methods employed by the engine manufacturer, and participated as an active observer in the operational testing program prior to submitting an interim report [1]* covering Phase I. Subsequently, FRC reviewed the metallurgical examinations, diesel engine standards, specifications and design rules, and crankshaft design and analysis methods available to the industry, as well as performed a review of the testing methods, data output, and failure analysis conclusions of the operational testing program. In addition, the design and analysis of the replacement was reviewed.

This report includes the salient features of the interim report (Phase I) and also reports on the subsequent events. In addition, it includes, as Appendix E, the commentary submitted by Mr. H. W. Hanners regarding selected problems experienced by the diesel engines manufactured by TDI. Mr. Hanners, an independent diesel engine consultant who was co-author of NUREG/CR-0660, "Enhancement of On-Site Emergency Diesel Generator Availability," participated with FRC in the initial onsite inspection.

*Numbers in brackets refer to references found in Section 7.

2. ACCEPTANCE CRITERIA

Diesel generators are manufactured and purchased in accordance with various industry standards. These standards include "Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines" by the Diesel Engine Manufacturers Association (DEMA) [2], "Rules for Building and Classing Steel Vessels" by the American Bureau of Shipping [3], and "Rules and Regulations for the Classification of Ships" [4], which includes "Guidance Notes on Torsional Vibration Characteristics of Main and Auxiliary Oil Engines" [5] by Lloyds Register of Shipping. Other rules and standards are available from European diesel manufacturer associations. In the absence of an ordered set of acceptance criteria for this review and evaluation, commentary regarding the applicable specifications and standards is included with the discussion rules, standards, and methodology in Section 4.2.1 of this report.

3. PRELIMINARY INSPECTION AND REVIEW

3.1 ONSITE INSPECTION

3.1.1 Preliminary Briefing

On September 1, 1983, a preliminary briefing about the current state of events and plan of action [6] at the Shoreham plant was held in the NRC Resident Inspector's office; a brief overview of the performance history of the three diesel generators was included. The briefing was conducted by the NRC Senior Resident Inspector and supplemented by the Director of LILCO's Office of Nuclear Power.

3.1.2 Inspection Tour of Diesel Generators

A visual inspection was made of the three diesel generators. Although the observations are described in Appendix A, a brief summary follows:

- Diesel generator 101 was located in its operational room and was being prepared for a limited test program. LILCO reported that this unit had performed the initial qualification testing program for nuclear plant service and had been subjected to dynamic torsional testing as a part of that program. However, cracks were observed in the crank pin fillets of cranks 5 and 7.

- Diesel generator 102 was the unit with the fractured crankshaft. The diesel engine and generator had already been moved to the main turbine deck where space and crane facilities were available to disassemble the unit, make a thorough inspection, and rebuild it with the 13 x 12 crankshaft now recommended by TDI. Considerable attention was paid to the crankshaft fracture in this inspection because of the imminence and magnitude of the failure. The entire engine was also studied to gain a perspective necessary for an adequate review of the many types of failures experienced previously by the Shoreham diesels in order to determine if there may be a root cause not evidenced by the July 1983 study [7].

- Diesel generator 103, reported to have crankshaft cracks developed to an extent that precludes further engine operation, was observed in its operational room. It was being prepared for movement to the main turbine deck for disassembly, inspection, and reassembly with the 13 x 12 crankshaft.

3.1.3 Preparation of Requests for Information

A meeting with the NRC representatives was attended at the Shoreham plant on September 1, 1983, during which the immediate and past problems experienced by the diesels at the Shoreham plant and their implication for similar diesels at other plants were discussed. Questions were prepared concerning the aspects of the diesels and their performance records that would be required for an adequate independent evaluation. These questions were submitted to LILCO by the NRC during the public meeting held at the Shoreham plant on September 2, 1983.

3.1.4 Public Meeting

Onsite activities included attendance at the public meeting held at the Shoreham plant on September 2, 1983 for discussion of the diesel engine problems. Representatives of the following organizations were in attendance:

Nuclear Regulatory Commission
Long Island Lighting Company
Hunton and Williams, LILCO legal counsel
Stone and Webster
Failure Analysis Associates
Counsel and Technical Consultant for Suffolk County
Newsday
Franklin Research Center.

During the meeting, the following points were established:

- There is no nuclear fuel at the Shoreham plant and consequently there is no demand on the safety systems.
- There is concern for similar diesels in other nuclear power plants.
- The problems with the Shoreham diesels are broader than the present crankshaft problem.
- The tests on the Shoreham diesels are significant for the whole nuclear power industry.
- The failure analysis team consists of LILCO, FaAA, and Stone and Webster Corporation.
- TDI is cooperating with the failure analysis team to provide disassembly, inspection, alternate crankshafts and rework as necessary, and reassembly of the engines.

- o TDI management is committed to the failure analysis and engine rebuilding program and will submit its own assessment and recommended actions; for objectivity, however, the program is under the direction of an independent investigator, FaAA.
- o Concern of the community is high as represented by counsel for Suffolk County and a technical consultant.
- o A comprehensive failure analysis effort will be carried out to fully understand the failures so that the corrective action will be most effective.

3.2 PRELIMINARY TECHNICAL REVIEW AND EVALUATION

3.2.1 Review of LILCO'S Master Plan

A copy of LILCO's master plan [6] for the failure analysis and recovery of the diesel engines was received and reviewed. Comments [8] were submitted to the NRC, indicating where the reviewer's direct participation as an observer would be advisable. The plan was found to be acceptable.

3.2.2 Review of Test Procedures

An early copy of LILCO's test procedure [9] for operational testing of DG 101 was also received. The procedure was reviewed and a copy was forwarded to Mr. H. W. Hanners, an independent diesel engine consultant. Commentary and recommendations of this review were combined with those of Mr. Hanners and reported to the NRC in early September 1983 before the start of operational tests. In the course of operational testing of DG 101, the original procedure and three revisions [10, 11, 12] were reviewed.

Recommendations for modifications and additions to the last revision [12] to the procedure were made on September 24, 1983, by a memo [13] submitted via the NRC Resident Inspector at the Shoreham plant for expediency. Although the technical aspects of these considerations are discussed at greater length in Appendix D of this report, the recommendations provided means to assure that (1) all voltage phase references would be available and known, (2) transients associated with attachment of the generator to the electrical grid and to

major electrical loads would be recorded, and (3) testing at a significant synchronous loading would be recorded for power factors ranging from 0.8 to 1.0. These considerations were included in the tests carried out on September 28.

3.2.3 Preliminary Review of Diesel Dynamics

Because the crankshaft failure and many of the earlier problems of the three diesels showed evidence of being associated with the dynamic response of the diesel engines, the torsional dynamics analysis summary prepared by TDI as a part of the original design effort was reviewed. These analyses, which were stated in the September 1 and 2 conferences at the Shoreham plant to be verified as sufficiently accurate by FaAA defined the equivalent mass-elastic torsional dynamic model of the mechanical system, including the flywheel and generator rotor. They included the calculated natural frequencies and critical speeds. This information was needed to form a basis of understanding by which the reviewers could evaluate the test data and recognize the response of the various vibratory modes to the engine excitation orders in the course of the test runs.

3.2.3.1 Review of the TDI Mass-Elastic Model

The mass-elastic model [14] employed by TDI to represent the dynamic natural frequencies and mode shapes of the engine is made up of 11 inertias and 10 torsional springs. The inertias, identified in their order of position from the gear case end of the diesel to the generator, are:

- o gear case and water pump inertia
- o eight equivalent inertias representing the piston, connecting rod, and rotating portion of the crankshaft for each cylinder assembly
- o flywheel
- o generator rotor.

Torsional springs equivalent to that portion of the crankshaft or generator rotor shaft were calculated by TDI for application between the inertias in the model.

TDI's analysis summary [14] indicated that the inertia for each crank assembly was the equivalent average inertia. This was an average of the real inertia comprised of a rotating crank, linear motion piston, and a connecting rod that combines both motions, all of which combine to form an inertia that varies with crank angle. Various methods are available to average these crank angle-dependent inertias to equivalent average inertias for use in the mass-elastic model yielding the natural frequencies. The detailed methods by which TDI calculated the equivalent inertia and torsional spring constant for each crank assembly were not evident from the analysis summary [9]. Information to identify the methods used was requested from TDI through NRC. Methods developed by the industry over many years have been known to yield generally reliable results for most engine designs. This is not to imply that TDI did not use these methods correctly, only that its detailed methods were not evident in the analysis summary.

For the interim report [1], it was assumed that the calculated inertias and springs, resulting natural frequencies, and critical speeds were sufficiently accurate. This premise was based upon the facts that qualification testing was reported to verify these frequencies and that FaAA reported in briefings during the Shoreham inspection visit that it had obtained virtually the same values using more comprehensive computer methods.

Using its model for the resonant frequencies, TDI calculated the participation of the various orders of known engine excitation and plotted the amplification factors of the more dominant excitation orders as shown in Figure 1, which is a reproduction of the TDI chart from Reference 14. Note that the I-4 (4th order) curve of amplification factor remains the single largest participant, providing more than double the amplitude of the 4 1/2th order during operation at 450 rpm. The excitation frequency of the 4th order is that of 4 times per revolution. This is the firing rate of the cylinders which generates a sharp and dominant excitation. In short, using its analysis, TDI expected to experience a large component of oscillatory torque at 30 Hz (engine firing rate) in the crankshaft. However, a large cyclic torque at 30 Hz is not necessarily bad, provided the associated stress levels

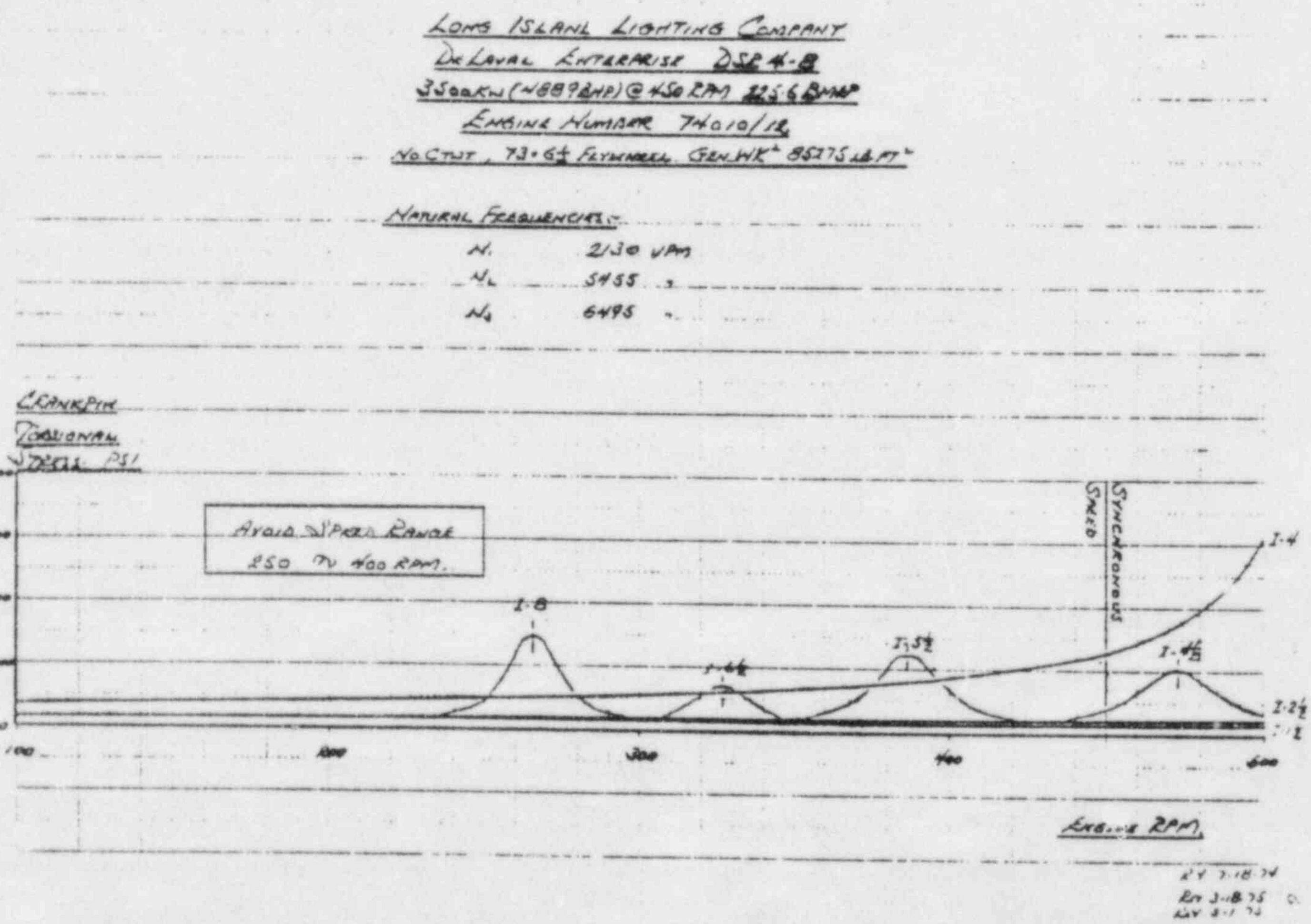

Figure 1. TDI Torsional Stress and Critical Speeds
[from Reference 14]

in combination with stresses from other sources in the crankshaft are adequately within the endurance limit of the material.

3.2.3.2 Investigation of Additional Constraints in the Torsional Mass-Elastic Model

The influence of rotor-stator electrical coupling upon the purely mechanical torsional model employed by TDI was investigated in this review. Rotor-stator coupling of a synchronous generator may be approximated as an equivalent spring rate between the generator rotor and the inertia of the electrical load. When the generator is connected to the electrical power grid, this equivalent inertia can be very large. In such cases, it is valid to approximate the effect by calculating the equivalent spring rate of the rotor-stator system and inserting it into the torsional mass-elastic model between the generator rotor inertia and a new fixed rigid member (infinite inertia).

Review of TDI's mass-elastic system revealed that:

- o The generator inertia was exceptionally large.
- o The flywheel inertia was only between one-third and one-half that of the generator rotor.
- o All other inertias representing crank assemblies, water pump, etc., were very small by comparison.

Thus, although the introduction of the rotor-stator equivalent spring had an exceedingly small effect upon the natural frequencies of the rotor and their dynamic response under the engine excitation, it did define a new mode of vibration not heretofore available from TDI's torsional model. This was essentially a rigid-rotor oscillation of the combined crankshaft, flywheel, and generator rotor system with the rotor-stator spring connecting the rigid-rotor system to the nearly infinite electrical load inertia mentioned previously. The natural frequency (resonance) of this vibratory mode was independently calculated to be approximately 3.0 Hz. This vibration mode contributed to the cyclic variation of power previously observed from the control room to be at 3.75 Hz, which is the frequency at which one complete

set of eight cylinders fires, or the rate at which any one cylinder fires. Fortunately, the natural frequency of just under 3.0 Hz was sufficiently far from the 3.75 Hz excitation to prevent large amplitude vibration with resulting large swings in power. Also, the amortisseur windings of synchronous generators provide damping under oscillatory motions to limit amplitude buildup.

For diesel generators that may be coupled to the electrical power grid, TDI should have addressed this electrical-mechanical, rotor-stator coupling.

3.2.3.3 Investigation of Other Mechanical-Electrical Dynamic Coupling

When received, the torsional analysis report [14] indicated that the 30-Hz firing rate of the engine (4th order) would be sufficiently close to the first mode natural frequency, 35.5 Hz, to build moderately high amplitudes of oscillation at 30 Hz, which is one-half the electrical generation frequency. Accordingly, recommendations were made [13] to ensure that any possible electrical-mechanical interaction would not be missed by the recording of data. These recommendations are included in Appendix C.

3.2.4 Review of Torsional Dynamics Testing

3.2.4.1 Initial Tests

The torsional dynamics testing program, with operation of instrumented DG 101, was conducted to establish correlation with a detailed computer dynamic model of the diesel formulated by FaAA, as well as to investigate the dynamic interaction of the diesel with various loadings and operational conditions. The torsional testing program was primarily concerned with the catastrophic failure of the crankshaft in DG 102 and near failure crack propagations in DGs 101 and 103. The testing was observed as a part of this review.

Listings of measured parameters, sensors and transducers, and data recording equipment are provided in the test procedures [9, 10, 11, 12]. These include most engine operational temperature and pressure data, i.e.,

lubrication temperature and pressure, combustion air pressure, each cylinder's exhaust pressure, etc.

Instrumentation for the measurement and recording of vital dynamic data included the following:

- o Cranks 5 and 7 were instrumented such that crankpin fillet and web dynamic strain were measured by three element strain rosettes bonded to the fillet and by a single gage on the crank web.
- o Dynamic torque in the crankshaft adjacent to the flywheel was measured by a strain gage torque bridge.
- o Cylinder firing pressure of cylinders 5 and 7 was measured with high-pressure piezoelectric transducers.
- o Shaft dynamic displacement was measured by a torsional displacement transducer mounted on the gear case end of the diesel crankshaft.
- o Linear acceleration of the engine base was measured by accelerometers mounted on the base at cylinders 5 and 7.
- o Vertical, horizontal, and axial acceleration (vibration) were measured for the bearing housing next to the flywheel.
- o Crankshaft position and revolution tachometer were referenced to top dead center of cylinder 7 provided by an optical sensor mounted on the generator shaft.
- o Generator output voltages were recorded to measure the voltage difference between phases, ($V_A - V_B$) and ($V_B - V_C$).
- o Generator output current was measured for individual recording of each phase.

Instrumentation on the rotating crankshaft was battery powered with signals transmitted by FM telemetry.

The initial tests were started on September 19, 1983, using the test procedure [11] dated September 15, 1983. Strain gage problems continued with gages dropping from service until five of the eight strain gages in the fillets and webs of the crankshaft were not operational. Testing was suspended at that point to repair the strain gage instrumentation. However, the test program had progressed through the initial checkouts, through the variable

speed torsiograph tests, and included the 1750-kW synchronous load test with the generator connected to the electric power grid. The full load tests, 3/4 load tests, and the TDI torsiograph tests remained to be accomplished.

3.2.4.2 Completion of Torsional Tests

Following repair and improvement of the strain gage instrumentation in the crank fillets and on the crank web, testing resumed at 2:44 am on September 28, 1983. These tests included the test program in the test procedure dated September 23, 1983 [12]. The test program, with instrumentation performing satisfactorily, continued to completion at approximately 7:30 am that same morning.

Testing began with Section 7.1 of the procedure [12], which involved measurements for verification of the analytic model at FaAA in Palo Alto, CA. This was a correlation procedure in which the initial dynamic measurements were telephoned to the FaAA offices and checked against the analytic (computer) model both to verify the model and to permit the model to predict the available run time on the engine before crack propagation would preclude further testing.

Testing continued through the balance of the test procedure, including measurements recommended prior to the test and in the course of this review, and concluded with tests requested by TDI using its own torsiograph and associated instrumentation.

Observations of data during the acquisition and recording of data on magnetic tape were somewhat limited, but these observations disclosed no instabilities with the electrical system or adverse transients in these tests upon connecting the diesel generator to various loads. As expected, the crankshaft torque signal and the crank fillet strain gages showed a significant 30-Hz component keyed to the pressure rise of each cylinder.

With the test data recorded on magnetic tape, the review plan at the completion of testing was to permit LILCO and its contractors to review and verify calibration and zero settings of the various data channels in their home facilities before conducting an independent review of the test data.

3.2.5 Preliminary Review of Diesel Status Prior to Crankshaft Failure

3.2.5.1 Review of Diesel Generator Test History

Documentation of the test program at TDI prior to delivery of the diesel generators to the Shoreham plant was requested but was not received for review; however, statements by LILCO and TDI at the September 1-2 briefing indicated that a number of manufacturer's operational tests were performed on the engines in addition to the nuclear qualification program performed using DG 101. It was also stated that the test data confirmed "to within 1%" the critical speeds calculated during design. No statements were made concerning whether these tests confirmed the amplification factors of each significant order of vibration.

Reference 15 is a summary of Shoreham's test program for the emergency diesel generators received for review in advance of complete documentation. This summary indicated that the test program was responsive to Regulatory Guides 1.108 and 1.9 and IEEE Std 387, in accordance with LILCO's commitments in the Shoreham FSAR.

The test program was described as being of the "building block type"; it started with checkout and initial operation tests for individual components, and the components were then combined into subsystems and tested again. The checkout and initial operation were stated to consist of 138 test packages in addition to 12 flush procedures, followed by 15 functional test procedures [15].

After the above tests, the diesel generators were operated for the first time as follows [15]:

DG 102 in October 1982
DG 103 in March 1982
DG 101 in April 1982.

Testing of each diesel continued according to procedure, and the final test was performed to demonstrate the capability of the diesel generators to complete successfully a total of 69 consecutive starts. According to Reference 15, "By June 24, 1983, the emergency diesel generator preoperational

test program, including all mechanical, electrical and qualification tests, was completed for all three diesel generators."

In August 1983, all three diesels underwent a cylinder head stud replacement program, and one diesel generator completed the high load retest [15]. LILCO's summary continues, stating that "one remaining demonstration of diesel capability was scheduled prior to fuel load; the integrated emergency core cooling system and emergency diesel generator operational demonstration."

LILCO reported [15] that as of the August 12, 1983 crankshaft fracture, the diesel generators had accumulated 2182 hours of operation as follows:

DG 101 -- 646 hours
DG 102 -- 718 hours
DG 103 -- 818 hours.

