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UNITED STATES OF AMERICA
NUCLEAR REGULATORY COMMISSION

OFFICE OF SECRETARY
DOCKETING & SERVICE
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BEFORE THE ATOMIC SAFETY AND LICENSING BOARD

In the Matter of)
)
THE CLEVELAND ELECTRIC)
ILLUMINATING COMPANY, ET AL.)
)
(Perry Nuclear Power Plant,)
Units 1 and 2))

Docket Nos. 50-440
50-441

APPLICANTS' DIRECT TESTIMONY OF
CHARLES D. WOOD III ON ISSUE NO. 16

1. My name is Charles D. Wood III. I am the Vice President of the Engines, Emissions, and Vehicle Research Division of Southwest Research Institute ("SwRI"), a position I have held since December of 1983. My business address is Southwest Research Institute, P. O. Drawer 28510, San Antonio, Texas 78284.

2. I am directly and personally involved with diesel engine research, design, and testing at SwRI, and supervise our diesel engine research staff. My experience with diesel engines ranges from high speed automotive diesels to medium speed railroad and ship diesels. Two twelve-cylinder railroad diesel

engines and two two-cylinder electromotive diesel engines are installed at SwRI. These engines are operated by my Division for various engine research programs, and the experience gained from working with these four engines is transferrable to the TDI engines.

3. I have overseen the design review work conducted for Cleveland Electric Illuminating Company ("CEI") on the sixteen Phase I components identified in the Owners Group Program. I have visited the Perry Nuclear Power Plant ("PNPP") and observed the Transamerica Delaval, Inc. ("TDI") engine installations.

4. I have been employed by SwRI since 1962. I joined the company as a Senior Research Engineer in the Automotive Research Department. I later acted as Section Manager for the Department and in 1974 became the Director of the SwRI Department of Engine and Vehicle Research. Prior to my work at SwRI, I was employed as a test engineer and, subsequently, as a propulsion engineer at Ling-Temco-Vought, Inc.

5. I have a Bachelor of Science degree as well as a Master of Science degree in mechanical engineering. I am a member of both the SAE and the Texas Society of Professional Engineers. I am a registered professional engineer in the State of Texas.

6. The following testimony addresses Ohio Citizens for Responsible Energy's ("OCRE") contention regarding the TDI diesel generators in place at PNPP. As admitted by the Licensing Board, Issue No. 16 states:

Applicant has not demonstrated that it can reliably generate emergency on-site power by relying on four Transamerica Delaval diesel generators, two for each of its Perry units.

The following testimony discusses SwRI's review of the Owners Group's analysis of potentially generic known problems with the diesels. Based on its review, SwRI has concluded that the diesel generators will reliably perform their intended safety-related functions, providing PNPP implements the Owners Group and SwRI's recommendations regarding maintenance and inspections.

I. THE OWNERS GROUP PROGRAM PLAN

7. The TDI Diesel Generator Owners Group Program was developed to assess the adequacy of the various TDI engine design configurations to perform their intended safety-related functions. The Program involves three major elements, which, by a combined approach involving design reviews and analyses of engine components, quality revalidations of important attributes, and expanded engine testing and component

inspections, will provide reasonable assurance of the ability of the TDI engines to provide reliable backup power supplies for nuclear power plant service. The first major program element, characterized as Phase I, involves the resolution of potentially generic known problems. Based on operational experience data pertinent to the TDI engines, the Owners Group determined that a limited number of components warranted prioritization and consideration as significant problems with potentially generic applicability. See Applicants' Direct Testimony of John C. Kammeyer on Issue No. 16.

8. The Owners Group prepared task descriptions specifying the analysis and evaluation to be conducted for each of these components. This work was completed by Owners Group consultants. Final reports have been completed covering each component.

II. OVERVIEW OF SWRI'S ROLE

9. SwRI has reviewed the TDI Diesel Generator Owners Group Program reports and backup material on the sixteen Phase I components with regard to their applicability to the PNPP diesel engines. This task was defined by three subtasks requiring review and critique of: (1) the component criticality definition as provided by the Owners Group; (2) the component function and attributes as defined by the Owners Group; and

(3) the assumptions, methodologies, codes and standards, results, conclusions, and recommendations in the Owners Group reports.