In response to a request for information by the NRC regarding the total number of operating hours on each diesel generator and the total number of hours at 3900 kW or greater, LILCO responded [16] as follows:

	Total Operating Hours on Each DG Unit			Total Operating Hours for Each DG Unit at 2-hour Overload Rating (≥ 3850 kW) (These hours included in total operating hours)		
DG Unit	At TDI	At Shoreham	Total	At TDI	At Shoreham	Total
101	128	518	646	3	16	19
102	30	688	718	3	19	22
103	40	778	818	3	20	23

3.2.5.2 Review of Conditions at the Time of Crankshaft Failure

In response to a request for information by the NRC about the test procedures in use at the time of the crankshaft failure, LILCO responded with the following description of the test [17]:

"Cylinder heads on DG 102 were replaced under R/RR R43-1001 with new design stress relieved heads. With all eight cylinders equipped with the new heads, the 102 DG was run for 12 hours to allow hot torquing of the exhaust header bolts and air start valve nuts. Following this run, a retest of the engine was begun under 8.7-R43-042. The specific scope of the retest under this 8.7 Form was to:

1. Verify proper diesel generator start to synchronous speed and rated voltage in less than 10 seconds.

2. Verify proper DG operation for four hours at the continous load rating.

3. Verify proper DG operation for 2 hours at the two hour overload rating.

Refer to the response to NRC Request for Information II.2, pages 10.5 through 10.17, for a copy of the retest procedure 8.7-R43-042, as completed up until the time of the failure of DG 102."

On page 10.1 of Reference 18, LILCO provided the following detailed description of the events just prior to the failure:

"The diesel generator prior to the performance of 8.7-R43/42 was in its normal standby condition. An interim operating instruction was performed to ensure proper breaker positions, proper valve lineup and correct initial conditions. The diesel engine was started from its remote location, the main control room. Proper starting, acceleration to synchronous speed and rated voltage within 10 seconds was verified by the test engineer and the OQA inspector. Plant Operator synchronized the diesel generator to BUS 102 by closing ACB 102-8 and then proceeded to increase the diesel generator load to 3500KW in less than 60 seconds. Once at the 3500KW/300KVar load the operator was instructed to maintain this load for four hours. He was instructed that any deviations, caused by the LILCO grid, away from 3500KW/300KVars should be corrected. Another plant operator was stationed in the engine room with verbal communications established between operators via headsets. During the course of the four hour full load run, a LILCO technician was also stationed in the diesel engine room with the task of recording all pertinent test information every 30 minutes. No abnormal readings were observed by either operator nor was the data written down by the technician found to be out of its normal operating range as specified by the engine manufacturer for this size load.

Since this test was handled similar to a Station Surveillance Procedure no special test equipment was utilized for data recording. All data written down was taken off of normal plant gauges either in the main control room or in the diesel engine room. The two exceptions were the generators bearing temperature and the generator stator temperature, both

of which were read off M&TE calibrated instruments. As stated in the 8.7 form high speed recorders were not used to record data on chart paper as a permanent record. Once, during the full load run the individual cylinder firing pressures were recorded and found balanced within manufacturer specified tolerances.

At the conclusion of the four hours the control room operator slowly increased the 102 generator load up to 3900KW/300 KVars. This load was to be maintained at this level for the remaining duration of the test and the operator was allowed to correct for any load deviations. During the increase in load, the lube oil low level alarm came in. The dipstick was checked and found to be below the shutdown level mark by 7-8". (This level is normal for high load operation of the DG units, and the alarm has been an occasional occurrence on all three engines). Lube oil pressure and turbocharger pressure were normal and the test was allowed to continue. Data readings were taken every 15 minutes. No abnormal noises were heard by the technician nor the local operator. Vibrations did not appear anything out of the ordinary; in fact the diesel engine seemed to be running fairly well.

The overload portion of the test was some one hour and 45 minutes into the two hour run when the diesel generator vibration was felt in the control room. The local operator reported no abnormal vibration. Generator load swings of 2.0 MW were observed in the control room meters, the operator reduced load to 1.0 MW and the oscillations, subceeded. It was at this point that the generator load shot up to 4.0 MW where the operator tripped the output breaker ACB 102-8 and manually depressed the 'stop' pushbutton. It was later observed that the engine overspeed trip had been activated and its alarm had been initiated. Other detailed descriptions of this failure are attached, as well as a copy of the data sheets. Again no traces are available for analysis. Inspection of diesel crankcase internals showed the crankshaft web in the area of No. 7 connection rod was cracked."

On page 10.3 of Reference 18, LILCO provided the following sequence of events when fracture is believed to have occurred:

"(Times are approximate and are intended to illustrate sequence rather than exact time of occurrence)

Background:	EDG at 3.9 MW for 1 hour and 45 minutes, 15 minutes from completion of scheduled 2 hour run. NASO - S. Livingston on headset at EDG 102 panel in Main Control Room, E. O. - M. O'Brien on headset in EDG 102 room.
5:15:00	Noticeable increase in vibration in Main Control Room - W. Uhl, W. Nazzaro, W. Gunther approached Main Control room panel. Slight, but normal, fluctuation in load around 3.9

MW - no other indication of problem. Communication to EDG room for observation of any problem - only response was technician was in area taking readings.

5:15:45 Vibration continued and suddenly load swing of 1.5 to 2.0 MW commenced between 2 MW and 4 MW. Communication from CR to field - 'are you doing anything'. Within 15 seconds, load was reduced by CR operator to 1 MW. Vibration ceased. This load was carried for about 15 seconds.

5:16:15 Load increased without cause to 4.0 MW. Vibration increased again. Again communication between CR and EDG room regarding what was going on. W. Nazzaro, instructed Livingston to decrease load. Load would not come down.

5:16:30 W. Nazzaro instructed Livingston to trip the machine who immediately opened the output breaker. Speed was noted to reach 600 RPM before coasting down to rest.

Elapsed Time - 1 1/2 minutes"

3.2.5.3 Review of Previous Vibration Survey

There was ample evidence of concern over the vibratory amplitudes of the diesels. Review of the partial listing [19] of selected previous problems with the diesel generators also provided evidence of high dynamic forces that had the potential of being associated, on preliminary evaluation, with large amplitude torsional vibration. Should subsequent thorough evaluation prove this to be true, then many of the various component failures would no longer be isolated independent events as previously reported but linked to a common cause.

Until more information is available, the following evidence of repeated failures in components directly connected to the crankshaft remains circumstantial:

Date	Failure Description
3/30/83	Holddown capscrews, rocker arm assembly (EDG-103)
9/17/82	Jacket water pump shaft (EDG-102 and 103)
10/05/81	Piston crown separated from skirt

Date	Failure Description
10/05/81	Failure of attachment stud bolts
10/05/81	Grooving of crankshaft bearing and crank pin discolored
10/05/81	Wrist pin grooved and pitted, wrist pin discolored.

Concern over vibration was sufficient to initiate a vibration testing program in the late spring and early summer of 1983 [7]. The conclusion of this study states:

> "On the basis of comparisons of vibration data taken, the Shoreham diesels have only the expected and normal vibration and are not subjected to any excessive vibration and this normal, expected vibration does not prevent the diesels from reliably performing their functions."

It is noted that the study was based entirely on linear vibration measurements without any measurement of torsional vibration. It is true that rotating machinery can suffer from high torsional vibration with little evidence of linear vibration. However, the crank mechanisms of diesel engines provide coupling between the torsional and linear vibratory systems so that there is usually evidence of linear vibration associated with torsional vibration.

4. TECHNICAL REVIEW AND EVALUATION

4.1 REVIEW OF CRANKSHAFT METALLURGICAL EXAMINATION

4.1.1 Material Specifications and Certifications

All documentation submitted by LILCO to the NRC regarding the crankshaft indicated that the only material specification for the diesel generators was that provided in the diesel generator purchase specifications [20, 21]. Pages 9 [20] and 1-10 [21] of the purchase specifications cite "Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines" [2], published by the Diesel Engine Manufacturer's Association (DEMA), under the heading of applicable documents. No other document defining diesel engine material specifications was noted. However, "Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines" [2] does not cover crankshaft materials other than to limit the cyclic stress level under torsional vibration. Although DEMA's recommended practices are discussed at greater length in Section 4.2.1, it may be stated here that no documentation defining a required minimum quality level of the crankshaft, or other engine component, was found.

References 22, 23, and 24 indicate that American Bureau of Shipping (ABS) Grade 3 steel was specified by the engine manufacturer, TDI, for the two crankshafts purchased from Ellwood City Forge, Ellwood City, PA. Reference 25 indicates that the third crankshaft was purchased from Mitsubishi Steel Manufacturing Company, Ltd. in Japan. However, Reference 25 does not include an indication of the grade of steel specified by TDI, but does show that the material conforms to ASTM A273, Gr. AISI C1042. Although the ABS rules covering steel machinery forgings are currrently written to be "in substantial agreement" with classes of ASTM A668 steel, the ABS rules did reference ASTM A235 in the years of engine manufacture, 1973-1975. Study of ASTM A235 indicated that ASTM A273 could be specified where the forging mill desired to use semi-finished steel for the forgings. Thus, ASTM A273 is a specification for carbon steel blooms, billets, and slabs for forging rather than a specification for carbon steel forgings (A235). The conformance to ASTM A235 after

using ASTM A273 forging material would have depended upon the subsequent forging processes. These processes were not indicated.

The ABS "Rules for Building and Classing Steel Vessels" (ships) [3] specifies steel forging material properties primarily by minimum material properties, by certain limitations in steel processing, and by "substantial agreement" with designated ASTM specifications. The minimum properties of ABS Grade 3 steel, shown in Table 1, were taken from the 1973 and 1980 editions of the ABS's rules. The 1973 edition indicates that ABS Grade 3 steel is to be in substantial agreement with ASTM A235-67 Class E and that the steel forgings are to be annealed, normalized, or normalized and tempered. No restrictions on steel chemistry are noted in the 1973 edition, but the 1980 edition of the rules specifies that the chemical composition is to be reported and the carbon content is not to exceed 0.35% unless specially approved. Although this carbon content stipulation was introduced after 1973 and before 1980, the interpretations of this rule appear to be such that if the steel meets the other chemistry and processing requirements, meets or exceeds the requirements of Table 1, and is in substantial agreement with the respective ASTM specification, the steel will usually meet with the approval of the ABS even though it may contain a greater carbon content. Thus, concerning the actual crankshaft steel, with properties as reported by the forging mills [26, 27] and shown in Table 2, approval to qualify as an ABS grade can only be granted by the American Bureau of Shipping following its submittal to them for review.

As part of the early analysis of the failed crankshaft, FaAA analyzed the steel's composition and tested its mechanical properties [28]. The results of these analyses and tests are shown in Tables 3 and 4 for comparison with Tables 1 and 2. The mechanical properties exceed the minimum requirements of Table 1 by a fair margin. With respect to chemical analysis, FaAA reported that "Except for the carbon and sulfur, which were determined by a combustion gas analysis method, the chemical analyses were obtained by an inductively-coupled plasma technique." FaAA concluded that the steel met the ASTM 235-67 Class E requirements in accordance with the specification of ABS Grade 3 steel. FaAA's chemical analysis indicated a chromium content of 0.3%.

Table 1. American Bureau of Shipping Tensile Property Requirements for Carbon Steel Machinery Forgings

	Size				Longitudinal		Transverse	
Grade	Over (in)	Under (in)	Tensile (psi)	Yield (psi)	Elongation (%)	Reduction in Area (%)	Elongation (%)	Reduction in Area (%)
3	8	12	75,000	37,500	22	35	18	28
3	12	20	75,000	37,000	20	32	18	28

Table 2. Mill Certified Crankshaft Properties

			Chemical Analysis (%)					Mechanical Properties			
Engine	Crankshaft Supplier	Material Spec.	C	Mn	Si	P	S	Tensile	Yield	Elongation	Reduction in Area
DG 101 74010-2604	Mitsubishi Steel	ABS Gr. 3 ASTM A273 AISI 1042	0.42	0.64	0.21	0.014	0.011	87,100 87,600	47,200 47,700	24.2 24.2	44.0 42.0
DG 102 74011-2605	Ellwood City Forge	ABS Gr. 3	0.47	0.83	0.18	0.008	0.010	94,500 97,750	51,500 52,000	24.0 25.0	50.3 51.7
DG 103 74012-2606	Ellwood City Forge	ABS Gr. 3	0.47	0.83	0.18	0.008	0.010	96,000 98,000	53,500 55,000	24.0 24.0	51.4 52.2

Table 3. Chemical Analysis* of Shoreham Crankshaft

Element	1st Piece	2nd Piece	Ellwood City Mill Chemistry	ASTM A235-67 Class E
C	0.47	--	0.47	0.4-0.47
Mn	0.6	0.65	0.83	0.9 max
Si	0.12	0.12	0.18	--
S	0.014	--	0.010	0.05 max
P	0.01	0.01	0.006	0.05 max
Cr	0.30	0.39	--	--
Ni	0.054	0.055	--	--
Mo	0.03	0.03	--	--
V	0.04	0.04	--	--
Cu	0.04	0.04	--	--
Al	0.004	0.004	--	--
Ti	--	0.03	--	--

*All elements are reported in weight percent.

Table 4. Summary of Tensile Tests

Specimen Number	Yield Stress (ksi) Upper	Yield Stress (ksi) Lower	Ultimate Strength (ksi)	Elongation (%)	Reduction in Area (%)
R1	46.6	45.0	89.0	25.4	42.0
R2	45.3	44.9	89.4	30.0	45.1
T1	47.1	45.9	87.6	37.1	49.1
T2	46.9	46.9	88.2	39.0	47.6
L1	47.3	45.9	89.5	25.1	35.3
L2	47.4	44.8	89.1	23.0	30.6

It is noted that neither ABS Grade 3 nor the ASTM A235-67 Class E specification limits the chromium content so long as it is a residual amount. A chromium content of 0.3% for this steel is considered to be residual.

FaAA's conclusion that the crankshaft steel, as analyzed, meets the requirements of ABS Grade 3 specification designated by the engine manufacturer is generally acceptable. However, it should be noted, as discussed above, that only the American Bureau of Shipping can approve a material as conforming to an ABS grade if its chemical content is different than the range specified by ABS.

4.1.2 Metallurgical Examination of Fractured Crankshaft

This section summarizes the metallurgical examination performed by FaAA on the fractured crankshaft from DG 102 and reported by FaAA in Section 4.0 of Reference 29.

Appendix 3-1 to FaAA's report contains the agreement reached by LILCO, Stone and Webster, TDI, and FaAA regarding the extent and procedure for cutting and sectioning the featured portion of the crankshaft for metallurgical analysis. The beginning of this process was described by FaAA [29] as follows:

> "The failed crankshaft had fractured into two pieces at the crank pin journal of cylinder No. 7. Fracture occurred mostly through the web connecting the No. 7 crank pin journal to the adjacent No. 9 bearing journal. The section examined was saw-cut from the crankshaft; cuts were made through the No. 8 and No. 9 main-bearing journals. This two-piece section containing the fracture was shipped to FaAA's laboratory in Palo Alto, California for laboratory examination.
>
> Both pieces of the fractured section were examined visually; then the metallurgical failure analysis was performed on the piece nearest to the No. 9 main bearing. The other piece, with the mating fracture surface, has been preserved for any additional examination that may become appropriate in the future."

The two pieces of the cutout fractured section of the crankshaft are shown in Figures 3-1 and 3-2 of FaAA's report [29]. Figures 3-3, 3-4, 3-5, and 3-6 of that report show the methodology of sectioning the half of the fractured segment used for metallurgical examination.

4.1.2.1 Visual Analysis

FaAA [29] reported the location, orientation, and characterization of the fracture surface as follows:

> "The fracture surface exhibited an obvious, unmistakable fatigue crack pattern. Concentric beach marks showed that the fatigue crack started at the surface of the machined fillet radius where the crank pin journal blends into the web. The orientation of the fracture plane at the origin can be described using a visual analogy of a clock face: (1) the clock position is viewed from the output end of the crankshaft, (2) the clock face is centered at the crankpin axis, and (3) the 12 o'clock position is at the point on the pin journal furthest from the crankshaft rotation axis. The location of the fracture initiation is 0.055 inch in the radial direction from the journal surface and at a 4:30 clock position."

FaAA supplemented its discussion of visual analysis with photographs that clearly show a classic development of a fatigue crack. FaAA's Figure 3-8 shows the initial crack development area in the fillet between the crankpin and the web, wherein the early crack development is characterized by fine beach marks indicative of slow progressive crack growth over a large number of stress cycles.

The FaAA report characterized the crack growth and its orientation as being similar to that associated with pure torsion in a cylindrical member. Because the developing crack plane deviated somewhat from the ideal torsional case, FaAA's discussion correctly noted the modifying influence of the transition in geometry from the cylindrical crankpin journal to the crank web.

FaAA's basic findings of the visual analysis and their observation that the crack's surface propagation is essentially normal to the maximum tension stress resulting from the combined cyclic and steady output torque in the shaft were determined to be correct during this review.

4.1.2.2 Scanning Electron Microscopy

FaAA used scanning electron microscopy (SEM) to investigate and characterize the point of origin. This analysis was reported in considerable detail, and 32 SEM photographs were included in the report [29].

In summary, the analysis confirmed that the fatigue crack began at one of a number of score marks on the crankpin fillet that was somewhat deeper than the adjacent machining marks and other score marks. The analysis also proved that the fatigue crack was not unduly influenced by the score mark or other small material imperfections present. No matter how ideal a material may be, small or even microscopic imperfections at which a crack will begin are always present. This means that, although the fatigue crack did initiate at a particular score mark, it would have initiated in that region a number of cycles later had the score mark not been present.

4.1.2.3 Metallography

After completion of the SEM examination, FaAA reported [29] that the sample containing the fracture origin was diamond saw-cut to expose a cross section through the point of origin. The cut surface was ground down in steps to permit viewing at several levels relative to the point of fracture origin.

The FaAA report included photomicrographs of the various cross sections. One optical photomicrograph showed the depth of the score mark to be 0.002 inch in that plane. FaAA's description and characterization of the metal structure and the score mark follow:

> "It is apparent, from the disturbed microstructure at the score mark, that local plastic deformation occurred when this score or anomaly was made on the machined surface; that is, the score mark was indented into the surface, not gouged out. This indicates that the score mark was made after machining. Local deformation resulting from the creation of the score mark may have left highly localized residual stresses that made this location more prone to be the point of origin of a fatigue crack than the surrounding machined surface.
>
> The microstructure was uniformily fine pearlite and ferrite. Figure 4-52 [9] shows this microstructure. The pattern of pearlite and ferrite reveals that the austenite grain size varied from ASTM 4 to ASTM 8. This duplex austenite grain size is consistent with the fact that the steel was not aluminum killed, and, therefore, it is not inherently fine grained. The microstructure is as to be expected for a steel of this type when it has been given a normalizing annealing treatment at 1600°F.
>
> The steel was clean and relatively free of nometallic inclusions. The inclusions observed were generally fine and were distributed uniformly

throughout. Figure 3-53 [9] shows the largest cluster of inclusions seen on the metallographic cross section at Plane B. No unusual inclusions were found near the fatigue crack origin. It is concluded that the microstructure was normal and proper for a forged steel crankshaft. The failure did not originate at an inclusion in the material."

These findings were reviewed and found to be satisfactory.

4.1.2.4 Macroetch Analysis

FaAA reported [29] that slab sections were macroetched to reveal flow line patterns, segregation, general inclusion distributions, and any forging or ingot defects. Their report, supplemented with photographs of the etched sections, indicated that the sections had smooth forging flow line patterns with no metallurgical anomalies.

FaAA's report that the macroetch results (as shown therein) indicate that the crankshaft forging was metallurgically sound was reviewed and found to be satisfactory.

4.1.2.5 Hardness Measurements

The following summary of hardness measurements made and reported by FaAA [29] was reviewed and found to be acceptable:

"Two conclusions can be drawn from these hardness test results. First, no systematic variation in bulk hardness values was observed. This supports the other metallurgical evidence that properties of the shaft are homogeneous and that any forging-induced or as-cast heterogenities have been effectively removed by subsequent heat treatments. Second, the hardness values are consistent with the mechanical properties measured on samples removed from the failed crankshaft, and the measured hardness values are appropriate for a properly heat-treated steel forging of this type."

4.1.2.6 Residual Stress Measurements

Residual stresses measurements were made and reported [29]. These stresses were reported to be made using a position-sensitive scintillation detector wherein a single exposure technique determined the residual stresses in a surface layer about 0.0005 inch thick. The report also indicated that

the area over which the stresses were averaged was 0.040 by 0.040 inch. Reproducibility was reported to be poor due to errors in achieving sufficient accuracy in reposition to a previously measured area.

Residual stresses were reported to be measured first using a specimen (designated Section H) of the failed crankpin and web. These stresses were reported to be low in value, and were judged to have been influenced both by the relatively small sample size and by the fracture process and subsequent failure events: the engine did continue to operate briefly and to increase in speed before being shut down following crankshaft fracture. The associated battering of the fractured surfaces could have instituted a stress relaxing process.

Residual stress values for a subsequent larger specimen cut from the No. 5 crankpin, where the post-fracture environment was much less severe, were reported to vary from approximately -20,000 to -55,000 psi and were reported [29] to be representative of the actual residual stresses in the machined fillet radius of the original crankshaft.

The residual stresses for the second, larger specimen could also be in error because the possibility of residual hoop stresses in the crankpin and fillet was not considered. In obtaining the second specimen, an axial cut was made through the crankpin that included only a small arc of the crankpin and fillet in the sample. If residual hoop stresses were present in the original crankshaft, they would have been largely relaxed by cutting out the material specimen. If these original hoop stresses, when considered alone, produced tensile stresses on the surface, then their relaxation in conjunction with other compressive residual stresses would have decreased the tensile component and moved the state of surface stress farther into the compressive region. This would have been true for both the hoop and axial stresses, which are coupled through Poisson's ratio. Reference to Table 3-9 of Reference 29 appears to give some evidence of hoop stress relaxation, but not enough to substantiate any conjecture that significant residual hoop stresses are being relaxed. It is noted in Table 3-9 of Reference 29 that the axial stresses for the edges of the specimen (points B & C and H & I) where relaxation would

be greater are higher in compressive stress than the points in the center (points E & F). The pattern appears to be symmetrical as expected in the specimen. If this were true, it would indicate that the state of surface stresses could have been much less into the compressive domain than that shown by Table 3-9 of Reference 29.

In addition to the above discussion of residual fillet surface stress in the original crankshaft, it is believed that the presentation of the residual stresses in a 0.0005-inch-thick material layer at the fillet surface is misleading without a discussion of their ramifications. It is true that a state of compressive residual stress is desirable in machine members subjected to cyclic (fatigue) loading. However, without the knowledge of how these stresses vary with depth, the stresses lose much of their meaning. They could represent a very shallow distribution sometimes induced by machining methods and may mask a subsurface state of stress conducive to earlier fatigue failure.

In summary, the conclusions in Reference 29 concerning residual stresses refer to residual stresses induced by machining. These residual stresses induced by machining appear to be compressive, although possibly not to the magnitude reported and, as such, would not contribute to a state of surface stress conducive to crack formation.

4.1.2.7 Metallurgical Summary

FaAA's conclusions about the metallurgical examination follow [29].

> "Failure of the crankshaft in the DG102 emergency diesel generator occurred by the formation of a fatigue crack. The fatigue crack started in a surface score on the fillet surface where a crank pin journal blends into a web section. The identification of the fatigue character of the fracture is unequivocable. Unmistakable beach marks are clearly visible on the fracture surface.
>
> Microstructure, composition, and mechanical properties of the forged steel crankshaft are proper as required by ABS Grade 3 or the equivalent ASTM A-235-67 Grade E, the pertinent steel specifications. The crankshaft is metallurgically sound.
>
> The location and planar orientation of the fatigue crack indicate that the crack was caused by cyclic torsion stresses superimposed on the constant torsion stress resulting from the engine output torque.

Since the metallurgical characteristics of the steel are good, it must be concluded that the fatigue failure occurred because of excessively high cyclic stress as applied to the crankshaft during testing.

The fatigue crack was not caused by the score where it initiated, nor did the surface imperfections on the machined surface of the fillet near the crack origin influence the formation of a fatigue crack. Had the machined surface finish been better, the fatigue life of the crankshaft probably would have been longer. Such features are to be avoided in fatigue-prone parts. Had the crankshaft not been subjected to excessively high service stresses, the fatigue crack would not have been initiated even at the deeper score mark."

The present review is in agreement with these conclusions.

4.2 REVIEW OF CRANKSHAFT DESIGN

4.2.1 Rules, Standards, and Methodology

Applicable rules and methodology of design vary widely, depending upon the application of the diesel engine. In general, design rules exhibit greater conservatism of design for applications where engine reliability is paramount. Rules governing the design of engines for ship propulsion are usually conservative and reflect both the need for safety and the more limited repair facilities at sea. The following sections discuss two sets of standards made applicable by the purchase specification and by the engine manufacturer's material specification. Crankshaft design methodology is also discussed.

4.2.1.1 Diesel Engine Manufacturer's Association (DEMA)

DEMA's "Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines" [2] was cited under the listing of applicable documents in the purchase specification [20, 21] of the diesel engines for the Shoreham Nuclear Power Station. The 6th Edition is the latest edition, published in 1972 before the purchase of the diesel engines.

These DEMA standards constitute a set of nonmandatory guidelines for the purchase of diesel engines. The scope of the standards are best expressed by the foreword from the 6th Edition:

"This book has been published to serve as a reference for consulting engineers, government agencies, users, suppliers, power plant superintendents, and engine operators. It provides generally accepted standards for nomenclature, installation, application, operation, and maintenance of engines and accessory equipment in various types of stationary engine installations.

It is not the purpose of this book to attempt to set forth basic design criteria for engines because such approach would be impossible within this volume and yet do justice to the many types of engines on the market, notwithstanding the fact that many technical texts are available to the student who may be undertaking the design criteria aspects of engines in general.

The existence of, or the adoption of, a standard by DEMA does not in any respect preclude any member or nonmember from manufacturing or selling products that differ from these standards."

With respect to the crankshafts, materials are neither specified nor recommended. However, Chapter 7 is devoted to vibration, where design objectives and criteria are discussed under the topic of torsional vibrations. Three aspects discussed in the section apply to the Shoreham diesel engines. These are as follows:

- "In the case of the constant speed units, such as generator sets, the objective is to ensure that no harmful torsional vibratory stresses occur within five percent above and below rated speed."
- "For crankshafts, connecting shafts, flange or coupling components, etc., made of conventional materials, torsional vibratory conditions shall generally be considered safe when they induce a superimposed stress of less than 5,000 psi, created by a single order of vibration, or a superimposed stress of less than 7,000 psi, created by the summation of the major orders of vibration which might come into phase periodically."
- "For the case of shaft elements variously know as 'quill shafts,' 'tuning shafts,' or 'torsionally resilient torque shafts,' and other elements which are specifically designed for the application, and manufactured from material of adequate physical properties, with careful attention to design and machining of keyways, fillets, etc., superimposed vibratory stresses at much higher levels may be acceptable. The design of such elements is always correlated in the torsional analysis."

With respect to the second item above, which recommends limits of 5000 and 7000 psi for crankshafts, it is generally conceded that these stress

limits apply to torsional stresses calculated for the crankpin diameter without the application of stress concentration factors. In fact, stress concentration in the fillets, or the oil hole, is compensated by recommending low cyclic stress limits for the crankshaft. However, such practices are not adequate if the stress concentrations are not limited by a parallel design standard covering crankpin oil hole and/or fillet geometry where stress concentrations can be high.

By contrast, the third item from the DEMA recommendations above permits stresses higher than the 5000 and 7000 psi limit for quill shafts and other elements specifically designed for the application and manufactured from material of adequate physical properties with careful attention to stress-concentrating geometries. The contrast is this: the crankshaft limits include an historical perspective of experience for similar crankshaft geometry, whereas this is not generally possible for special equipment; therefore, full engineering analysis and judgment are required if these values are to be exceeded.

4.2.1.2 American Bureau of Shipping (ABS)

The "Rules for Building and Classing Steel Vessels" [3] by ABS were not invoked as standards for the design of the Shoreham diesel engines, other than the engine manufacturer specified ABS Grade 3 steel in the procurement of the crankshafts. However, the ABS rules are representative of the various rules used to classify ships, of which probably the world's best known rules are Rules and Regulations for the Classification of Ships [4] and Guidance Notes on Torsional Vibration [5] by Lloyds Register of Shipping. Lloyds rules, as well as the ABS rules, are published each year to reflect constant updating.

Ship classification associations, such as the American Bureau of Shipping and Lloyds Register of Shipping, represent possibly the oldest machinery review and evaluation associations functioning today. Lloyds Register began operations in 1760 and published its first set of rules in 1834. As ships and ship propulsion systems became more sophisticated, the classification associations served as design review agents to evaluate functional adequacy and safety. Considerable experience in the review and evaluation of diesel

engines was realized from the long-term use of diesel engines for propulsion and for electric power generation in ships. The ship classification rules probably represent the most extensive experience in large diesel engines available.

4.2.2 Torsional Dynamic Response Analysis

4.2.2.1 Summary of Analysis Methods

Reciprocating engines, especially diesel engines, have always constituted a challenge for the dynamicist. The motions (kinematics) of the engine parts are complex; diesels, in particular, are subjected to sharp cylinder pressure rises that serve to further excite vibratory natural frequencies in the complete power system as well as in the engine.

The dynamic response analysis of an engine or complete power system to internal or external excitation begins with some mathematical model of the system's dynamic properties. The approach most used is based upon an appropriate consolidation of inertias (torsion vibration) and spring constants such as that shown in Figure 2. With the mathematical model established, the dynamicist has a choice of action:

- o solve the dynamic equations directly to yield both the transient and steady-state solutions
- o modify the equations and procedure of direct solution to drop the transient response and retain the steady-state vibratory response
- o make an electric-analog model and simulate the mechanical vibration
- o resort to specialized numerical procedures that have been devised over the years prior to the introduction of digital computers (e.g., Holzer's methods).

In theory, the first course of action is the simplest: write equations equating the summation of forces (torques) on each mass (inertia) to the acceleration of that mass (inertia). Direct solution of these equations will provide the transient and steady-state solutions to yield a simulation of the engine system's dynamic response consistent with the accuracy and completeness of the model and equations employed. Prior to the introduction of computers,

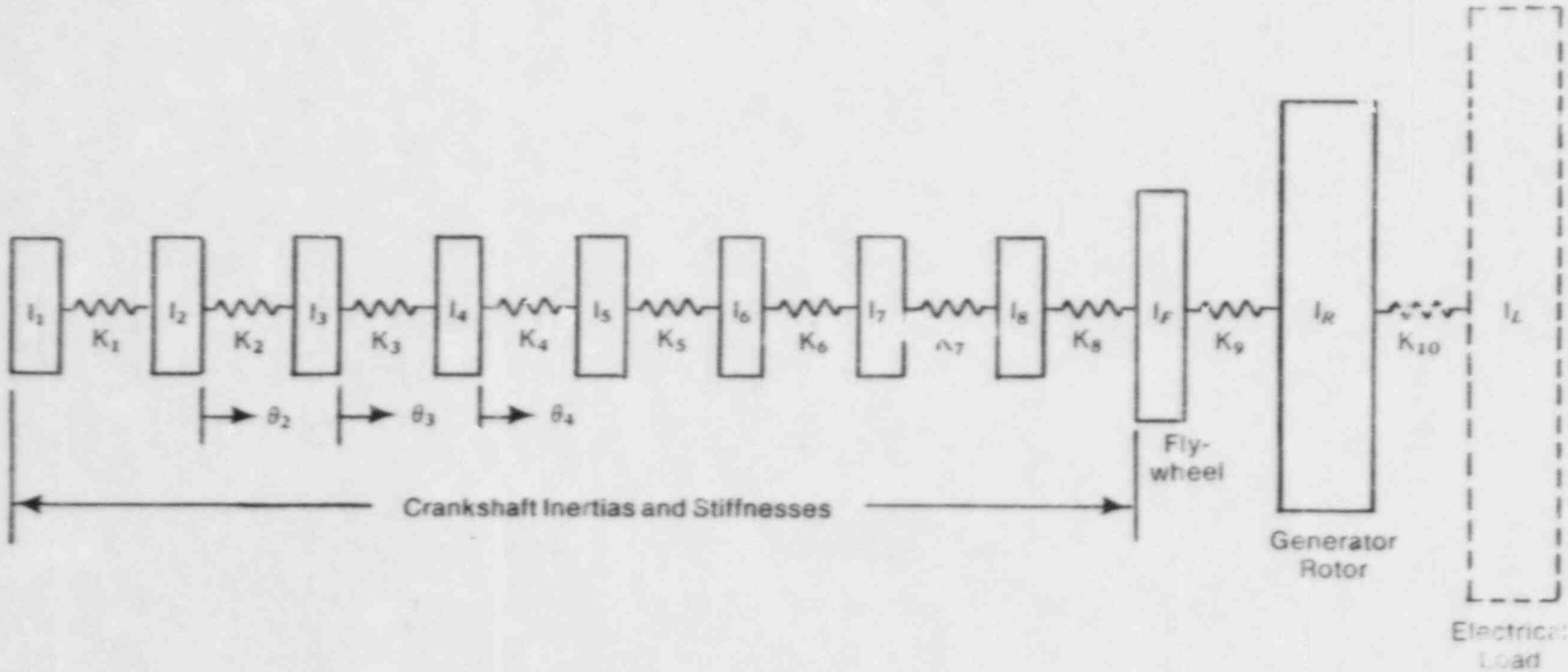

Figure 2. Typical Lumped-Parameter Torsional Mathematical Model

however, there was no feasible method of solving the equations. Even now, with the use of digital computers, a direct solution of the equations tends to consume more computer time than the modified direct solution methods discussed below.

The second course of action is a modification of the direct solution method wherein the solution procedure is modified to introduce certain types of expected steady-state harmonic response and the solution is made to determine the amplitudes and phase relationships of the harmonic responses. This often called the mode superposition method [30], because the normal modes of vibration of the system are determined, each mode is subjected to the excitation, and the resulting responses of the separate modes are superimposed to yield an overall solution to the steady-state problem. In general, this is a little less comprehensive but also less computer-intensive than the direct solution. Damping (energy loss) during vibration is not handled well by this method and approximate values for each vibratory mode must be substituted. However, the method is highly recommended for application to diesel-generator torsional vibration away from an actual resonant point. FaAA used this method for its analytical (computer) model of the Shoreham diesel torsional dynamic response.

Electric-analog methods are powerful and effective but very time consuming to set up and use. The method has largely been supplanted by the digital computer which offers greater versatility.

Since the turn of the century, various numerical tabulation techniques have been developed, of which Holzer's method remains probably the best known today. These earlier methods are important because they form the foundation of most diesel engine torsional dynamics programs in use today for engine design. Further discussion of Holzer's method is presented in the next section.

4.1.2.2 Historical Diesel Industry Analysis Method

Tabular methods, such as Holzer's, have become the key analysis used by the diesel engine industry to determine the amplitude of torsional vibrations

excited by engine operation parameters and load variations [31]. Although tabular techniques are very limited compared to computerized direct solutions, the industry had little else to use throughout most of the years of diesel engine development. During the 1930s and 1940s, the industry refined the tabular methods for the estimation of limited ranges of dynamic response. The trend was so well established that, when introduced, the digital computer was used to carry out, and to further develop, the familiar tabular techniques. Perhaps this occurred because the computer was able to consolidate the basic analysis with certain of the classification rules [5] into one computer program. Although engine manufacturers developed their own computer programs, at least one program (TORVAP by Structural Dynamics Research Corporation), appearing to be based upon Lloyds Register rules [4], was available between 1973 and 1978 on widely used computer time-sharing facilities.

In conjunction with the tabular methods that were developed for dynamic response analysis, Fourier series methods were developed to characterize the cylinder firing excitation and to provide a means by which it could be input meaningfully to the dynamic response analysis. To do this, a curve of cylinder pressure versus crank angle was employed with analysis of the crank positions to develop a diagram of tangential effort as shown in Figure 3. The tangential effort curve connects points of the instantaneous torque, normalized with respect to (divided by) crank radius and piston area. The effort curve is represented by the Fourier series as follows:

$$T = Tm + \Sigma\ (An \sin n\theta + Bn \cos n\theta)$$

or

$$T = Tm + \Sigma\ Tn \sin (n\theta + \phi n)$$

where Tm = constant mean tangential effort

Tn = resultant coefficient of the nth order of torque excitation

$Tn = [An^2 + Bn^2]^{1/2}$

θ = crank angle

n = order number, or the number of cycles of excitation of the component per revolution of the engine.

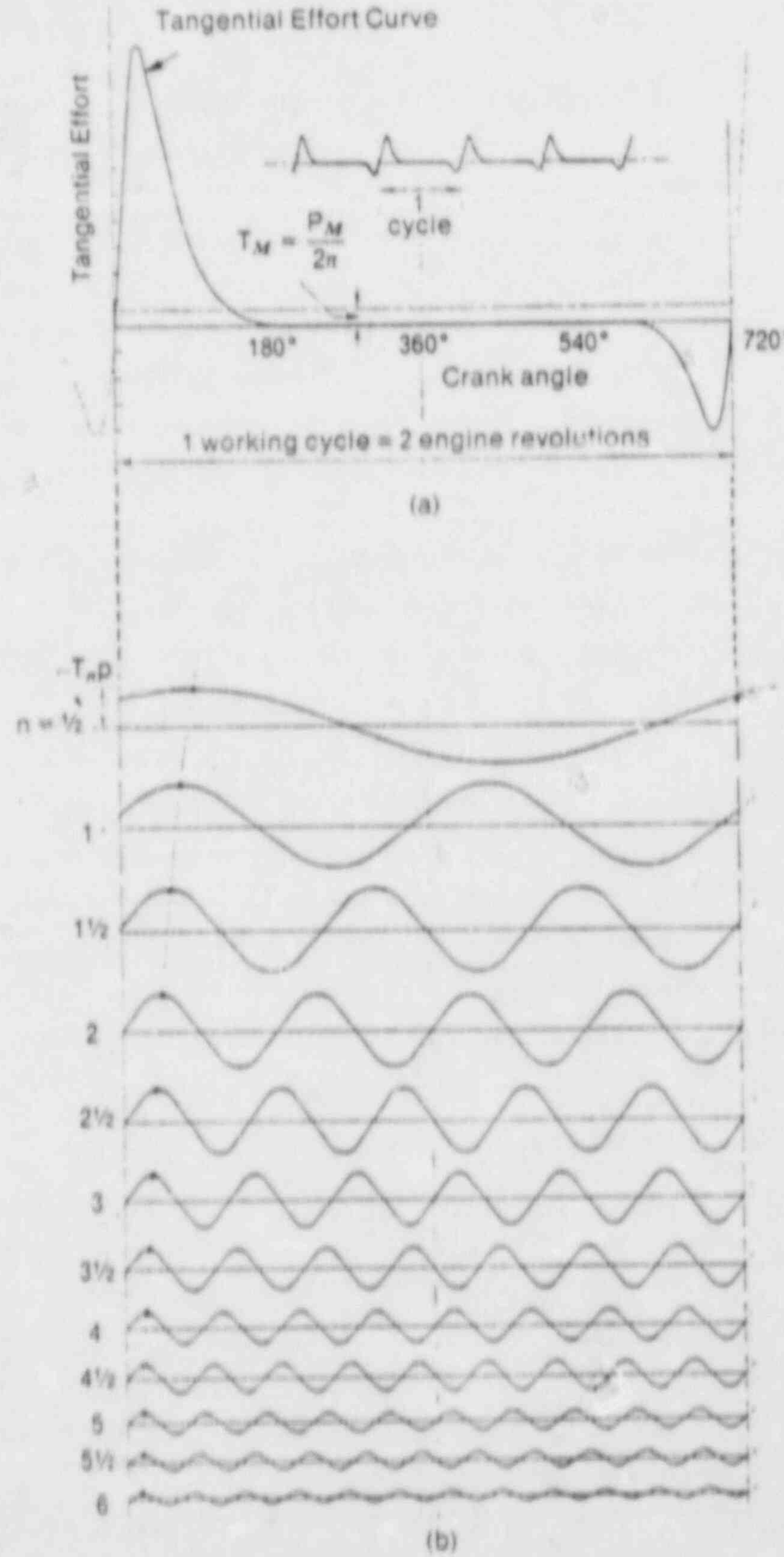

Figure 3. Tangential Effort Diagram and Harmonic Components [from Reference 32]

Table 5. Accepted Values of Tn from Classical Sources

(Tn for mean effective pressure = 225 psi; gas pressure only)

Order	Porter [34] (1943)	Nestorides [33] (1958)	Ker Wilson [32] (1963)	Lloyds Register [4] (1972)
0.5	73.5	76.0	77.0	80.0
1.0	79.8	90.0	79.0	88.0
1.5	69.5	79.0	75.0	83.0
2.0	59.5	66.0	66.0	69.0
2.5	46.0	55.0	55.0	57.5
3.0	42.0	44.0	43.0	47.5
3.5	35.3	33.0	32.0	38.5
4.0	28.7	25.0	25.0	30.5
4.5	22.8	20.0	19.0	23.6
5.0	18.5	15.3	15.0	18.0
5.5	15.1	12.0	11.0	13.8
6.0	12.2	9.5	8.9	10.5
6.5	9.4	7.3	7.3	8.5
7.0	7.3	6.0	6.0	6.8
7.5	4.4	4.9	4.8	5.5
8.0	4.7	4.1	4.0	4.4
8.5	3.5	3.1	3.4	3.7
9.0	2.6	2.7	2.8	3.2
9.5	2.0	--	2.4	2.8
10.0	1.6	--	2.1	2.5
10.5	1.1	--	1.8	2.3
11.0	0.7	--	1.6	2.0
11.5	0.36	--	1.4	1.8
12.0	0.25	--	1.2	1.6

Please refer to References 32 and 33 for full discussions of tangential effort. Accepted values of Tn [4, 32, 33, 34, 35] are shown in Table 5. For order numbers through 9.0, for which all values are shown, the values cover a span of 29 years and many diesel designs, including turbocharging (all are 4-stroke cycle). The values are very consistent.

It should be noted that the values of Tn in Table 5 represent cylinder pressure only. A full solution must include the additions to Tn contributed by the following effects [32]:

- o inertia forces of reciprocating parts
- o dead weight of reciprocating parts
- o inertia of the connecting rod.

In the analysis for the amplitude of dynamic response, the Fourier coefficients, Tn, are the root values of torque excitation magnitude at a given frequency. Because of the amount of tabulation necessary, the analysis may be carried out for only those orders that appear to yield higher response amplitudes. In any event, superposition of events, including the full phase relationship between orders of excitation and vibratory modes, is not generally attempted.

Where superposition is not attempted, the amplitude responses for each order are plotted as shown in Figure 1 (also see TDI critical speed analysis in Appendix A). Total amplitude of response at any particular operating speed is taken as the sum of the individual responses at that speed. These values are converted to shaft torque and to shaft stress by the usual procedures.