10. Staff members chosen to conduct the reviews were selected for their experience in the required disciplines such as engine design and testing, metallurgy, stress analysis, fracture mechanics, and finite element modeling. Where necessary, copies of supporting materials, such as calculations, metallurgical reports, etc., were also obtained. Three of SwRI's staff visited the Owners Group offices in Charlotte, North Carolina where supporting material and engine drawings were reviewed.

11. Each of the Owners Group reports was reviewed to assess the overall methodology applied, the assumptions and parameters used, the analytical approaches applied, accuracy of analysis, the conclusions derived, and finally, the recommendations made. In some instances, SwRI made independent calculations using different equations, different assumptions, and/or different parameters.

12. The following discussion concerning each of the components reviewed compares SwRI's results, conclusions, and recommendations with those of the Owners Group. It includes:

(1) the component function and its critical attributes; (2) its failure history; (3) a review of the Owners Group and SwRI

methodology, and a discussion of any differences; (4) a discussion of any testing; and (5) a review of the Owners Group and SwRI results, conclusions, and recommendations, and a discussion of any differences.

III. SwRI REVIEW OF THE PHASE I COMPONENTS

A. Rocker Arm Capscrew

13. The rocker arm capscrews in TDI standby diesel generators transmit resultant loads from the valve springs, valve opening pressure, pushrods, and rocker arm assemblies to the subcover and cylinder heads. In formulating the design review task description for this component, the Owners Group determined that the critical functional attribute of the rocker arm capscrews was that they have sufficient strength to withstand preload and oscillation loads without fatigue cracking, unacceptable preload relaxation, or thread distortion.

14. The rocker arm capscrews were included among the sixteen Phase I components due to isolated failures resulting from insufficient preload application in TDI nuclear standby diesel engines. Two rocker arm capscrew designs were evaluated by the TDI Owners Group, the original "straight shank" type capscrew and a modified "necked shank" design.

15. A stress analysis was performed for both of the rocker arm capscrew designs by the Owners Group.^{1/} The analysis included a determination of the applied stress and the endurance limits of the two designs, a fatigue life analysis, a thread distortion analysis, and a thermal stress evaluation.

16. Satisfactory fatigue life has been demonstrated by several engines with more than 750 hours (10^7 cycles) of operation. This operational history verifies that fatigue failures will not be a problem.

17. SwRI reviewed the Owners Group report on rocker arm capscrews to verify its applicability to the "straight shank" capscrews in place at PNPP. Design parameters used by SwRI were in agreement with those used by the Owners Group. Certain assumptions made by SwRI in reviewing the design analysis were, in general, more conservative than those used by the Owners Group. These assumptions concerned the: 1) modifying factors used to calculate the endurance limit (OG: 8.7 ksi, SwRI: 8.6 ksi); 2) intake valve spring damper force (OG: 0.0, SwRI: 121 lbf./in.); and 3) percent of torque contributing to torsional stress (OG: 75%, SwRI: 50%).

^{1/} A detailed discussion of the analysis performed is included in Stone & Webster Engineering Corporation, "Emergency Diesel Generator Rocker Arm Capscrew Stress Analysis," March, 1984 and its Supplement, April, 1984, both prepared for the Owners Group.

18. SWRI performed a number of analyses on both the original and modified designs to evaluate their respective attributes. The maximum cyclic loading on the capscrews was calculated to be 10,271 lbs. The adequacy of the capscrews to withstand this loading was determined using a series of standard engineering calculations culminating with the use of the modified Goodman line failure criteria to consider both mean (original design: 43.2 ksi, modified design: 42.5 ksi) and alternating (original design 2.73 ksi; modified design: 2.02 ksi) stresses. The maximum biaxial stresses were calculated for the capscrews during tightening to the specified torque (365 lbf.-ft). The principal stresses were compared to the failure criteria of the distortion energy theory. Analyses were also conducted to determine performance under thermal stresses, whether creep would result in loss of preload or capscrew failure, if thread stripping in the subcover was likely, and the impact on the capscrew if the preload is lost. These analyses, conducted by SwRI, were to confirm the critical functional attributes, namely, that the studs have sufficient strength to withstand preload and cyclic loads without fatigue failure, and that neither loss of preload nor thread distortion would occur.