4.2.3 Review of TDI Torsional Critical Speed Analysis

The TDI torsional critical speed analysis included in this report as Appendix A was reviewed and found to contain substantial deviations from the magnitude of the dynamic excitation employed, when compared with consistent accepted values, that is, the published values of Tn discussed above. This and other significant points are discussed in the following sections.

4.2.3.1 TDI Mathematical Model

TDI formulated a mathematical model, similar to that shown in Figure 2, for the calculation of natural frequencies and modes of vibration. The calculated natural frequencies were:

First mode	2130 cpm
Second mode	5455 cpm
Third mode	6495 cpm

These frequencies, with respect to the 4th, 4.5th, and 5th orders, yielded critical speeds of 532, 473, and 425 rpm, respectively, as those nearest to the operating speed of 450 rpm.

The TDI mathematical model did not include the load inertia or its coupling to the rest of the mathematical model through the rotor-stator electrical interaction. Although this has been proven to be of minor significance by the torsional testing of the engine, it prevented the recognition of an approximate 3.0-Hz natural frequency when the generator was connected to the electrical power grid as discussed in Section 3.2.3.2.

4.2.3.2 TDI Dynamic Response Analysis

The results of TDI's dynamic response analysis are shown in Appendix A by the chart of response curves for each signficant order. TDI's response analysis summary is included in Appendix A and was performed in accordance with the historic analysis method discussed in Section 4.2.2.2 of this report.

In the course of this review, TDI's use of the Tn values was studied with considerable interest. First, TDI used a different set of Tn values for each of the three mode shapes calculated (Appendix A), but, most significantly, TDI's values are about half the magnitude of the values shown in Table 5 which lists published Tn values. As a consequence, the stress in the crankshaft as calculated by TDI was about half the value calculated from dynamic response analyses employing the published Tn values.

The TDI values of Tn should not, however, be compared directly to those of Table 5 because the values in Table 5 represent only cylinder pressure. Tn

values for diesel engine dynamic response analysis should include all contributions as discussed in Section 4.2.2.2. Because TDI neither described its source for Tn values nor defined the contributors included, it is not possible to discern TDI's technical content. The source and technical justification of TDI's Tn values were requested but have not been provided as of this report. A review of the contributions from sources other than cylinder pressure indicates that the added value is small for the most part, except for the lower orders. For example, piston reciprocating mass provides the largest added value to Tn, -2.845 for the 4th order, 0.0 for the 4.5th order, and +0.408 for the 5th order. These values, added to or subtracted from the historic values, still provide about double the values used by TDI for the critical orders.

TDI has recently made available its range of Tn values used for diesel engine design over the past 10 years. These are shown in Table 6 for comparison with TDI's design values for the Shoreham diesels and are taken directly from TDI's report [35]. TDI, however, still did not disclose the source or content of these values.

4.2.3.3 Comparison of TDI Crankshaft Stresses to the DEMA Rules

The discussion of the DEMA rules [2] in Section 4.2.1.1 of this report shows that the stress criterion for a single order of torsional vibration is 5000 psi, with 7000 psi serving as the criterion for the summation of orders that can coincide periodically.

In Table 7, the "old" stresses follow from TDI's original torsional critical speed analysis (Appendix A), whereas the "new" stresses reflect the use of TDI's latest set of Tn values in that same analysis (11-inch crankpin). These stresses were calculated following TDI's analysis (Appendix A) during this review.

Comparing the "old" stresses to the DEMA rules, it is seen that the largest stress for a single dynamic order is 2582 psi, well under the DEMA single order limit of 5000 psi, and that the summation of the most significant orders remains well under 7000 psi.

Table 6. TDI Harmonic Coefficients
(from Reference 35)

Year Listing From Harmonic	1974-1975 LILCO Group 1	1975 CP&L Group 2	1975-1977 MP&L Group 3	1977----- Stride Group 4
0.5	11.00	90.88	97.00	155.45
1	20.62	89.78	94.34	94.21
1.5	19.00	94.88	100.70	129.21
2	24.06	45.43	42.53	42.61
2.5	20.20	62.38	65.10	71.51
3	19.97	14.84	16.57	16.52
3.5	16.70	38.91	40.61	42.72
4	13.30	29.04	30.25	27.62
4.5	9.85	12.48	12.73	12.72
5	7.30	9.21	9.39	9.38
5.5	5.65	7.01	7.14	7.14
6	4.18	5.55	5.68	5.68
6.5	3.29	4.39	4.49	4.49
7	2.66	3.60	3.69	3.68
7.5	2.23	2.98	3.05	3.04
8	1.87	2.46	2.52	2.52
8.5	1.61	2.20	2.26	2.26
9	1.42	1.92	1.97	1.97
9.5	1.25	1.50	1.53	1.52
10	1.11	1.25	1.27	1.27
10.5	1.00	1.13	1.14	1.14
11	0.91	1.01	1.02	1.01
11.5	0.82	0.88	0.89	0.89
12	0.74	0.78	0.79	0.79

Table 7. TDI Stresses for DEMA Rules (11-inch Crankpin)

Order	Old Tn	Old Dynamic Stress (psi)	New Tn	New Dynamic Stress (psi)	Selected Orders
1.5	19.00	320	129.2	2176	2176
2.5	20.20	425	71.5	1504	
3.5	16.70	281	42.7	718	
4.0	13.30	2582*	27.62	5362*	5362
4.5	9.85	790	12.72	1020	
	Σ	4398	Σ	10,780	7538

*Values used for comparison with single order recommended limits.

For "new" stresses, the 5362-psi stress for the 4.0th order is beyond the single order criterion, and the summation of only two orders exceeds the 7000-psi criterion. Carrying the summation further, Table 7 indicates that for orders 1.5, 2.5, 3.5, 4.0, and 4.5, which can coincide periodically but at a lower frequency, the sum of the stresses is double the DEMA criterion.

A check of well-known literature would have indicated that the Tn values should have been questioned. Also, the use of rules such as Lloyds Register [4] would have brought about an automatic check of the Tn values.

4.2.3.4 Comparison of TDI's Crankshaft Design to the ABS Rules

Selected paragraphs from the 1980 Rules for Building and Classing Steel Vessels [3] by the ABS are provided in Appendix B. The TDI 13 x 11 crankshaft is compared with applicable paragraphs, and commentary about the 1973 rules is provided where differences are apparent.

<u>ABS Paragraph 34.3.4, Torsional Vibration Stresses</u>

This ABS rule requires submittal of calculations including tables of natural frequencies and vector summations for critical speeds of all orders up to 120% of rated speed, and stress estimates for criticals whose severity approaches or exceeds the limits in ABS paragraph 34.57 and ABS Table 34.3. ABS Table 34.3 limits the stress for a single harmonic (order) to the following values:

ABS Grade 2: ± 2134 psi
ABS Grade 3: ± 2490 psi
ABS Grade 4: ± 2679 psi

ABS paragraph 34.57.1 establishes that, for designs differing from previous installations, stresses at single harmonic critical speeds are not to exceed the above limiting stresses. ABS Paragraph 34.57.1 also indicates that total vibratory stress in the interval of 90% to 105% of rated speed is not to exceed 150% of the above stresses. During this review, it was assumed that the method of totaling is not fixed and that stresses due to all significant orders of vibration may be reasonably totaled according to appropriate vector sums if the orders do not coincide periodically.

Note that the ABS limits for cyclic stress are approximately half the DEMA [2] recommended values of 5000 psi and 7000 psi, respectively.

ABS Paragraph 34.17.1, Minimum Diameter of Crankpins and Journals

The formulas constituting the ABS analysis method shown in Appendix B were used with the substition of the following values from the diesel generator design to calculate the minimum crankpin journal diameter.

D = 17.0 inches, diameter of cylinder bore

P* = 1774 psi, estimated maximum cylinder pressure at 3900 kW

L = 16.5 inches, span between main bearings

H = 3900 kW/0.746 kW per hp = 5228 hp

R = 450 rpm

C = 1.00, engine with more than 6 cylinders

F = 2140, Grade 3 forgings (a constant; see Appendix B)

M = $0.131\ PD^2L = 1.108 \times 10^6$ (formula from Appendix B)

T = $63{,}000\ H/R = 0.7319 \times 10^6$ (formula from Appendix B)

d = 11.12 inches, minimum crankpin or journal diameter (see formula, Appendix B)

Note: DG 101, DG 102, and DG 103 had 11.00-in crankpins.

These calculations for minimum crankpin diameter under the ABS rules show that, based on ABS Grade 3 (for which constant F = 2140), the required minimum crankpin diameter is d = 11.12 in. For ABS Grade 3 material, the diesel generator crankshafts with crankpin diameters of 11.00 inches would be underdesigned. However, the actual crankshaft material has physical properties more nearly equal to ABS Grade 4 and, except for one reduction of area measurement reported by FaAA (Table 4), could qualify as an ABS Grade 4 material. Thus, using the material constant, F = 2310, for ABS Grade 4

* Cylinder pressures were not available. However, an informal telephone communication with Dr. Johnston, FaAA, on February 7, 1984 indicated that peak cylinder pressure was 1680 psi at the 3500-kW load. Therefore, a maximum peak pressure of 1774 psi at 3900 kW was computed as the square root of the ratio of the power levels.

material in the above calculation yields a minimum allowable crankpin diameter of d = 10.84 in. This is less than the 11.00-in actual crankpin diameter. In summary, if full credit were taken for the actual crankshaft material properties, the 11.00-in crankpin diameter of the diesel generators just meets the minimum ABS crankpin requirements as shown by calculations performed as a part of this review.

ABS Paragraph 34.17.4, Solid Crankshaft Web Dimensions

In order to provide adequate bending stiffness in the web, ABS requires that the web dimensions satisfy the following inequality:

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

where w = 21.0 inches, width of web

t = 4.5 inches, web thickness

d = 11.00 inches, crankpin diameter.

Thus,

$$(21)(4.5)^2 \geq 0.35(11.0)^2$$
$$425 \geq 42.4$$

and the inequality appears to be satisfied. (The values for w and t were acquired informally by a telephone conversation with Dr. Wells, FaAA, on February 9, 1984, and are assumed to be sufficiently accurate for this calculation.)

Summary of ABS Rules Application

Comparison of the TDI 13 x 11 crankshaft design with the ABS rules indicated that the crankshaft geometrical proportions were within the ABS rules, but the dynamic stresses in the crankpin were not. Thus, the ABS rules are significantly more conservative with respect to harmonically induced stress than the DEMA rules, or about half the DEMA recommended limits. Again, this reflects the conservative design believed to be required for safety at sea, and probably is derived from the culmination of long-term experience in that industry.

4.2.4 Summary of Crankshaft Design Review

The review of this section on crankshaft methodology and design may be summarized by the following statements:

- o The DEMA rules [2] are not design specifications and standards. Supplementary specifications and standards are required.
- o It is advisable to employ the more comprehensive direct or modified direct solution of the mathematical model equations for torsional dynamics. With the present development of computer methods and accessibility of computer systems, the direct solution methods are not more labor intensive than the present computerized tabular methods and do provide more comprehensive design assistance.
- o TDI used Tn values for torsional excitation that are very low compared to values recognized in the industry since at least 1942 [36].
- o The TDI crankshaft (11 x 13) does not meet the DEMA or ABS rules for dynamic stress when the revised TDI values of Tn are employed.

4.3 REVIEW OF CRANKSHAFT DYNAMIC TESTING

Dynamic testing of the crankshaft is regarded in this review as the essential element of the failure investigation because it is only through carefully conducted measurements that the actual engine dynamics and local component stresses are confirmed. Accordingly, great attention was paid to each aspect of the test program.

Dynamic testing of DG 101 using an instrumented crankshaft was performed on September 20 and 28, 1983 at the Shoreham Nuclear Power Station. Reviews of preparations and procedures and an account of test observations were reported previously [1].

Instrumentation for the measurement and recording of vital dynamic data included that are shown in Section 3.2.4.1

Since the completion of testing, the recorded data were reduced and reported [37] by Stone and Webster, and the implications for the crankshaft

failure investigation were reviewed and reported [29] by FaAA. This section is primarily a review and evaluation of the reported test data [37] and the failure investigation conclusions [29] that were reached.

4.3.1 Instrumentation, Signal Conditioning, and Data Recording

Reference 37 provides a description and statement of applicability of transducers employed, including those for strain, torque, torsional shaft displacement, cylinder pressure, generator voltage and current, linear vibration, and the combination of crankshaft position and rotational speed. A table listing their pertinent characteristics and applicable ranges is also shown. The instrumentation was evaluated and its installation observed by the reviewer at the time of dynamic testing.

For the most part, data output from the transducers was good. Earlier problems of strain gages and data transmitters on the rotating crankshaft were largely corrected before completion of testing on September 28, 1983, although the reported data [37] do include noisy, but apparently functioning, strain gage signals, e.g., on the No. 7 crankpin fillet. Also, the transducers for cylinder pressure seemed to function satisfactorily but appeared to provide pressure data lower in value than the actual pressure. The application of instrumentation in these environments is difficult and the experienced experimental test engineer anticipates certain aberrations in these data channels. Indeed, the essence of the test engineer's work is to plan and conduct the test to maximize the good data extracted. Data from the strain gages on the crankshaft were telemetered to nonrotating receivers and were conditioned and recorded along with the other data on a 14-channel, FM mode tape recorder. With proper planning of signal channels prior to a test run, this afforded an opportunity to record simultaneous events on parallel channels. The signal conditioning and recording equipment are described in Reference 37.

The application of transducers, signal conditioning, and data recorders was reviewed and found to be satisfactory.

4.3.2 Calibration Procedures

Measured values are not necessarily more accurate than analytical estimates; experimental measurements are only as accurate as the accuracy of their calibration, and then only if the proper instrument was chosen for the task.

4.3.2.1 Strain Gage and Torque Bridge Calibration

Fillet strain gages and the torque bridge (employing strain gage) were calibrated by the shunt resistance method, wherein a precision resistor of known value is shunted in succession across the available arms of the bridge circuit.

Shunt resistance of the strain gages provides calibration not only of the strain gages, but also of the conditioning circuitry and recording equipment. However, it calibrates the gage only for measurement of surface strain in the metal on which the gage is located. This is sufficient calibration for the crankpin fillet gages which were for the measurement of surface strain.

Calibration of the torque bridge, which used strain gages, required additional procedures because the measured quantity was that of shaft torque and not strain at a point. Consequently, the test engineers employed static torque tests and test operation of the engine at zero electrical output to confirm the calibration of the torque bridge.

The static torque test yielded measured torque plotted against applied mechanical torque as shown in Figures A-10 and A-11 of Reference 37. Considerable hysteresis is noted in these figures due to the friction in the engine and possibly due, in part, to strain gages that are not fully exercised following their installation. Industry experience has shown that the relationship would be much more linear in actual operation, where the bearing surfaces would be operating on developed oil films to greatly reduce hysteresis due to friction in the engine, and the strain gages would become "exercised" for greater linearity.

The zero-output tests of the instrumented engine are discussed in Section A.2.2 of Reference 37, which includes a table of values measured at four

electrical loads. The normalized values of "kW/1000 lb-ft" showed a spread of +4% and -6% about an arithmetic mean value. Using linear regression, the mean ratio of the measured values of "kW/1000 lb-ft" was calculated by Stone and Webster to be 1.21. Although Reference 37 explains this to be the stress concentration in the shaft on which the strain gages are mounted, evaluation during this review indicated that the actual stress concentration is on the order of 1.16 and that the balance of the factor is due to the experimental measurement spread of the "kW/1000 lb-ft" values previously discussed.

Shifts in zero reference of the data recordings were investigated as a part of the data analysis as discussed in Section A.3 of Reference 37. The overall error due to static strain ranged from 1.0 to 4.2%. Thus, the static offset does affect the calculation of principal stresses by a small percentage because these are based upon both the static and instantaneous cyclic stress. It should be noted, however, that the stress range of the cyclic stresses is not affected by this offset.

4.3.2.2 Calibration of the Torsional Vibration Displacement Transducer

The torsional vibration transducer is the unit attached to the gear case end of the crankshaft for the direct measurement of vibrational amplitude. Sections 3.2.2, 4.3, 6.3, and A.4 of Reference 37 describe the application and calibration of this unit, wherein calibration is performed easily by means of fixed limits on displacement built into the unit.

A problem that arose with the use of the transducer was corrected during data reduction. As described in Reference 37, an internal filter selection switch remained set to a 10-Hz cutoff frequency. This attenuated all signal components above 10 Hz. Data reduction procedures were developed to amplify the attenuated signal components in an effort to correct the error. The procedure was reviewed, and the results of the error-correcting efforts shown in Reference 37 were evaluated and found to be satisfactory.

4.3.2.3 Calibration of Accelerometers

Sections 3.2.5 and 4.5 of Reference 37 cover the application and calibration of accelerometers for linear vibration measurement. The accelerometers

were calibrated with the use of a B&K Model 4291 calibrator, which could serve as a transfer standard from the National Bureau of Standards.

This review showed that any use of this transfer standard capability was not stated in Reference 37 and that the calibration source was not known. Although these data were not necessary in forming a conclusion regarding the cause of failure, calibration of the accelerometers and other instrumentation should nevertheless be traceable to the National Bureau of Standards.

4.3.2.4 Calibration of Cylinder Pressure Instrumentation

Sections 3.1.4, 3.2.3, 4.4, and 7.3.5 of Reference 37 describe the measurement of cylinder pressure and its calibration. Time history pressure measurement was attempted by means of precalibrated piezoelectric transducers installed in the compression test cocks of engine cylinders 5 and 7. Calibration of the data signal circuitry between the transducer and the tape recorder was performed using the B&K Model 4291 calibrator mentioned previously.

The cylinder pressure measurements were unacceptably low. Efforts by Stone and Webster and FaAA following these tests concluded that the gas flow path geometry (see Figure A.4, Reference 37) was responsible. Accurate cylinder pressure measurement was not necessary in this test for conclusions regarding the cause of failure.

4.3.3 Review of the Experimental Data

Dynamic tests of engine operation were run at zero-output load (variable speed tests) and at loads of 100 kW, 1695 kW, 1706 kW, 1750 kW, 2250 kW, 2550 kW, and 3500 kW, with constant speed (450 rpm) operation. Data for these tests were reduced by Stone and Webster and are presented as charts in References 29 and 37.

The test data as presented [24, 37] are dominated by presentations of torque and crankpin fillet strain. Torque, as presented in Figure 4-21 of Reference 29, is characterized by a 30-Hz oscillation of varying amplitude superimposed upon a steady-state value. Torque oscillatory amplitudes for

3500-kW operation reach a value of ±175,000 ft-lb (350,000 ft-lb, peak-to-peak torque range) superimposed upon a steady torque of 57,000 ft-lb. Note that this cyclic torque is a little over 3 times (6 times for peak-to-peak range) the steady torque required to produce an electrical output of 3500 kW from the generator. This single amplitude ratio of 3 stands in contrast to the ABS rules [3] where the single amplitude dynamic component is expected to be on the order of the value of the steady-state component (power transmitted). This is explained as follows. Refer to Section 4.2.3.4 of this report and note that the allowable crankpin single-order torsional stress, using the example of ABS Grade 4 steel, is ±2679 psi. For the 100% load rating of the diesel generators (3500-kW output), the engine torque at the flywheel shaft (torque bridge location) is 57,040 ft-lb, which yields a crankpin torsional shear stress of 2619 psi. This is very close to the limiting torsional stress level allowed by the ABS rules. This example was calculated for the 100% load rating of 3500 kW, the maximum load in the torsional dynamic tests performed. For the intermittent 3900-kW diesel generator loads projected for actual service, the steady state, and cyclic stresses would be proportionally higher.

The engine firing rate is 30 Hz. This engine firing rate is sufficiently close to the first mode torsional natural frequency of 35.5 Hz to produce the large dynamic response in the absence of significant damping.

Measured fillet strains on Crank No. 5 varied to a maximum peak-to-peak range of 1800 microstrain (1800 x 10^{-6} inch/inch) as reported for stain gage 5-1 in Figure 4-21 of Reference 29. Table 6-2 of Reference 29 reports the major principal stress component of the measured strains to be 57,300 psi at Crank No. 5, corresponding to a measured total peak positive torque of 230,000 ft-lb (cyclic and steady-state) and negative torque of -153,000 ft-lb.

In the absence of direct access to the data and data reduction instrumentation, observation of the diesel generator tests plus analytic investigations of the data reported in References 29 and 37 performed during this review provide basic arrangement with the range and characteristics of torque and crankpin fillet stress reported by References 29 and 37. Note that these are measured values subjected to the measurement errors discussed previously. However, it does appear that these values are accurate to within ±10%.

In addition to indicating high cyclic torques and stresses in the crankshaft, the test program yielded the following observations, with which this review concurs:

- o The rotor-stator electrical coupling within the generator which acted to couple the electrical load inertia to the engine dynamic model produced varying generator output current at 3.75 Hz when connected to the electric power grid, but did not contribute to the failure of the crankshafts.

- o Operation at 0.8, 0.9, and 1.0 power factors at the 2500-kW load range indicated that operation in this range of power factor did not contribute to the crankshaft failure or dynamics of the system.

- o The 30-Hz major dynamic response of the engine is not compounded by any observed interaction with the electric loads, electric power grid, or plant loads.

- o The sudden initiation of plant loads was observed to cause a smooth-orderly response of the engine and generator and was not seen to cause cyclic fluctuations.

- o Connection of the diesel generator to the electric power grid was observed to be smooth and without significant transients, although it is realized that considerable care was taken at the time to make a proper connection. Connection of generators to the electric power grid without adequate synchronization can be damaging.

4.4 REVIEW OF FaAA DYNAMIC MODEL AND CRANKSHAFT STRESS ANALYSIS

4.4.1 Dynamic Response Model

In the course of the failure investigation, FaAA prepared and used a digital computer dynamic response model. From a discussion,* it was learned that the model is generally of the mode-superposition type discussed in Section 4.2.2.1 of this report. Reference 29 indicates that the model used the same basic lumped-parameter (inertias and spring constants) model as formulated by TDI (Appendix A) with the addition of the rotor-stator equivalent spring constant and the electrical load inertia (see Figure 2 also).

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* Discussion with Dr. P. Johnston, FaAA, during test of DG 103 on January 7, 1984, at the Shoreham Nuclear Power Station.

FaAA's computer model output, as indicated by Figures 5-3 through 5-6 of Reference 29, has a remarkable similarity in character and amplitude to the values measured by the engine test. FaAA did not initially provide, in its report [29], the list of Tn values employed in its mathematical model. When it was suggested that the Tn values would be valuable for comparison to TDI's design values and to those from other published sources, FaAA made them available.* Table 8 includes FaAA's values with accepted values from Lloyds Register and Ker Wilson which were included here from Table 5 to facilitate comparison. Comparison with values employed by TDI was made using the TDI values of Table 6. In these comparisons, it was observed that the FaAA values compared favorably with those of Lloyds Register and Ker Wilson. The FaAA values were more than twice TDI's design values (TDI 1974-1975 list in Table 6) in the critical range of orders 4.0, 4.5, and 5.0, and even greater for other orders. Thus, the Tn values for FaAA's mathematical model for which FaAA reported [29] excellent agreement of computed dynamic response with that experimentally measured further confirms the validity of published Tn values over that employed by TDI for design.

Even if FaAA's excitation had been prepared only to achieve the same dynamic response amplitudes as measured in the engine tests, the model would have provided a highly useful interpolation function in portraying the dynamic action at points not available for measurement.

As discussed in Section 4.2.2.1, computer models following from the direct solution of the dynamic equations are very powerful in describing the full dynamics and interactions of a system. FaAA's computer model confirms this. The first task for the model was the prediction of the available cyclic life of DG 101 remaining throughout the course of diesel generator testing on September 20 and 28, 1983. Here, initial dynamic response data measured at the beginning of each test session were introduced to the computer model for comparison and prediction of the available life cycles remaining.

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* Telephone call from Dr. P. Johnston, FaAA, March 9, 1984.

Table 8. Comparison of FaAA's Tn Values with Those of Lloyds and Ker Wilson

Vibration Order	FaAA*	Lloyds Register**	Ker Wilson***
0.5	74.5	80.0	77.0
1.0	86.0	88.0	79.0
1.5	75.1	83.0	75.0
2.0	75.6	69.0	66.0
2.5	54.0	57.5	55.0
3.0	12.3	47.5	43.0
3.5	37.7	38.5	32.0
4.0	28.7	30.5	25.0
4.5	24.7	23.6	19.0
5.0	20.7	18.0	15.0
5.5	16.9	13.8	11.0
6.0	13.8	10.5	8.9
6.5	11.2	8.5	7.3
7.0	9.4	6.8	6.0

* Calculated independently by FaAA. Includes effects of reciprocating masses.

** From Table 5. Not known what effects, such as reciprocating masses, are included.

*** From Table 5. Values for cylinder pressure only.

4.4.2 Crankshaft Stress Analysis

4.4.2.1 Finite-Element Model

FaAA formulated a finite-element structural mathematical model using three-dimensional, eight-node, isoparametric elements to represent one throw of the crankshaft. With application of torques from the dynamic response analysis, the model had the capability to indicate the highly stressed points in the complex crankshaft geometry. Unless extremely fine element grids are employed, finite-element models generally underestimate the stress concentration at local regions. Accordingly, FaAA used the same element distributions in an axisymmetric model of the same diameter and fillet radius so that the lack of stress concentration definition could be assessed by comparison to well-established values [38]. The ratio of the established value and the finite-element stress concentration factor was used as a multiplier for the final stresses predicted in the fillet region by the crankshaft throw finite-element model. This was reviewed and found acceptable. The alternative method of using many more elements in the fillet would have been much more costly in both modeling and computer run time.

FaAA did not include a description of its method of torsional load application in its report [29]. However, when it was shown that the method of torsional load application employed by FaAA in the finite-element model was needed to complete the review of FaAA's crankshaft analysis, Dr. Wells (FaAA) provided a verbal description of the torsional loading method during a document review at the Shoreham Nuclear Power Station on March 8, 1984. The loading method was said to consist of a unit angular displacement applied to the journal end of the crank-throw finite-element model, plus a lateral displacement constraint applied to the side of the journal to represent the lateral constraint provided by the journal bearing. The axial location of the lateral constraint representing the journal-bearing reaction was said to have been varied to study its effect upon the computed stresses in the crankpin fillet. This effect was said to be relatively small. During the review of the crank-throw finite-element analysis and method of loading, it was noted that the unit angular displacement method of torsional load application along with the lateral displacement constraint to induce the journal-bearing

reaction is a generally accepted method, which was deemed acceptable by this review.

4.4.2.2 Bending Stresses

On pages 6-8 and 6-11 of Reference 29, FaAA discussed an investigation of bending stresses in the finite-element model due to an effective piston load at top-dead-center. When the associated bending stresses were indicated to be on the order of 4500 psi, as compared to approximately 40,000 psi for the torque load, the contribution of the connecting rod load in consideration of the fillet stresses was considered to be negligible, especially when the maximum fillet stresses occured when the crank was 130 degrees or so after top-dead-center.

Bending stresses, however, did appear to play a part in the stressing of the fillet as indicated by Figure B-100 in Reference 29. This bending action however, appeared to be local bending in the web and crankpin as part of the gross torsional loading. Consequently, it became a part of the stress concentrating mechanism that caused the highly stressed region to develop at an angle of approximately 130 degrees from the 12 o'clock position on the crankpin.

4.4.2.3 Crankshaft Stress Analysis Summary

The usefulness of a comprehensive stress model is readily apparent. The stresses predicted by the finite-element model appear to be in good agreement with experimentally measured values, even acknowledging the fact that the experimentally measured values contain an inherent error band of up to about ±10%.

Although the use of finite-element models for theoretical analysis, as well as for extending experimental investigations to regions not measurable, is to be strongly encouraged, the validity of the failure investigation was considered during this review to be most relevant in the experimental measurement of crankshaft fillet stresses in actual engine operation. Analytic techniques, such as the dynamic model and the finite-element crank-throw model, while quite powerful, were looked upon in this review as supplemental and confirming investigations.

4.5 REVIEW OF REPLACEMENT CRANKSHAFT DESIGN

Following failure of the crankshaft of the Shoreham diesel generators, the engine manufacturer, TDI recommended the use of an improved crankshaft design, designated the 13 x 12 crankshaft. Whereas both the failed crankshaft (13 x 11) and the recommended replacements had 13-inch main journal diameters, the replacement crankshaft featured an increase in the crankpin diameter from 11 to 12 inches, as well as an increase in the crankpin fillet radius from one-half to three-quarters of an inch. Analyses of the replacement crankshaft by FaAA [39] and TDI [40] are reviewed in this section of the report.

4.5.1 Review of Analysis by Transamerica Delaval, Inc (TDI)

TDI used the same method of analysis as shown in Appendix A for the analysis of the original 13 x 11 crankshafts, with the exception that they substituted the Tn values shown in Group 4 of Table 6 of this report. Here the Tn value for the 4th order is 27.62 as compared to the previous value of 13.30.

In summary, although the critical 4th order Tn excitation value was doubled, the following considerations produced a reduction in the calculated stress for comparison to the DEMA-recommended values:

- o The larger crankpin permitted a 22% reduction in crankpin nominal torsional stress.
- o The increased natural frequency from 35.5 to 38.7 Hz reduced the dynamic magnifier for a 30-Hz excitation from 3.51 to 2.51.

This yielded a 4th order stress of 2990 psi as calculated by TDI for comparison to the DEMA recommendation of <5000 psi.

4.5.2 Review of FaAA Dynamic Response Analysis and Crankshaft Stress Analysis

4.5.2.1 Response Analysis

FaAA employed its computer dynamic model using mode superposition to analyze the dynamic response with all significant modes considered. Inertia and spring constant elements for the model are shown in Table 3-1 of Reference 39, and the resulting natural frequencies are shown in Table 3-4 of that same

reference. The dynamic response was computed by FaAA for "full load". The use of the term "full load" does not carry full definition since the design rating of the diesel generator is 3500 kW, but it is expected to operate at 3900 kW for short periods. For this review, 3500-kW generator output is inferred to be "full load".

Comparison of TDI and FaAA dynamic stress values to the DEMA recommendations follows:

Method of Analysis	Average Torsional Stress (psi) Due to 4th Order	Average Torsional Stress (psi) Due to Summation of Orders
TDI Analysis	2990	--
FaAA Modal Superposition	3300	5640
DEMA Recommendation	<5000	<7000

Comparison of these stresses to those updated stresses for the 13 x 11 crankshaft, as shown in Table 7 of this report, indicates reductions in stress by a factor of 1.79.

Comparison of these stresses to the ABS rules, similar to that shown in Section 4.2.3.4 of this report, indicates that the ABS rules may or may not be satisfied depending upon the interpretation that would be approved by the ABS following its review.

Assuming that an ABS Grade 4 steel was used for the crankshaft, the ABS allowable stress for a single harmonic is 2680 psi (see Section 4.2.3.4), whereas the calculated stress (TDI) is 2990 psi. Thus, TDI's stress of 2990 psi and FaAA's stress of 3300 psi were both in excess of the ABS allowable stress for a single harmonic using a nominal ABS Grade 4 material.

The actual mechanical properties of the replacement crankshaft material, however, were shown by the quality control documents at the Shoreham plant to be those provided in Table 9. Whereas Appendix B shows an ABS Grade 4 material to have an ultimate tensile of 83,000 psi, the minimum ultimate tensile

Table 9. Properties of Replacement 13 x 12 Crankshafts

Mechanical Properties

Crankshaft Number	Yield Point (psi)	Ultimate Stress (psi)	Elongation (%)	Production Area (%)	Brinnel Hardness	Sample Location
693 (DG 103)	58,310	100,360	25.0	54.1	205	--
	59,470	106,460	24.0	58.9	212	--
694 (DG 102)	57,290	101,820	25.0	50.9	210	--
	58,310	106,460	25.0	48.7	215	--
695 (DG 101)	52,650	100,800	24.0	50.9	205	Top
	48,590	100,800	23.0	49.8	210	Bottom

Chemical Analysis

Crankshaft Number	Heat	C (%)	Si (%)	Mn (%)	P (%)	S (%)	Cr (%)	Al (%)
693 (DG 103)	821-487	0.50	0.05	0.70	0.006	0.010	0.63	0.003
694 (DG 102)	821-487	0.50	0.05	0.70	0.006	0.010	0.63	0.003
695 (DG-101)	811-167	0.46	0.12	0.65	0.010	0.008	0.69	--

strength of the replacement crankshaft materials as shown in Table 9 is 100,360 psi. To take full advantage of this material, an allowable value of 3090 psi for a single harmonic could be presented to ABS for approval in accordance with Note 4 of ABS Table 34.3 in Appendix B.

If full advantage of the material is to be taken, then it is also appropriate to use the full calculated dynamic response due to a single harmonic exciting factor. TDI's stress of 2990 psi was calculated using only the first mode response. Although TDI's analysis does show a small response for the second and third modes of torsional vibration, the second and third modes are seen to add very little to the first mode stress of 2990 psi. Thus, should the interpretation of the ABS rules discussed above be accepted by ABS, TDI's single harmonic stress would be within the ABS limits. However, FaAA's calculated stres of 3300 psi for a single harmonic excitation, based upon a somewhat higher value of Tn and upon greater modal participation, would not.

ABS also requires that the total vibratory stress from all harmonic excitation not exceed 150% of the allowable stress for a single harmonic exciting factor. For a nominal ABS Grade 4 material, this allowable stress is 4020 psi. For the interpretation of the ABS rules to use the full properties, the allowable stress is 4640 psi. TDI's total stresses cannot be compared to these ABS allowables because their analysis methods do not facilitate such summation of stresses. FaAA's calculated torsional stress for the summation of excitation orders is 5640 psi, which is well beyond even the interpreted ABS allowable stresses.

4.5.2.2 Crankshaft Stress Analysis

FaAA used the finite-element method of analysis reviewed in Section 4.4.2 of this report to compute the stress magnitude and distribution for the replacement crankshaft.

Stresses were reported to be reduced from the previous cyclic principal stress range of 60,000 psi to a range of 37,000 psi. This constitutes a reduction by a factor of 1.78 to a cyclic range that is only 56% of the former cyclic range. The reduction was due to the larger crankpin and increased

stiffness with resulting increased natural frequencies as previously discussed, and was supplemented by the increase in the crankpin fillet radius from one-half to three-quarters of an inch. The analysis was considered to be acceptable.

4.5.3 Crankshaft Shotpeening

FaAA reported [39] that shotpeening was introduced to the crankshaft processing in an effort to assure a "consistent, high level of compressive residual stress in the surface and to eliminate machining marks." The report continued by stating that the fillets "will be inspected by a high-resolution, eddy-current method after the break-in run."

Shotpeening has a long history of use in closing microscopic surface cracks and establishing a surface layer of the material in compressive stress. Although the basic idea is good, it was noted during the review that while various levels of shotpeening are available, no description of the process was provided.

Accordingly, the NRC arranged for a document review at the Shoreham Nuclear Power Station on March 8, 1984, during which quality control documents pertaining to crankshaft shotpeening were reviewed, and an informal discussion was held with Dr. Wells of FaAA. It was learned from Dr. Wells that two of the three replacement crankshafts, Nos. 693 (DG 103) and 694 (DG 102), arrived from TDI with the crankpin fillets already shotpeened.

The crankshafts were inspected and the results of the inspection are described by Stone and Webster Engineering Corporation's Coordination Report No. F-46109-G [41] as follows:

> "Problem Description: Delaval has identified 'holidays' or lack of peen coverage in the fillet areas of new diesel crankshafts purchased in accordance with E&DCR F-46109-C. These 'holidays' have been dispositioned as functionally acceptable by TDI, however, recent analysis performed by Failure Analysis Associates indicates that 100% peening coverage is beneficial."

In conjunction with the review of documents on March 8, 1984, photographs of the original shotpeening supplied by TDI were reviewed. Although the

photographs did not provide the desired detail, the photographs gave an impression of surface texture more like grit blasting than shotpeening, i.e., the surface appeared to have been gauged by sharp particles instead of dimpled by round, smooth particles. Although the photographs provided only a limited view of the fillet surfaces, this evaluation of the initial shotpeening concurs with the results of the inspection [41] by Stone and Webster Engineering Corporation.

Stone and Webster's Coordination Report No. F-46109-G [41] provided a recommended solution as follows:

> "Problem Solution: Since the crankshafts are delivered to the site, Metal Improvement Company, a local firm with expertise in shotpeening will perform the rework. The fillet areas shall be repeened in accordance with the requirements of MIL-S-13165B to assure 100% coverage of the fillet areas. Peening shall be performed by Metal Improvement Company on site and the crankshafts inspected by OQA for 100% peening at the fillet areas."

Accordingly, LILCO Repair/Rework Request R/RR R43-1632 specified shotpeening to include the following parameters:

- o Shot size; MI-550
- o Intensity, 0.008-0.010, Almen "C" test strips
- o MIL-S-13165B, Amendment 2.

Quality control documents were reviewed and indicated that the Almen test strips for the repeening provided readings within the specified intensity of 0.008 to 0.010 inch (arc height) with the exception of one test strip which was measured at 0.011 inch.

Photographs of the repeened surface were reviewed and show an improvement in surface texture, indicating an improvement in the quality of shotpeening of the crankpin fillets.

Crankshaft No. 695 for DG 101 was received at the Shoreham plant direct from the supplier, Krupp-Stahl in Germany, without shotpeening. Crankshaft No. 695 was shotpeened at the Shoreham plant to the same specifications as those described for crankshaft No. 693 and No. 694 above. Records reviewed at the Shoreham plant showed that the Almen test strips for crankshaft No. 695

shotpeening indicated that the intensities remained within the specified range of 0.008 to 0.010 inch arc height.

Shotpeening of this intensity is shown by Figure 4 to induce a compressive stress to a depth of from 0.027 to 0.034 inch, with the induced stress distributed as shown in Figure 5. Figures 4 and 5 are taken from Reference 42.

The purpose of shotpeening is to induce a compressive stress in the material at the surface of the crankpin fillets. Since the smooth surface is being disturbed by the particle impacts, it is necessary, once shotpeening is begun, to assure that the shotpeening coverage is uniform and of an intensity, with the right size of smooth shot, to achieve a suitable depth of material in compressive stress. Otherwise, improper shotpeening could serve as a source of added stress concentrations to make the crankshaft more susceptible to fatigue.

The actual peened surface were not available for inspection in the course of this review; therefore, this evaluation was made using the specified parameters, recorded Almen test strip measurements, and photographs of the peened surface. The shotpeening performed at the Shoreham plant is acceptable for the new crankshaft (No. 695) not subjected to shotpeening in advance and will serve to increase the fatigue life of the crankpin fillets. Inspection of crankshaft Nos. 693 and 694 revealed inadequate initial shotpeening; for these crankshafts, the rework shotpeening discussed above would be sufficient to counter the undesirable effects of the previous shotpeening, provided that the shotpeened surfaces that were photographed and made available for this review were representative of all crankpin fillet shotpeening. With this provision, the rework shotpeening is acceptable.

As an alternative to shotpeening, a surface layer under compressive stress can be induced into crankpin fillets by rolling techniques. This is accomplished by pressing a rolling element against the fillet surface with sufficient force to produce stresses in the fillet surface material that are just beyond the yield point. With the proper design of rolling element, the distribution of induced compressive stresses can be controlled to an ideal

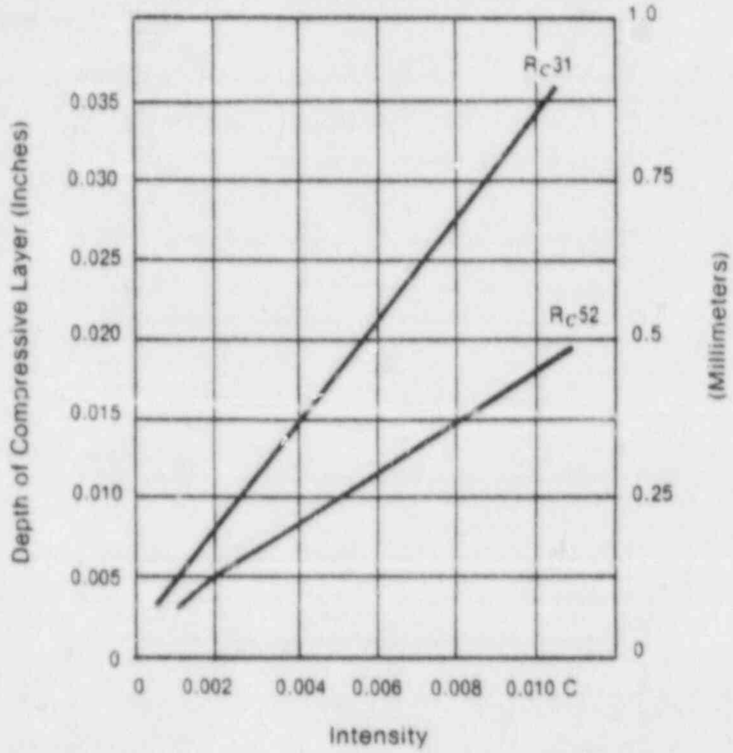


Figure 4. Depth and Compressive Stress vs. Almen Intensity for Steel [from Reference 42]

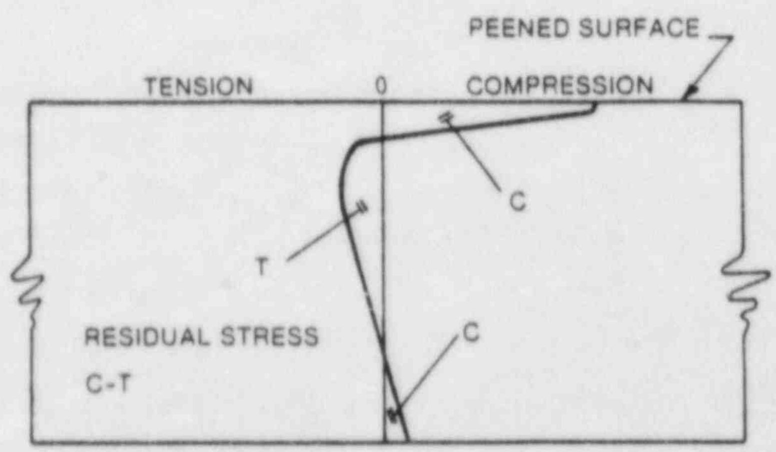


Figure 5. Distribution of Stress in Shotpeened Beam with No External Load [from Reference 42]

profile of magnitude and depth in addition to providing a smooth fillet face for optimum fatigue resistance. Fillet rolling provides many advantages; however, there are fillet geometeries for which it is difficult to design a roller, e.g., recessed fillets similar to those of the TDI crankshafts. In addition, the technique requires the proper machinery to hold, load, and rotate the crankshaft and roller. Where this technique is possible, benefits follow. Lacking the means, shotpeening is recommended.