19. No significant differences between the results obtained in the Owners Group analysis and those of SwRI were

observed. There was a difference in the calculated cyclic load on the capscrews (OG: 8.3 ksi, SwRI: 10.3 ksi) and the calculated fatigue life safety factors for both the original (OG: 1.61, SwRI: 1.46) and modified (OG: 1.88, SwRI: 1.68) capscrews. The calculated capscrew endurance limits were also slightly different (OG: 8.7 ksi, SwRI: 8.6 ksi) as were the safety factors against failure during tightening (OG: 1.4, SwRI: 1.7). Both safety factors were satisfactory.

20. No significant thermal stresses will be introduced to the capscrew. Results indicate both the capscrew and the rocker arm shaft will be exposed to the same temperature and that both have the same coefficients of thermal expansion. This means that no significant difference will exist between the thermal expansion of the capscrews and that of the rocker arm shaft over the clamped length.

21. Analyses by SwRI and the Owners Group also indicate that the maximum operating temperature (190°F) is well below the point of significant temperature effects (usually 600°F). The maximum operating stress in the capscrew is approximately 38% of the ultimate strength. This is low enough, when combined with the relatively low temperatures, to dismiss creep as a concern.

22. Based on the results of its analyses, the Owners Group concluded that both the original and the modified designs were adequate for nuclear service. The modified "necked shank" design was found to be somewhat more resistant to fatigue failure and less likely to lose its preload due to its lower cyclic load. The threads utilized for both designs were determined to adequately resist distortion during preload application and the material utilized for the modified rocker arm capscrew design was determined to meet or exceed the requirements of ASTM-A193. The Owners Group recommended a quality revalidation review (material verification) on a sample basis to confirm capscrew material properties. The Owners Group further recommended that all materials used in the "straight shank" design, if not AISI 4140,^{2/} have a minimum proof strength of 90,000 psi.

23. SwRI concludes that either capscrew design is satisfactory for use at PNPP. Neither will fail due to fatigue loading, thermal stresses, or creep, when properly torqued. While the modified capscrew design is slightly more resistant to fatigue failure, the difference is slight (15% higher), and the original design, with a safety factor of 1.46, has more than adequate fatigue resistance. Both capscrew designs will

^{2/} AISI 4140 and ASMT-A193 are comparable in their requirements for chemical composition.

fail with a loss of preload. However, loss of preload will not occur with proper maintenance. The subcover threads will not strip with the thread-engagement length provided, and with the capscrew properly torqued.

24. Because of the acceptability of the original design and the fact that there is only a slight improvement associated with the modified design, SwRI does not recommend a change to the "necked shank" design at PNPP. SwRI recommends that the capscrews be retorqued periodically during initial engine operation until no movement is detected. Torque should be checked at every outage after this initial period in accordance with the maintenance recommendations provided by the Owners Group for the PNPP engines. SwRI further recommends that during capscrew installation, all mating surfaces be cleaned, and the capscrew threads lubricated with a 50/50 oil/graphite mixture as specified by TDI.

B. Fuel Oil Injector Tubing

25. The fuel oil injector tubing on the TDI standby diesel generators transfers high pressure fuel from the individual cylinder fuel pumps to the injection nozzles. The fuel oil injector tubing was included among the Phase I components due to leaks in the low pressure tubing and in one high pressure circuit. In drafting the design review task description for this

component, the Owners Group determined that the critical functional attributes were: 1) that the tubing have adequate fatigue strength to withstand cyclic pressure and vibration without failure; 2) that it be resistant to internal corrosion and erosion; and 3) that the tubing connectors be able to withstand the same conditions as the tubing. Both shrouded and unshrouded tubing designs were reviewed by the Owners Group. PNPP currently has the unshrouded tubing design, however, shrouded tubing is scheduled to be installed.

26. The Owners Group performed an analysis to determine applied stresses due to the pulsating fuel pressure.^{3/} A comparison of these stresses to the tubing material yield-strength and endurance limits was done. A fracture mechanics analysis was conducted to determine the maximum inner diameter flaw size that would not propagate to failure. Test procedures were evaluated to determine those which would detect unacceptable flaw sizes. A separate evaluation of the connectors and of the corrosion and erosion resistance of the inner surface tubing was also performed.