4.5.4 Summary of Replacement Crankshaft Design

The increase in crankpin diameter from 11.00 to 12.00 inches provided a significant crankpin stress reduction by reducing the direct torsional stress in the crankpin due to larger cross section and by stiffening the shaft to produce a higher natural frequency and thus reduce the dynamic multiplication factor.

Stresses calculated by TDI and FaAA were within the DEMA [2] recommendations for a single harmonic excitation. FaAA's summation of stresses for all excitation orders was also within DEMA's recommended values. TDI's analysis did not permit comparison of total stresses with those recommended by DEMA.

TDI's crankpin stress for single harmonic excitation does not satisfy the ABS limiting values [3] for ABS Grade 4 steel, except through an interpretation of the rules in which full advantage of the crankshaft material properties is taken. Such interpretation would require study and approval by ABS. TDI's analysis did not permit the comparison of total stress due to summation of orders with the ABS allowable values. Crankpin torsional stresses calculated by FaAA for both single harmonic excitation and summation of orders were in excess of ABS allowable values, including the higher allowable values determined by an interpretation of the ABS rules that used the full material properties of the crankshaft material.

Crankpin fillet shotpeening of the replacement crankshafts was evaluated through the review of documentation and photographs at the Shoreham plant.

Crankshafts No. 693 (DG 103) and No. 694 (DG 102) were found to have been previously shotpeened by TDI. When inspection at the Shoreham plant indicated that the initial shotpeening was unsatisfactory, the crankpin fillets were repeened at the Shoreham plant. Crankshaft No. 695 (DG 101), received direct from the supplier in Germany, was not initially shotpeened by TDI and was shotpeened only at the Shoreham plant. The crankshafts could not be inspected directly, and the shotpeening was evaluated only through the review of documentation and inspection of photographs of local regions. The shotpeening and rework shotpeening performed at the Shoreham plant were found to be acceptable insofar as the photographs inspected are representative of all shotpeened surfaces.

It must be noted that all of the TDI and FaAA stresses reviewed herein pertain to the 3500-kW electrical output loading (100% design load) and not to the short-term 3900-kW load required by the Shoreham plant.

5. CONCLUSIONS

Based on the findings of the failure investigation reviewed herein, it is concluded that:

- The crankshaft of diesel generator 102 failed in high cycle fatigue.
- Sufficient cause for the high cycle fatigue failure was crankshaft design based upon exceptionally low values of cyclic torque excitation (Tn) coupled with a natural frequency fairly close to the dominant excitation frequency.
- The specified design standards were not definitive and contributed to the failure by not providing design review material by which the design would have been evaluated and found to be in question prior to the diesel generator's application as safe shutdown equipment.

With respect to the replacement crankshaft design, it is concluded that:

- The combined static and dynamic effects of a 1.00-inch increase in crankpin diameter from 11.00 to 12.00 inches serve to reduce the crankshaft stresses calculated by TDI and FaAA to within the DEMA recommended values for single order excitation and for summation of order excitation.
- Although stresses from TDI's analysis for the replacement crankshaft do not satisfy the ABS rules for a single harmonic using a nominal Grade 4 material, they would just meet an interpretation of the ABS rules for a single harmonic wherein the actual properties of the crankshaft material are used. However, such interpretation of the ABS rules is subject to review and approval by ABS.
- TDI did not present an analysis by which their summation of stresses from all orders can be compared to the ABS limiting value for that condition.
- FaAA's crankshaft analysis predicts higher dynamic stresses due to (1) the use of slightly larger amplitudes of excitation (Tn values) than those used by TDI and (2) the superposition of modes resulting from the direct solution of the equations of crankshaft dynamics. Vibratory stresses computed by FaAA do not satisfy the ABS requirements for a single vibratory order or for the summation of orders, even considering an interpretation of the ABS rules to fully use the mechanical properties of the crankshaft steel.

- o All analysis of stresses performed by TDI and FaAA pertained to the 3500-kW full load condition and not to the 3900-kW short-term overload required by the Shoreham plant.

- o Crankpin shotpeening of one crankshaft and rework shotpeening of two crankshafts performed at the Shoreham plant were found to be acceptable only insofar as the evaluation from documents and photographs of localized shotpeened areas is representative of all crankpin fillet areas.

From the broad evaluations performed in the course of this review, it is summarily concluded that a set of standards more definitive than DEMA's "Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines" is required for diesel engines essential for safe shutdown of the Shoreham plant; that "Rules for Building and Classing Steel Vessels" by the American Bureau of Shipping is representative of definitive standards for safety at sea; and that, with the possible exception of TDI's stress for a single harmonic, the stresses evaluated in this review do not meet the requirements of the ABS standard.

6. RECOMMENDATIONS

The following recommendations are offered:

- The application of a torsional vibration damper on the Shoreham diesel generators to reduce the present high amplitude of torsional vibration and the associated high amplitudes of cyclic crankshaft stresss should be investigated. The higher torsiona' amplitude of the crankshaft is the face, or gear case, end which is available for damper attachment.

- Specifications and standards employed by diesel engine manufacturing groups and user groups in the United States and Europe should be evaluated ~~in order to~~ for the purpose of compiling an appropriate set of standards and specifications for the procurement of diesel generators for nuclear power stations so that these standards and specifications can also serve as acceptance criteria for design and performance review. Although this recommendation is made for the review of the crankshafts, the recommendation is also applicable to other engine components.

7. REFERENCES

1. Franklin Research Center, Evaluation of Diesel Generator Failure at Shoreham Unit 1, Interim Report on Phase 1, Failure Cause Evaluation FRC Project 5506, Task 20426, November 18, 1983

2. Diesel Engine Manufacturers Association, Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines, Sixth Edition, New York, New York, 1972

3. American Bureau of Shipping, Rules for Building and Classing Steel Vessels, New York, New York, 1973 and 1980 Editions

4. Lloyds Register of Shipping, Rules and Regulations for the Classification of Ships; Part 5, Main and Auxiliary Machinery, Chapter 2, Oil Engines

5. Lloyds Register of Shipping, Guidance Notes on Torsional Vibration Characteristics of Main and Auxiliary Oil Engines, 1972

6. Master Plan, Crankshaft Failure Analysis/Recovery, Emergency Diesel Generator 102
Shoreham Nuclear Power Station
August 23, 1983

7. P. J. Holden, Project Engineer
Diesel-Generator Operational Review Program, Prepared for LILCO, Submitted to E. J. Youngling, LILCO
July 14, 1983

8. S. Ahmed, FRC
Informal Technical Communication to J. T. Beard, NRC
Subject: Regarding LILCO Master Plan for Recovery from EDG 102 Crankshaft Failure
September 9, 1983

9. LILCO Procedure for Crankshaft Testing, Emergency Diesel Generator No. 101
September 2, 1983

10. Revision, LILCO Procedure for Crankshaft Testing, Emergency Diesel Generator No. 101
September 9, 1983

11. Revision, LILCO Procedure for Crankshaft Testing, Emergency Diesel Generator No. 101
September 15, 1983

12. Revision, LILCO Procedure for Crankshaft Testing, Emergency Diesel Generator No. 101
September 23, 1983

13. R. C. Herrick, Principal Engineer, FRC
Memo to C. Petrone, NRC Resident Inspector, Shoreham Nuclear Plant
Subject: On-site for Torsion Test Monitoring
September 24, 1983

14. Delaval Engine and Compression Division
Torsional and Lateral Critical Speed Analysis, Prepared for Long Island Lighting Co.
August 1974; Revised March 18, 1975

15. LILCO Preliminary Response to NRC Request for Information I.3
Transmitted September 20, 1983

16. LILCO Response to NRC Request for Information: 5

17. LILCO Response to NRC Request for Information II.1
Transmitted September 20, 1983

18. LILCO Response to NRC Request for Information II.2
Transmitted September 20, 1983

19. J. T. Beard, NRC
Memo to Gary M. Holahan, Chief, Operating Reactors Assessment Branch, NRC
Subject: Recent Failure Experience of Transamerica Delaval, Inc., Emergency Diesel Generators
August 22, 1983

20. Stone and Webster Engineering Corporation, Specification for Diesel Generator Sets, Boston, MA, October 3, 1973, page 9

21. Stone and Webster Engineering Corporation, Specification for Diesel Generator Sets, Revision 2, Boston, MA, January 26, 1983, page 1-10

22. Long Island Lighting Company, Conference/Trip Report--Trip to Ellwood City Forge Co., Ellwood City, PA, August 31 - September 1, 1983

23. Stone and Webster Engineering Corporation, Crankshaft Quality Document File, Control No. 140, Engine No. 2605

24. Stone and Webster Engineering Corporation, Crankshaft Quality Document File, Control No. 141, Engine No. 2606

25. Stone and Webster Engineering Corporation, Crankshaft Quality Document File, Control No. 258, Engine No. 2604

26. Mitsubishi Steel Manufacturing Company, Ltd., Japan, Results of Testing and Inspection Crankshaft for Delaval Turbine, Inc., June 29, 1972

27. Ellwood City Forge Company, Crankshaft Test Report for Delaval Turbine/Enterprise Division, Elwood City Forge Shop Order No. 0-957, Ellwood City, PA, June 10, 1970

28. Failure Analysis Associates, Preliminary Metallurgical Evaluation of Failed Crankshaft from DG102, Palo Alto, CA, October 1983

29. Failure Analysis Associates, Emergency Diesel Generator Crankshaft Failure Investigation, Shoreham Nuclear Power Station, Palo Alto, CA, October 31, 1983

30. S. Timoshenko, D. H. Young, and W. H. Weaver, Jr., Vibration Problems in Engineering, 4th Edition, John Wiley and Sons, New York, 1974

31. Ker Wilson, Practical Solution of Torsional Vibration Problems, Volume 1, Frequency Calculations, 3rd Edition, Revised, John Wiley and Sons, Inc., New York, 1956

32. Ker Wilson, Practical Solution of Torsional Vibration Problems, Volume 2, Amplitude Calculations, 3rd Edition, John Wiley and Sons, Inc., New York, 1963

33. E. J. Nestorides, Handbook of Torsional Vibration, Cambridge University Press, 1958

34. P. F. Porter, Harmonic Coefficients of Engine Torque Curves, Journal of Applied Mechanics, March 1943

35. Transamerica Delaval, Inc., Letter from C. S. Matthews, Vice President (TDI) to T. M. Novak, Assistant Director of Licensing, NRC, Transmitting TDI Response to Users Group Questions, December 16, 1983

36. R. S. Stansfield, The Measurement of Torsional Vibrations, Proceedings of the Institution of Mechanical Engineers, London, England, Volume 147, No. 5 (1942)

37. E. Bercel and J. R. Hall, Field Test of Emergency Diesel Generator 101, Stone and Webster Engineering Corporation Report B1-1160-041-1, October 1983

38. R. E. Peterson, Stress Concentration Factors, John Wiley and Sons, New York, 1974

39. Failure Analysis Associates, Analysis of Replacement Crankshaft for Emergency Diesel Generators, Shoreham Nuclear Power Station, Report FaAA-83-10-2, PA07396, Palo Alto, California, October 31, 1983

40. Transamerica Delaval, Inc., Proposed Torsional and Lateral Critical Speed Analysis for Engine Nos. 74010/12, Oakland, California, August 22, 1983

41. Stone and Webster Engineering Corporation, Coordinating Report No. F-46109-G, dated September 16, 1983

42. Metal Improvement Company, Inc., Shotpeening Applications, 6th Edition, 1980

APPENDIX A

TORSIONAL CRITICAL SPEED ANALYSIS BY TRANSAMERICA DELAVAL, INC.

TORSIONAL AND LATERAL CRITICAL SPEED ANALYSIS

ENGINE NUMBERS 74010/12

DELAVAL-ENTERPRISE ENGINE MODEL DSR-48

3500 KW/4889 BHP AT 450 RPM

FOR

STONE & WEBSTER ENGINEERING CORP.

LONG ISLAND LIGHTING COMPANY

DELAVAL ENGINE & COMPRESSOR DIVISION
550 - 85th AVENUE
OAKLAND, CALIFORNIA 94621

John J Barichfe

Revised 5-1-1975

By: ROLAND WANG
August 1, [illegible]
Revised [illegible]-18-1975. [illegible] Same

7-18-74

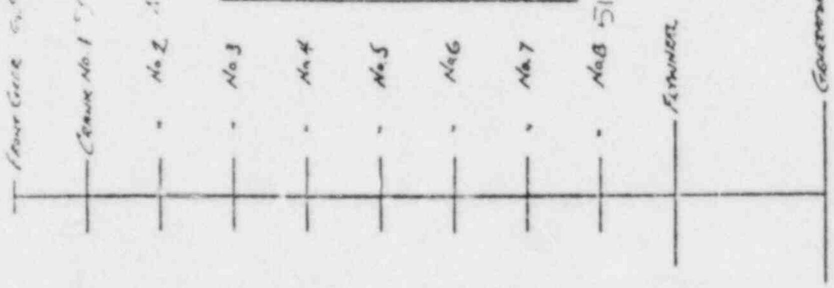
LONG ISLAND LIGHTING COMPANY

DELAVAL ENTERPRISE DSR 4-8

3500KW (4889 BHP) @ 450 RPM

225.6 BMEP

ENGINE NUMBER 74010/12



MASS ELASTIC DATA.

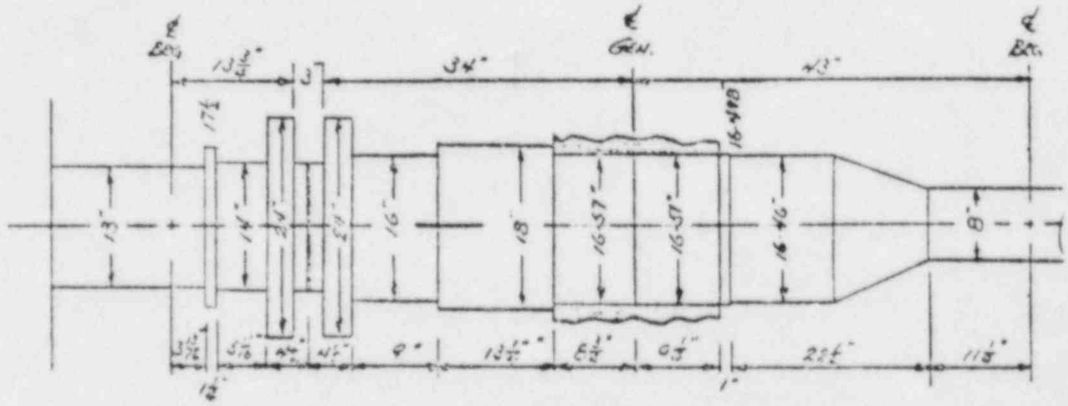
CRANKSHAFT GEAR	26.52	
WATER PUMP DRIVE	63.10	
CAM & IDLER	119.70	
SHAFT	9.63	
	218.95 LB.FT²	I = 6.805 LB. FT SEC²
CRANK NO. 1		
CRANK	1592.52	
SHAFT	41.81	
	1634.33 LB. FT²	I = 50.796 "
CRANK NO 2 - 7		
JOURNAL	44.07	
PIN	168.17	
2 WEBS . NO CTWT	737.40	
RECIP WT.	309.99	
ROTAT. WT.	332.89	
	1592.52 LB. FT²	I = 49.497 "
CRANK NO 8		
CRANK	1592.52	
SHAFT	71.63	
	1664.15 LB. FT²	I = 51.723 "
CRANKSHAFT FLANGE	255.08	
FLYWHEEL 73 × 6½	34764.	
GENERATOR SHAFT FLANGE	374.00	
	35393.08 LB. FT²	I = 1100.052 "
GENERATOR ROTOR	85275 LB. FT²	I = 2650.432 "

TOTAL WK² : 133740 LB. FT²

REVISED 3-18-?
5-1-?

EQUIVALENT LENGTH DG-1

FRONT GEAR TO CYLINDER NO. 1	.001769"	K = 54.572 × 10^6 FT.LB./RAD
BETWEEN CYLINDERS	.001355"	K = 71.246 × 10^6 FT.LB/RAD
CYLINDER NO. 8 TO FLYWHEEL	.001362"	K = 70.880 × 10^6 FT.LB/RAD.
FLYWHEEL TO GENERATOR		



$$L_e = \frac{3 - .0315 \times 16.25}{20^4} + \frac{.055 \times 16 + 9 + .03 \times 16}{16^4} + \frac{-.02 \times 16 + 13.85 - .021 \times 16.572}{18^4} + \frac{.021 \times 16.572 + 16.?}{16.572^4}$$

$$= .0000132 + .0001580 + .0001163 + .0000593 = .0003485"$$ $K = 276.773 \times 10^6$ FT.LB/R

SHAFTING DIAMETERS

FRONT GEAR TO CYLINDER NO. 1	8 in
BETWEEN CYLINDERS	11 in.
CYLINDER NO. 8 TO FLYWHEEL	11 in
FLYWHEEL TO GENERATOR	16 in.

FLYWHEEL WEIGHT = 6935 LB.
GENERATOR ROTOR WEIGHT = 17150 LB.

TORSIONAL NATURAL FREQUENCIES

FIRST MODE	2130 VPM
SECOND MODE	5455 VPM
THIRD MODE	6495 VPM

GENERATOR SHAFT LATERAL NATURAL FREQUENCY. 3191 VPM.

$$N = \frac{36200}{460}\sqrt{\frac{112?0 \times 60}{130740}} = 175 \text{ VPM} \qquad \text{RATIO} = \frac{225}{175} = 1.256$$

mode 1
omega squared in (radians/second)**2 = 0.04974543
natural frequency in v.p.m. = 2129.84

no.	inertia	theta	iom2t	sigma m	shaft k	dtheta
1	6.8	1.00000	0.339	0.339	54.6	0.00620
2	50.8	0.99380	2.511	2.850	71.2	0.04000
3	49.5	0.95380	2.348	5.198	71.2	0.07296
4	49.5	0.88084	2.169	7.367	71.2	0.10340
5	49.5	0.77743	1.914	9.281	71.2	0.13027
6	49.5	0.64716	1.593	10.875	71.2	0.15264
7	49.5	0.49452	1.218	12.092	71.2	0.16973
8	49.5	0.32480	0.800	12.892	71.2	0.18095
9	51.7	0.14384	0.370	13.262	70.9	0.18711
10	1100.1	-0.04327	-2.368	10.895	276.8	0.03936
11	2650.4	-0.08263	-10.894	0.000		

mode 1
omega squared 0.04974543
natural frequency 2129.84
sigma i*theta**2 2431.8705
sigma i*theta**2 2755.3412
T INT 10628.20
T EXT 2837.24
stressed diameter of external shaft 16.00
equilibrium amplitude 0.00099526
f in 7487.33
f int 10.59
f ext 2.83
f e 15202095.62
f a 0.
f cr 0.
f cs 0.
f p 0. [illegible] fourier

order	rpm	tn	vec	tstint	tstext	phi	tmaxi	tmaxe
0.5	4259	11.03	0.719	83.7	22.4	0.223	2372.	633.
1.0	2129	20.62	0.149	32.6	8.7	0.087	924.	247.
1.5	1419	19.00	1.432	288.2	76.9	0.768	8162.	2179.
2.0	1064	24.06	0.384	97.8	26.1	0.261	2769.	739.
2.5	851	20.20	1.432	306.4	81.8	0.816	8678.	2317.
3.0	709	19.97	0.149	31.6	8.4	0.084	894.	239.
3.5	608	16.70	0.719	127.1	33.9	0.339	3601.	961.
4.0	532	13.30	5.216	734.6	196.1	1.958	20807.	5555.
4.5	473	9.85	0.719	75.0	20.0	0.200	2124.	567.
5.0	425	7.30	0.149	11.5	3.1	0.031	327.	87.
5.5	387	5.65	1.432	85.7	22.9	0.228	2427.	648.
6.0	354	4.18	0.384	17.0	4.5	0.045	481.	128.
6.5	327	3.29	1.432	49.9	13.3	0.133	1413.	377.
7.0	304	2.66	0.149	4.2	1.1	0.011	119.	32.
7.5	283	2.23	0.719	17.0	4.5	0.045	481.	128.
8.0	265	1.87	5.216	103.3	27.6	0.275	2926.	781.
8.5	250	1.61	0.719	12.3	3.3	0.033	347.	93.
9.0	236	1.42	0.149	2.2	0.6	0.006	64.	17.
9.5	224	1.25	1.432	19.0	5.1	0.051	537.	143.
10.0	212	1.11	0.384	4.5	1.2	0.012	128.	34.
10.5	202	1.00	1.432	15.2	4.0	0.040	429.	115.
11.0	193	0.91	0.149	1.4	0.4	0.004	41.	11.
11.5	185	0.82	0.719	6.2	1.7	0.017	177.	47.
12.0	177	0.74	5.216	40.9	10.9	0.109	1158.	309.

mode 2
omega squared in (radians/second)**2 = 0.32637297
natural frequency in v.p.m. = 5455.41

no.	inertia	theta	iom2t	sigma m	shaft k	dtheta
1	6.8	1.00000	2.221	2.221	54.6	0.04070
2	50.8	0.95930	15.904	18.125	71.2	0.25440
3	49.5	0.70490	11.387	29.512	71.2	0.41423
4	49.5	0.29067	4.696	34.208	71.2	0.48014
5	49.5	-0.18947	-3.061	31.147	71.2	0.43718
6	49.5	-0.62665	-10.123	21.024	71.2	0.29509
7	49.5	-0.92174	-14.890	6.134	71.2	0.08609
8	49.5	-1.00783	-16.281	-10.147	71.2	-0.14242
9	51.7	-0.86541	-14.609	-24.756	70.9	-0.34927
10	1100.1	-0.51614	-185.308	-210.064	276.8	-0.75898
11	2650.4	0.24284	210.060	-0.004		

mode 2
omega squared 0.32637297
natural frequency 5455.41
sigma i*theta**2 2733.5484
sigma i*theta**2 8207.5240
I INT 27414.38
I EXT 54704.17
stressed diameter of external shaft 16.00
equilibrium amplitude 0.00005098
f in 7487.33
f int 1.40
f ext 2.79
f e 112111670.00
f a 0.
f cr 0.
f cs 0.
f p 0.

order	rpm	tn	vec	tstint	tstext	phi	tmaxi	tmaxe
0.5	10910	11.00	1.389	21.3	42.6	0.058	1602.	3197.
1.0	5455	22.51	0.438	13.8	27.5	0.038	1034.	2063.
1.5	3636	19.00	3.731	99.1	197.7	0.271	7436.	14839.
2.0	2727	7.81	1.238	13.5	27.0	0.037	1015.	2025.
2.5	2182	20.20	3.731	105.3	210.2	0.288	7906.	15776.
3.0	1818	14.27	0.438	8.7	17.4	0.024	655.	1308.
3.5	1558	16.70	1.389	32.4	64.7	0.089	2432.	4854.
4.0	1363	13.30	1.656	30.8	61.4	0.084	2311.	4611.
4.5	1212	9.85	1.389	19.1	38.1	0.052	1435.	2863.
5.0	1091	7.30	0.438	4.5	8.9	0.012	335.	669.
5.5	991	5.65	3.731	29.5	58.8	0.081	2211.	4412.
6.0	909	4.18	1.238	7.2	14.4	0.020	543.	1083.
6.5	839	3.29	3.731	17.2	34.2	0.047	1288.	2569.
7.0	779	2.66	0.438	1.6	3.2	0.004	122.	244.
7.5	727	2.23	1.389	4.3	8.6	0.012	325.	648.
8.0	681	1.87	1.656	4.3	8.6	0.012	325.	648.
8.5	641	1.61	1.389	3.1	6.2	0.009	235.	468.
9.0	606	1.42	0.438	0.9	1.7	0.002	65.	130.
9.5	574	1.25	3.731	6.5	13.0	0.018	489.	976.
10.0	545	1.11	1.238	1.9	3.8	0.005	144.	288.
10.5	519	1.00	3.731	5.2	10.4	0.014	391.	780.
11.0	495	0.91	0.438	0.6	1.1	0.002	42.	83.
11.5	474	0.82	1.389	1.6	3.2	0.004	119.	234.
12.0	454	0.76	1.656	2.2	4.5	0.006	167.	334.

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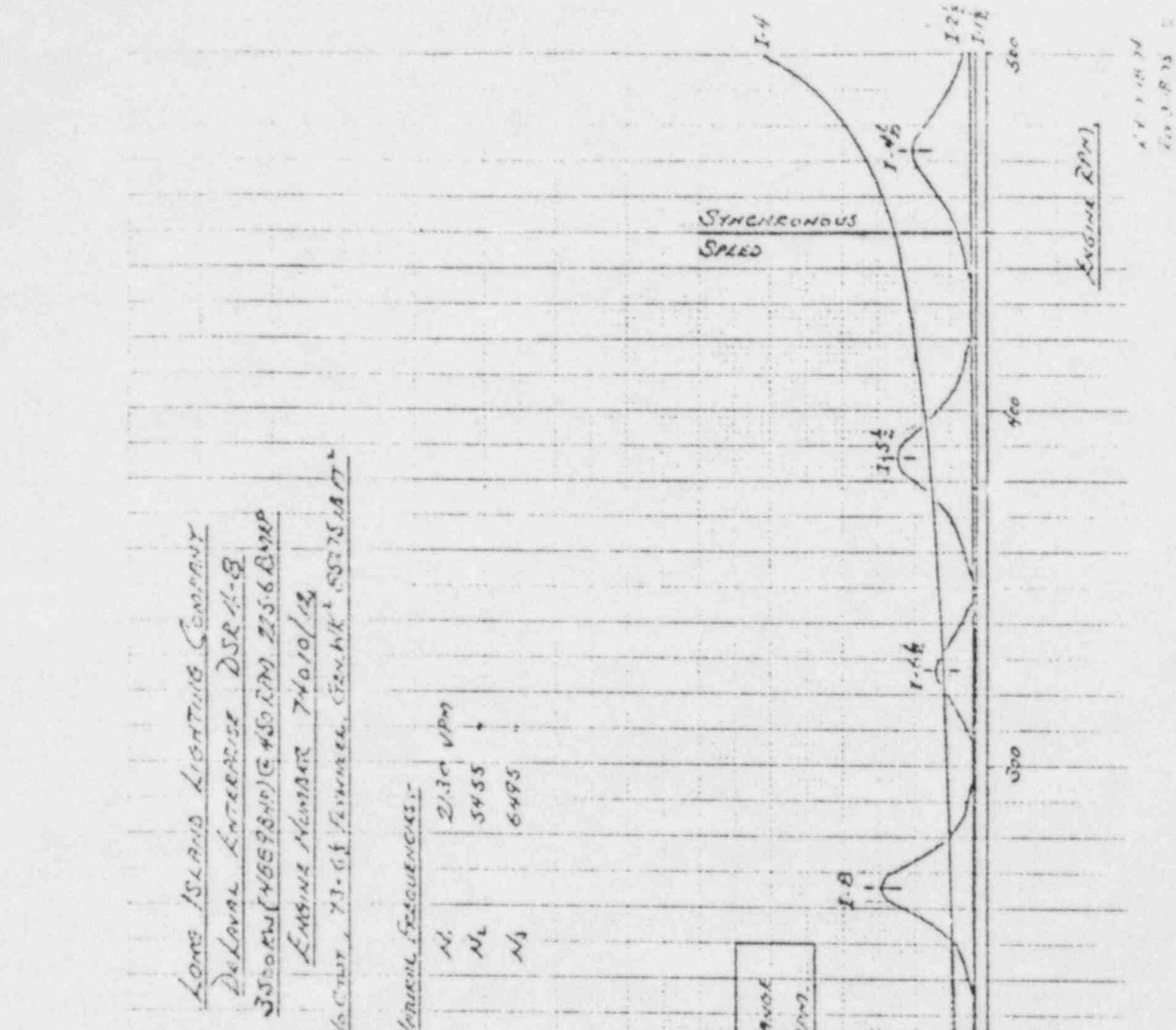
mode 3
omega squared in (radians/second)**2 = 0.46264306
natural frequency in v.p.m. = 6495.21

no.	inertia	theta	iom2t	sigma m	shaft k	dtheta
1	6.8	1.00000	3.148	3.148	54.6	0.05769
2	50.8	0.94231	22.145	25.293	71.2	0.35501
3	49.5	0.58729	13.449	38.742	71.2	0.54378
4	49.5	0.04352	0.996	39.739	71.2	0.55777
5	49.5	-0.51425	-11.776	27.963	71.2	0.39248
6	49.5	-0.90673	-20.764	7.199	71.2	0.10104
7	49.5	-1.00777	-23.077	-15.878	71.2	-0.22287
8	49.5	-0.78490	-17.974	-33.852	71.2	-0.47515
9	51.7	-0.30976	-7.412	-41.265	70.9	-0.58216
10	1100.1	0.27242	138.642	97.378	276.8	0.35183
11	2650.4	-0.07941	-97.377	0.000		

mode 3
omega squared 0.46264306
natural frequency 6495.21
sigma i*theta**2 2421.4814
sigma i*theta**2 3603.3370
I INT 33069.87
I EXT 25355.85
stressed diameter of external shaft 16.00
equilibrium amplitude 0.00008013
f in 7487.33
f int 2.65
f ext 2.03
f e 140778740.00
f o 0.
f cr 0.
f cs 0.
f p 0.

order	rpm	tn	vec	tstint	tstext	phi	tmaxi	tmaxe
0.5	12990	11.00	0.861	25.1	19.2	0.029	955.	732.
1.0	6495	32.85	0.948	82.5	63.2	0.095	3136.	2405.
1.5	4330	19.00	2.946	148.3	113.7	0.171	5640.	4325.
2.0	3247	82.42	2.820	615.9	472.3	0.708	23423.	17961.
2.5	2598	20.20	2.946	157.7	120.9	0.181	5997.	4598.
3.0	2165	17.21	0.948	43.2	33.1	0.050	1643.	1260.
3.5	1855	16.70	0.861	38.1	29.2	0.044	1449.	1111.
4.0	1623	13.30	1.950	68.7	52.7	0.079	2614.	2004.
4.5	1443	9.85	0.861	22.5	17.2	0.026	855.	655.
5.0	1299	7.30	0.948	18.3	14.1	0.021	697.	534.
5.5	1180	5.65	2.946	44.1	33.8	0.051	1677.	1286.
6.0	1082	4.18	2.820	31.2	24.0	0.036	1188.	911.
6.5	999	3.29	2.946	25.7	19.7	0.030	977.	749.
7.0	927	2.66	0.948	6.7	5.1	0.008	254.	195.
7.5	866	2.23	0.861	5.1	3.9	0.006	194.	148.
8.0	811	1.87	1.950	9.7	7.4	0.011	368.	282.
8.5	764	1.61	0.861	3.7	2.8	0.004	140.	107.
9.0	721	1.42	0.948	3.6	2.7	0.004	136.	104.
9.5	683	1.25	2.946	9.8	7.5	0.011	371.	285.
10.0	649	1.11	2.820	8.3	6.4	0.010	315.	242.
10.5	618	1.00	2.946	7.8	6.0	0.009	297.	227.
11.0	590	0.91	0.948	2.3	1.8	0.003	87.	67.
11.5	564	0.82	0.861	1.9	1.4	0.002	71.	55.
12.0	541	0.74	1.950	3.5	2.9	0.004	145.	112.


LONG ISLAND LIGHTING COMPANY
DeLAVAL ENTERPRISE DSR-16-8
3500KW (4659BHP) @ 450 RPM. 225.6 BMEP
ENGINE NUMBER 74010/12
NO CWT, 73-6 1/2 FLYWHEEL, GEN.WK² 55775 LB FT²
NATURAL FREQUENCIES:-
N1 2130 VPM
N2 5455 "
N3 6495 "
CRANKPIN TORSIONAL STRESS PSI
8000
6000
4000
2000
0
AVOID SPEED RANGE 350 TO 400 RPM.
I-8
I-6 1/2
I-5 1/2
I-4 1/2
I-4
I-2 1/2
I-1 1/2
SYNCHRONOUS SPEED
100
200
300
400
500
ENGINE RPM.

J SIMBN

IRBM   11:31PDT   07/18/74

9.374900IE+01

left reaction = -16858.
right reaction = -12894.

delta	dia	weight	inertia	shear	moment	slope	defl
0.	0.	0.	0.	16858.1	0.	0.000127	0.
.812	13.000	143.2	1402.0	16714.9	63990.	0.000124	0.000480
1.250	13.000	47.0	1402.0	16668.0	84854.	0.000122	0.000633
.687	14.000	247.8	1885.7	16420.2	178941.	0.000108	0.001292
.500	24.000	576.1	16286.0	15844.1	251535.	0.000106	0.001776
0.	0.	6935.0	16286.0	8909.1	251535.	0.000106	0.001776
4.500	24.000	576.1	16286.0	8333.0	290330.	0.000104	0.002249
.000	16.000	512.1	3217.0	7820.9	363022.	0.000073	0.003053
.000	18.000	432.1	5153.0	7388.8	408651.	0.000059	0.003450
.250	18.000	522.1	5153.0	6866.7	460327.	0.000035	0.003801
.750	16.570	534.0	3700.5	6332.7	518074.	-0.000000	0.003970
0.	0.	17150.0	3700.5	-10817.3	518074.	-0.000000	0.003970
.250	16.510	560.4	3647.2	-11377.7	415422.	-0.000040	0.003776
1.000	16.498	60.5	3636.6	-11438.2	404014.	-0.000044	0.003735
.000	16.468	602.8	3610.2	-12041.0	286618.	-0.000075	0.003130
.500	16.468	693.2	3610.2	-12734.2	144160.	-0.000098	0.002115
1.250	8.000	160.0	201.1	-12894.2	-0.	-0.000233	-0.000000

lateral frecuency by raleigh's method = 3191.

PROGRAM STOP AT 1688

USED .41 UNITS

000.75 CRU 0000.05 TCH 0002.21 KC

OFF AT 11:33PDT 07/18/74

APPENDIX B

AMERICAN BUREAU OF SHIPPING, RULES FOR BUILDING AND CLASSING STEEL VESSELS

APPENDIX B

Selected Sections Pertaining to Crankshaft Design
Rules for Building and Classing Steels Vessels
American Bureau of Shipping, 1980

The following are selected paragraphs from ABS Section 34, Internal Combustion Engines, and ABS Section 44, Materials for Machinery, Boilers, Pressure Vessels, and Piping.

Although these selections are from the 1980 rules, the technical content differs very little from the 1973, 1974, and 1975 editions of the rules.

34.1 Construction and Installation

34.1.1 General

Construction and installation of all internal-combustion engines and reduction gears intended for propulsion in classed vessels and auxiliary engines and reduction gears of 135 horsepower (hp) and over are to be carried out in accordance with the following requirements, to the satisfaction of the Surveyor. Smaller auxiliaries are to be of approved construction and are to be equipped in accordance with good commercial practice, but need not be inspected at the plant of the manufacturer, whose guarantee of the engine will be accepted subject to satisfactory performance witnessed by the Surveyor after installation. For engines driving generators see also 35.21.

34.1.2 Construction-survey Notification

Before proceeding with the manufacture of materials subject to test and inspection, the Bureau is to be notified in writing that survey is desired during construction, such notice to contain all the necessary information for the identification of the machinery to be surveyed.

34.1.3 Certification on Basis of an Approved Quality Control Program

Upon application, consideration will be given to the acceptance of standardized, mass-produced engines and reduction gears without test and inspection of individual units subject to approval of the manufacturer's quality control program.

34.3 Plans and Particulars to Be Submitted

34.3.1 All Engines

The particulars to be submitted for all engines are to include the type of engine, maximum continuous brake horsepower and revolutions per minute, maximum firing pressure, mean indicated pressure, critical-speed data, and weights of reciprocating parts, weight and diameter of flywheel or flywheel effect for the engine. Material specifications are also to be submitted for approval.

34.3.2 Main Engines

In addition to the plans showing the general arrangement of machinery in the vessel, shafting, stern-bearing details, the sizes and types of various auxiliaries and the sizes and purposes of suction and discharge connections of the pumps, as required in other sections of the Rules to be submitted for approval, the following plans are to be submitted in quadruplicate for approval:

Sectional assembly, bedplate or crankcase including details of breather arrangement, sump ventilation and explosion relief valves, cylinder including jacket and liner, cylinder head, piston and connecting rods, shafting, couplings, clutches, vibration dampers, tie rods if fitted, pressure piping, air containers and details of the following when driven by the main engine: air compressors, scavenge pumps or blowers, turbochargers or superchargers; for indirect drive, plans of the gears, clutches, couplings, generators and motors are to be submitted in accordance with Sections 33 and 35.

34.3.3 Auxiliary Engines

Plans for auxiliary engines are to include a sectional assembly and crankshaft, piston rods, connecting rods, couplings, clutches, vibration dampers, together with pressure piping and air containers and, where fitted, supercharger or turbocharger in sufficient detail for design analysis. The plans are to show details of the breather arrangement, sump ventilation and explosion relief valves when they are required.

34.3.4 Torsional Vibration Stresses

The design equations do not take into consideration the possibility of dangerous torsional vibration stresses, and where propulsion critical-speed arrangements are such that dangerous torsional vibration may occur within the operating range, calculations are to be submitted including tables of natural frequencies, vector summations for critical speeds of all significant orders up to 120% of rated speed, and stress estimates for criticals whose severity approaches or exceeds the limits indicated in 34.57 and Table 34.3.

34.7 Material Tests and Inspection

34.7.1 Specifications and Purchase Orders

Except as indicated in 34.1.3 and 34.7.3, the following material intended for engines which are required to be constructed under survey is to be tested and inspected in accordance with Table 34.1. The material tests so indicated are to be witnessed by the Surveyor in accordance with the requirements of Section 44, and copies in duplicate of purchase orders and specifications for material are to be submitted to the Bureau for the information of the Surveyor. The Surveyor will inspect and test material manufactured to other specifications than those given in Section 44, provided that such specifications are approved in connection with the designs and that they are clearly indicated on purchase orders which are forwarded for the Surveyor's information. All other tests in Table 34.1 are to be carried out by the manufacturer whose affidavit of tests may be accepted by the Bureau.

34.7.2 Steel-bar Stock

Hot-rolled steel bars up to 229 mm (9 in.) in diameter may be used when approved for use in place of any of the items indicated in Table 34.1. See Section 44.

34.7.3 Alternative Test Requirements

Material for engines and reduction gear units of 500 hp or less, including shafting, gears, pinions, couplings, and coupling bolts will be accepted on the basis of the manufacturer's certified mill test reports and a satisfactory surface inspection and hardness check witnessed by the Surveyor.

34.15 Cylinders and Covers, Liners, and Pistons

Parts such as cylinders, liners, cylinder covers, and pistons which are subject to high temperatures or pressures are to be made of material suitable for the stresses and temperature to which they are exposed. When the cylinder diameter is over 230 mm (9 in.), a relief valve, set to relieve at not more than 40% in excess of the maximum firing pressure is to be fitted on each cylinder of reversible engines and engines using air for starting. For auxiliary engines other effective means for determining the maximum cylinder pressure, such as a maximum-pressure indicator, will be specially considered.

34.17 Crankshafts

34.17.1 Diameter of Pins and Journals

The diameter of the crankshaft pins and journals, in mm or in., is not to be less than d as determined by the following equation.

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

Metric Units	*Inch Pound Units*
$M = 1.86PD^2L$	$M = 0.131PD^2L$
$T = 1.02 \times 10^6 H/R$	$T = 63{,}000H/R$

D = diameter of cylinder bore, in mm or in.
P = maximum firing pressure, in kg/cm^2 or psi
L = span between bearings, measured over the web, in mm or in.
H = hp at rated speed   brg. edge to brg. edge
R = rpm at rated speed
c = 1.16 for one-cylinder engines
= 1.13 for two-cylinder engines
= 1.10 for three-cylinder engines
= 1.07 for four-cylinder engines
= 1.04 for five-cylinder engines
= 1.02 for six-cylinder engines
= 1.00 for engines with more than six cylinders
f = 1,900 for Grade 2 forgings
= 2,140 for Grade 3 forgings
= 2,310 for Grade 4 forgings

Values of f for other materials are subject to special consideration.

Note The above equation will usually apply to engines where a bearing adjoins each side of each crank and where single impulses occur at equal intervals. It may apply to other engines if M is modified to reflect the appropriate bending moments. Increased dimensions may be required where critical-speed arrangements or stress concentrations are not favorable. Where crankshaft dimensions are proposed which are less than those determined by the above equation complete supporting data, including detailed stress analysis, are to be submitted for special consideration.

34.17.2 Maximum Firing Pressure and BHP

The Surveyor is to verify the maximum firing pressure P and brake horsepower during the full power trial of the engine. When the engine builder has demonstrated to the Surveyor by means of tests on a pilot engine that the design value of P is not exceeded within established limits of production tolerances and settings which would affect it, verification of P will not be required for an engine built on a production line, provided the engine delivers its rated power within the established limits.

34.17.3 Higher Ratings

Subsequent adjustments for the purpose of obtaining higher powers or higher maximum pressures will be subject to special consideration.

34.17.4 Solid Crankshaft Webs

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

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

w = effective width of web in mm or in.
t = thickness of web in mm or in.

Where the proportions are such that pins and journals overlap, t may be taken to be the minimum diagonal distance through the web.

34.57 Torsional Critical-speed Arrangements and Stress Limits

34.57.1 Allowable Stresses

Where torsional critical-speed arrangements differ significantly from previous installations, the torsional vibration stress in propeller shafts and propulsion-engine crankshafts, due to a single harmonic exciting factor at the resonant peak, is not to exceed the limit indicated in Table 34.3. Total vibratory stress in the interval from 90% to 105% of rated speed due to resonant harmonics and the dynamically magnified parts of significant nonresonant harmonics is not to exceed 150% of the allowable stress for a single harmonic exciting factor.

34.57.2 Barred Ranges

When torsional vibratory stresses exceed the foregoing limits, at an rpm within the operating range but less than 90% of rated speed, a barred range is to be provided. The tachometer is to be marked, and a warning notice fitted to the engine and at the operating controls, to the effect that continuous operation within the barred range is to be avoided. The width of such barred range is to take into consideration the breadth and severity of the critical, but is to extend

at least 5% above and 5% below the speed at the resonant peak. A barred range is not acceptable in the interval from 90% to 100% of rated speed.

34.57.3 Other Effects

Because torsional vibration has deleterious effects other than shafting fatigue, the limits in Table 34.3 are not intended for direct application as design factors, and it is desirable that the service range above 90% of rated speed be kept clear of torsional criticals insofar as practicable.

34.57.4 Torsiograph Tests

When the calculations indicate that criticals occur within the operating range whose severity approaches or exceeds the limits in Table 34.3, torsiograph tests may be required to verify the calculations and to assist in determining ranges of restricted operation.

34.57.5 Vibration Dampers

When torsional vibratory stresses exceed the limits in Table 34.3 and a barred range is not acceptable, the propulsion system is to be redesigned, or vibration dampers are to be fitted to reduce the stresses.

34.57.6 Gears

When the propeller is driven through reduction gears, or when auxiliary equipment, such as a blower, is driven through gears, a barred range is to be provided at the critical speed if gear-tooth chatter occurs during continuous operation at the critical.

TABLE 34.1 (continued)

Item[1]	*Material Tests*	*Nondestructive Tests*[2,3]
Steel gear wheels for camshaft drives	above 400 mm (15.7 in.) bore	above 400 mm (15.7 in.) bore
Supercharger, turbocharger shaft and rotor	above 300 mm (11.8 in.) bore	above 400 mm (15.7 in.) bore
Cast steel elements, including their welded connections for bedplates (e.g. main bearing housing)[5]	all	all
Bedplate of welded construction, plates and transverse bearings girders made of forged or cast steel	all	—
Frame and crankcase of welded construction	all	—
Entablatures of welded construction	all	—
Thrust shafts, lineshafts, couplings, coupling bolts, propeller shafts, also generator shafts and motor and gear shafts for indirect drive	*all	—

Notes

1. This table does not cover parts of the engine such as pipes and accessories of the starting air system and other pressure systems.
2. Magnetic particle, liquid penetrant or equally effective tests are to be carried out where nondestructive testing is indicated.
3. Ultrasonic testing is additionally required.
4. Ultrasonic testing is additionally required where the bore exceeds 400 mm (15.7 in.).
5. Nondestructive testing may be required by the Surveyor for cast-steel components. See 44.23.8a.

*Material tests to be witnessed by the Surveyor.

TABLE 34.2

Test Pressures for Parts of Internal-combustion Engines

P is the maximum working pressure in the part concerned

Item	*Test Pressure*
Cylinder cover, cooling space	7 kg/cm² (100 psi)
Cylinder liner, over the whole length of cooling space	7 kg/cm² (100 psi)
Cylinder jacket, cooling space	4 kg/cm² (57 psi) but not less than 1.5*P*
Exhaust valve, cooling space	4 kg/cm² (57 psi) but not less than 1.5*P*
Piston crown, cooling space (after assembly with piston rod)	7 kg/cm² (100 psi)
Fuel-injection system	
Pump body, pressure side	1.5*P* or *P* + 300 kg/cm² (4270 psi) whichever is less
Valve	1.5*P* or *P* + 300 kg/cm² (4270 psi) whichever is less
Pipe	1.5*P* or *P* + 300 kg/cm² (4270 psi) whichever is less
Scavenge-pump cylinder	4 kg/cm² (57 psi)
Turboblower, cooling space	4 kg/cm² (57 psi) but not less than 1.5*P*
Exhaust pipe, cooling space	4 kg/cm² (57 psi) but not less than 1.5*P*
Engine-driven air compressor, cylinders, covers, intercoolers and aftercoolers	
air side	1.5*P*
water side	4 kg/cm² (57 psi) but not less than 1.5*P*
Coolers, each side	4 kg/cm² (57 psi) but not less than 1.5*P*
Engine-driven pumps (oil, water, fuel, bilge)	4 kg/cm² (57 psi) but not less than 1.5*P*

TABLE 34.3

Allowable Stress Values for Crankshafts and Tail Shafts Due to a Single Harmonic (Grade 2 Steel)

Engine Speed	*0.3R or less*	*0.8R*	*0.95R–1.00R*	*1.05R*
Diameter 300 mm (11.8 in.) or less	±400 kg/cm² (5,689 psi)	±250 kg/cm² (3,556 psi)	±150 kg/cm² (2,134 psi)	±250 kg/cm² (3,556 psi)
Diameter 600 mm (23.6 in.) or more	±320 kg/cm² (4,551 psi)	±200 kg/cm² (2,845 psi)	±120 kg/cm² (1,707 psi)	±200 kg/cm² (2,845 psi)

Notes

1. Stress limits for speeds intermediate between those shown in Table 34.3, and for shafts between 300 and 600 mm (11.8 in. and 23.6 in.) in diameter, may be obtained by interpolation. In the Table, *R* is rpm at rated speed, which is the speed at maximum continuous rating for regular operation in service. Stresses are nominal values based on diameter of crankpins or on the minimum propeller-shaft diameter between the big end of the taper and the forward stern gland, disregarding stress-concentration factors.
2. Where the service is such that the vessel will operate for a significant portion of its service life at speeds below 90% of rated speed, the stress limits in the interval 0.95*R*–1.00*R* of Table 34.3 are to be used in such speed ranges.
3. If torsional critical-speed arrangements are similar to previous installations proven by service experience, consideration will be given to higher stresses upon submittal of full details.
4. Stress limits for crankshafts made of Grade 3 or 4 or approved alloy-steel forgings may be increased by two-thirds of the percentage increase in ultimate tensile strength over 42 kg/mm² (60,000 psi).

44.19 Steel Machinery Forgings

Note In substantial agreement with ASTM designations:
A668—Class B for ABS Grade 2
A668—Class D for ABS Grade 3
A668—Class F for ABS Grade 4

44.19.1 Process of Manufacture

a ***General*** The following requirements cover carbon-steel forgings intended to be used in machinery construction. This does not preclude the use of alloy steels as permitted by 44.1.1. The steel is to be fully killed and is to be made by one or more of the following processes: open-hearth, basic-oxygen, electric-furnace, or such other process as may be approved. The cross-sectional area of the main body of the unmachined, finished forging is not to exceed one-third of the area of the ingot; palms, flanges and similar enlargements on the forging are not to exceed two-thirds of the area of the ingot. A sufficient discard is to be made from each ingot to secure freedom from piping and undue segregation.

b ***Chemical Composition*** The chemical composition is to be reported and the carbon content is not to exceed 0.35% unless specially approved. Specially approved grades having more than 0.35% carbon are to have S marked after the grade number.

44.19.2 Marking and Retests

a ***Marking*** In addition to appropriate identification markings of the manufacturer, the Bureau markings, indicating satisfactory compliance with the Rule requirements, and as furnished by the Surveyor, is to be stamped on all forgings in such location as to be discernible after machining and installation. In addition Grade 2, Grade 3, and Grade 4 forgings are to be stamped $\frac{AB}{2}$, $\frac{AB}{3}$, and $\frac{AB}{4}$ respectively.

b ***Retests*** If the results of the physical tests for any forgings or any lot of forgings do not conform to the requirements specified, the manufacturer may re-treat the forgings, but not more than three additional times. Retests of an additional specimen or specimens are to be made and are to conform to the requirements specified.

44.19.3 Heat Treatment

a ***General*** Unless a departure from the following procedures is specifically approved, Grade 2 and 3 forgings are to be annealed, normalized or normalized and tempered. Grade 4 forgings are to be normalized and tempered or double-normalized and tempered. The furnace is to be of ample proportions to bring the forgings to a uniform temperature.

b ***Cooling Prior to Heat Treatment*** After forging and before reheating for heat treatment, the forgings are to be allowed to cool in a manner to prevent injury and to accomplish transformation.

c ***Annealing*** The forgings are to be reheated to and held at the proper austenitizing temperature for a sufficient time to effect the desired transformation and then be allowed to cool slowly and evenly in the furnace until the temperature has fallen to about 455C (850F) or lower.

d ***Normalizing*** The forgings are to be reheated to and held at the proper temperature above the transformation range for a sufficient time to effect the desired transformation and then withdrawn from the furnace and allowed to cool in air.

e ***Tempering*** The forgings are to be reheated to and held at the proper temperature, which will be below the transformation range, and are then to be cooled under suitable conditions.

44.19.4 Tensile Properties

a *Carbon-steel Forgings* The carbon-steel forgings are to conform to the requirements of Table 44.11 as to tensile properties.

b *Large Forgings* In the case of a large forging requiring two tension tests, the range of tensile strength is not to exceed 7 kg/mm^2 (10000 psi).

c *Application* Subject to the approval of the appropriate material for each design application, Grade 2 is approved for all purposes; Grades 3 and 4 are approved for all purposes excepting propeller shafts.

d *Alloy or Special Carbon Steels* When alloy steels or carbon steels differing from the above requirements are proposed for any purpose, the purchaser's specification is to be submitted for approval in connection with the approval of the design including such application. Specifications such as ASTM A237 or A470 or other steels suitable for the intended service will be considered.

44.19.5 Test Specimens

a *Location of Specimens* The physical properties are to be determined from test specimens taken from prolongations having a sectional area not less than the body of the forging. Specimens may be taken in a direction parallel to the axis of the forgings in the direction in which the metal is most drawn out or may be taken transversely. The axis of longitudinal specimens is to be located at any point midway between the center and the surface of solid forgings and at any point midway between the inner and outer surfaces of the wall of hollow forgings. The axis of transverse specimens may be located close to the surface of the forgings.

b *Hollow-drilled Specimens* In lieu of prolongations, the test specimens may be taken from forgings submitted for each test lot; or if satisfactory to the Surveyor's test specimens, may be taken from forgings with a hollow drill.

c *Small Forgings* In the cases of small forgings weighing less than 114 kg (250 lb) each, where the foregoing procedures are impracticable, a special forging may be made for the purpose of obtaining test specimens, provided the Surveyor is satisfied that these test specimens are representative of the forgings submitted for test. In such cases the special forgings should be subjected to approximately the same amount of working and reduction as the forgings represented and should be heat-treated with those forgings.

d *Identification of Specimens* The test specimens are not to be detached from the forgings until the final heat treatment of the forgings has been completed nor until the test specimens have been stamped by the Surveyor for identification.

44.19.6 Number of Tests

a *Large Forgings* In the case of large forgings with rough machined weights over 4080 kg (9000 lb) each, one tension test is to be made from each end of the forging.

b *Smaller Forgings* In the case of forgings with rough machined weights less than 4080 kg (9000 lb) each, except as noted in the following paragraph, one tension test is to be made from each forging.

c Small Forgings In the case of small forgings with rough machined weights less than 227 kg (500 lb) each, one tension test may be taken from one forging as representative of a lot of 908 kg (2000 lb) or less, provided the forgings in each such lot are of similar size, are of one grade and kind only, are made from the same heat and are heat-treated in the same furnace charge. For lots over 908 kg (2000 lb), only one tension test need be taken from one small forging as representative of a lot provided 20% of the other forgings in each such lot, not subjected to tensile tests, are subjected to Brinell hardness tests and meet the following requirements.

Grade	*Brinell Hardness Test* — *Minimum 10 mm ball, 3000 kg load*
2	120
3	150
4	170

d Special Situations In the case of a number of pieces cut from a single heat-treated forging, individual tests need not necessarily be made for each piece, but such forging may be tested in accordance with whichever of the foregoing procedures is applicable to the primary heat-treated forging involved.

44.19.7 Inspection

All forgings are to be inspected by the Surveyor after final heat treatment and they are to be found free from defects.

TABLE 44.11
Tensile Property Requirements for Carbon-steel Machinery Forgings

Grades	*Size* — *Over mm (in.)*	*Size* — *Not over mm (in.)*	*Tensile Strength, min kg/mm² (psi)*	*Yield Point/ Yield Strength kg/mm² (psi)*	*Longitudinal Specimens* — *Elongation in 50 mm (2 in.) min percent*	*Longitudinal Specimens* — *Reduction of Area, min percent*	*Transverse Specimens* — *Elongation in 50 mm (2 in.) percent*	*Transverse Specimens* — *Reduction of Area, min percent*
2		300 (12)			25	38		
	300 (12)		42 (60000)	21 (30000)	24	36	20	29
3		200 (8)			24	40		
	200 (8)	300 (12)	53 (75000)	26.5 (37500)	22	35	18	28
	300 (12)	500 (20)			20	32		
	500 (20)				19	30		
4			58.5 (83000)	30.5 (43000)	20	35	17	27

Note When tangential specimens are taken from wheels, rings, rims, discs, etc. in which the major final hot working is in the tangential direction, the tensile test results are to meet the requirements for longitudinal specimens.

APPENDIX C

INSPECTION COMMENTS CONCERNING DIESEL GENERATORS

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APPENDIX C

The comments herein are based on an inspection made as part of an initial visit to the Shoreham Nuclear Power Station on September 1, 1983.

C.1 Diesel Generator 101

Diesel generator (DG) 101 was located in its operational room and was being prepared for a limited test program. The crankshaft was exposed for inspection, cracks in cranks 5 and 7 were being ground out to reduce the associated stress concentration, and preparations were being made to install instrumentation. Instrumentation specialists at DG 101 indicated that the planned instrumentation included a strain gage torque bridge on the crankshaft adjacent to the flywheel, strain gage rosettes in the crankpin fillets of cranks 5 and 7, vibration transducers (accelerometers) on various bearing journals, pressure transducers in two combustion cylinders, an angular displacement transducer (torsiograph) on the free end of the crankshaft (opposite the flywheel and generator), and a sensor on the generator shaft to indicate shaft position relative to top dead center of crank 7. The instrumentation was in various stages of preparation and installation.

The cracks on the crankshaft appeared to have been nearly ground out in accordance with the torsional test procedure.* Cracks in the crankshaft of DG 101 were reported to have been approximately 1 inch deep prior to grinding. Crack locations included cranks 5 and 7, and the cracks made an approximate angle of 45° relative to the crankshaft (or crankpin) longitudinal axis. The crack on crank 7 was located in a 5 o'clock position relative to top dead center of crankpin 7 and on the fillet toward crank 8. The crack on crank 5 was located in a similar manner, but in a 7 o'clock position on the crank pin fillet toward crank 6.

C.2 Diesel Generator 102

DG 102 was the unit with the fractured crankshaft. DG 102 had already been moved to the main turbine deck where space and crane facilities were

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* Revision, LILCO Procedure for Crankshaft Testing, Emergency Diesel Generator No. 101, September 23, 1983.

available to disassemble the unit, make a thorough inspection, and rebuild it with the 13 x 12 crankshaft. Disassembly and inspection of the whole engine was progressing part by part; although it had not progressed to the point of removing the fractured crankshaft, the fracture was clearly visible and open to close inspection through the sides of the engine block where the cover plates had been removed. Inspection of crank 7 revealed a fracture through the crank web and partially through the crankpin, with the fracture passing through the crankpin fillet at approximately the 5 o'clock and 7 o'clock positions with respect to top dead center of that crank. The tip of the V-shaped crack propagating out into the crankpin reached approximately to the midpoint of the crankpin bearing surface.

Further inspection of the fractured crankshaft (still assembled in DG 102) revealed that one edge of the web at the fracture had a large discolored area characteristic of heating to a temperature range of 400°F to 600°F. This discoloration was attributed to the considerable energy dissipated in sliding contact at the point of fracture and against the connecting rod during the short time (approximately 1 1/2 minutes) that the diesel was believed to be under power (see page 3 of Reference 18) following the fracture.

Inspection of the sump revealed considerable debris under crank 7 as compared to other crank positions. Although accumulated dirt in the engine sump was heaviest toward the flywheel and generator end of the sump, the excessive accumulated debris of crank 7 proved to be mainly bearing material scraped out of the connecting rod bearing by the displaced fractured segment of the crankpin that acted as a sharp cutting tool during those moments of operation following crankshaft fracture.

Further inspection of the crankshaft failure was not conducted because the crankshaft was to be removed over the holiday (Labor Day) weekend and transported to the facilities of Failure Analysis Associates, Inc., in Palo Alto, CA, for immediate extensive examination.

C.3 Diesel Generator 103

DG 103 was observed to be under disassembly in its operational room in preparation for moving it to the turbine deck.

In the initial briefing, the crankshaft of DG 103 had been reported to contain cracks of sufficient depth and magnitude to preclude further operation. Testing was to be performed using DG 101 only. The action plan for DG 103 reportedly called for complete disassembly, inspection, rework as necessary, and reassembly with a 13 x 12 crankshaft (13-in journal bearing diameter and 12-in crankpin diameter) now recommended and supplied by Transamerica Delaval, Inc. Analytical studies of the engine with this crankshaft were to be carried out concurrently, with updates to the analysis being made as test data on DG 101 became available.

APPENDIX D

RECOMMENDATIONS FOR MECHANICAL AND ELECTRICAL COUPLING INVESTIGATION

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APPENDIX D

The following three recommendations were made* to ensure the recording of possible electrical-mechanical dynamic interactions in the course of diesel generator testing.

1. The generator output voltages for the three phases were not all slated for recording -- only the voltage difference between phases, ($V_A - V_B$) and ($V_B - V_C$). Hence, if electrical interactions occurred during the tests, a positive voltage phase reference would not be assured, but would be dependent upon the 3-phase generator voltage output remaining balanced. That is, only two measurements were being made, and that fact required the third voltage to be calculated from the 3-phase electrical vector relationship. This is possible only with the assumption that the voltage remains balanced on all three phases. When it was reported that the third voltage for recording could only be obtained with considerable difficulty, it was recommended that the voltage on each of the three phases be read and recorded separately (from the control room) so that each voltage would be known, should it be required for vector calculations. Although it was believed that the electrical power grid would certainly remain balanced during the recording of data during the synchronous load tests, the reading of the voltages would remove all doubt. For loadings derived from plant equipment (core spray, etc., versus the electrical power grid), the measurement of voltage was more meaningful.

2. Even though the generator rotor inertia was large, it was believed prudent to provide for the investigation of generator instabilities, especially since there could be significant cyclic torque at 30 Hz. Accordingly, it was recommended that all vibrational data be recorded at power factors between 0.8 and 1.0 in synchronous load tests under as high a load as feasible. A load of 2550 kW was chosen from the test procedure to minimize engine run time at or near full load.

   The background of this recommendation is that synchronous generators tend to be more unstable with low excitation (1.0 power factor) than with higher excitation (0.8 power factor). Also, page 1 of LILCO's response to the NRC Request for Information II.2 (September 20, 1983) indicated that a significant amount of the testing on DG 102 was performed at a power factor of 1.0 and that DG 102 was operating at

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* R. C. Herrick, FRC
Memo to C. Petrone, NRC Resident Inspector, Shoreham Nuclear Plant
Subject: On-site for Torsional Test Monitoring
September 24, 1983

power factor of 1.0 at the time of failure. Further, means were not available immediately prior to the test to determine if the 30-Hz cyclic torque could aggravate generator instability. Hence, this consideration appeared to be a prudent course of action for thorough coverage of possible instabilities or mechanical-electrical interactions.

3. It was also recommended that assurance be provided for the recording of any transients, mechanical and/or electrical, associated with synchronization and attachment of the diesel generator to the electrical power grid, and to any of the isochronous loads.

APPENDIX E

COMMENTS BY H. W. HANNERS ON THE SUMMARY OF SELECTED FAILURES AND EVENT REPORTS OF TDI DIESEL GENERATORS

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APPENDIX E

Comments on the items as dated:

08/12/83

The broken crankshaft is believed to have been the result of excessive stress due to torsional vibration.

03/30/83

These screws may have failed as the result of inertial forces from engine operation at or near a torsionally critical speed as well as possible low quality in material, design or manufacture.

03/08/83

The cracked cylinder heads could have been the result of design, but the new design apparently needs to be tested by actual use and acceptance tests.

03/03/83

The high pressure fuel line failures could surely be reduced by design improvements of the shroud (usually called sleeve). Again, sufficient proof remains to be seen through in-service experience and acceptance tests.

12/13/82

Better quality obviously needed.

09/17/82

The omission or removal of the keyway in the water pump shafts could be avoided by eliminating the "stress raiser" effect of the keyway in torsional vibration. An impeller design change to reduce the rotary moment of inertia could also help.

07/22/82

Probably fixed by the change of design.

06/23/82

Change to neoprene is certainly an improvement over isoprene.

05/13/83

This probably should have been 05/13/82 from the "backtracking" review scheme being used.

05/13/82

The shorter capscrews may be satisfactory, but were the original screws actually too long or the threaded tapped holes too shallow?

03/19/82

Neither the problem nor the solution (or fix) are clearly explained. The 53-minute bleed down time is too long as a practical fix. Even 53 seconds is rather a long time to consider acceptable. Successive starts should certainly be allowed more often than 53 minutes apart. Seismic qualification of the sensing line is recommended.

03/15/82

This implies that the rear crankcase cover is a stress bearing part. Neither the strongest bolts nor the reasons for the basic failure give faith in the TDI remedy or explanation of the failure. More proof and further explanation are needed.

12/09/81

The TDI remedy of a lower oil cooler mourting seems reasonable, but the complete system should be reviewed.

11/05/81

The use of Belleville washers in the two-piece piston design may or may not be satisfactory, depending on whether the heat from the hot piston crown anneals the Belleville washers. Heat barrier design may also be required for success.

The cylinder liner grooving and the bearing grooving may very well be caused by "built in" dirt and chips in the original factory assembly. All three engines should certainly have the crankcases or bedplates thoroughly cleaned and all bearings examined and replaced as needed. A tedious, careful, and expensive job is indicated. This means not only the bearings, but also the surface of the mating parts such as the crankshaft crankpin and main journals and other parts would be damaged.

07/14/83 (or 07/14/81)?

Another indication of possible excessive vibration and or lack of proper clamping to prevent cracking of oil lines. Danger of fire from oil line fracture should certainly be given more attention.

03/23/81

Motors should certainly be qualified and not merely be stated to be equivalent.

12/16/80

A redesign of the lube oil system is indicated as necessary so that the turbocharger bearings get oil immediately after a start. This may mean a change to an intermediate drain back sump in addition to the main oil sump of the turbocharger. Acceptance of occasional "fast starts" is not sufficient as this is tantamount to saying that dry bearings are tolerable.

All of the remarks and critique of the items regarding the TDI engines are intended to be constructive and helpful. However, practically all of the suggestions are subject to testing in actual service and qualification under NRC regulations.

During the nuclear power plant survey and inspections done in 1978 and 1979, performed at the University of Dayton (Dayton, Ohio), a grand total of 288 items were investigated. Most of these subject items were found in every power plant of this survey.