^{3/} A detailed discussion of the Owners Group review of this component is contained in: "Emergency Diesel Generator Fuel Oil Injector Tubing," April, 1984, prepared by Stone & Webster Engineering Corporation.

27. The Owners Group also reviewed the TDI endurance test procedures and test results. The test report from the connector supplier (Bendix) was also reviewed. This report included tests demonstrating that cavitation and erosion were not a problem at the flow levels required in the TDI engines. In addition to these manufacturers' tests, several utility engines with TDI tubing, have accumulated more than 750 hours (10^7 cycles) of operation. This provides confirmation that fatigue failure of the fuel oil injector tubing will not be a problem.

28. Based on its analyses, the Owners Group determined that the fuel oil injector tubing meets the stress criteria of ASME III Class 2 piping design which is that the maximum allowable stresses be less than or equal to the smaller of either the tubing material tensile strength (12.3 ksi) or 62.5% of the tubing material yield strength (15.6 ksi). It concluded that fatigue failures would not be a concern with the maximum engine cyclic fuel pressure, which results in a maximum tubing stress of 11.6 ksi (which is less than the 12.3 ksi allowed). It also concluded that engine and seismic vibration loads are not a concern.

29. The fracture mechanics analysis concluded that a 0.0048 inch deep maximum flaw size could be contained on the inner surface of the tube and not propagate. Because the

testing method (eddy current) used to detect cracks has a resolution of ± 0.0005 inches, the actual allowable flaw size detected by this method will not exceed 0.004 inch deep.

30. The compression fittings used to connect the tubing to the pump and injector were considered satisfactory, if installed correctly, given their testing and in-service performance. Proper installation instructions are provided in the TDI maintenance manual and are utilized at PNPP.

31. SwRI reviewed the Owners Group report on the fuel oil injector tubing to verify its applicability to the PNPP engines. The assumptions and methods of analysis used by the Owners Group were acceptable to SwRI and no other analyses were necessary.

32. The Owners Group recommended inspection of the inner surface of the high pressure tubing using eddy current techniques and rejection of any tubing that exhibits a flaw size greater than 0.004 inch deep. This inspection has been conducted on the PNPP engines. No flaws were found.

33. The Owners Group report and maintenance instructions recommend that the tubing and fittings be checked visually each month for fuel oil leaks while the engines are operating. Also, the tubing supports should be checked at each outage to

assure that the elastomer inserts are functional and to check for any excessive fuel oil line vibration and deflection. SwRI agrees with each of these recommendations. Although not an Owners Group requirement, PNPP will be installing shroud lines around the fuel oil injection tubing as an added precaution.

C. Main Bearing Saddle, Bearing Caps, and Fasteners

34. The main bearing saddle, bearing caps, and the fastener assembly support the engine crankshaft in the TDI standby diesel generators. The cylinder firing pressure exerts a load on the engine piston which is transmitted to the main bearing saddle assembly through the connecting rod and the crankshaft. Therefore, the entire load on the piston is supported by the saddle-bearing cap assembly.

35. In formulating the design review task description for this component, the Owners Group determined that the critical functional attribute of the saddle-bearing cap assembly was that it have sufficient strength to carry the lateral loads imposed by crankshaft inertia, and to react to the vertical compression and tension loads imposed by the engine firing and the crank/rod/piston inertia loading. The casting nut pockets for both main bearing studs and through-bolts must have sufficient strength to carry the nut preload, inertia and dynamic loading from the crankshaft and firing loads transmitted from the upper

engine. The studs, bolts, and nuts (fasteners) must have sufficient strength to carry the imposed preloads, dynamic loads, and firing loads. The clamping force provided by the main bearing studs and nuts must be sufficient to prevent lateral movement of the bearing caps under lateral crankshaft loading. The main bearing caps must have sufficient strength to withstand imposed crankshaft loads and the base must be sufficiently rigid to maintain adequate main bearing alignment during operation.

36. The saddle-bearing cap assembly was included among the sixteen generic concerns due to cracks observed in the bearing pedestals of DSR-4 inline engines, a nut pocket failure in a DSRV-16-4 engine, and through-bolt failures on a DSR-46 engine. The cracks in the bearing pedestals were attributed to an improper engine disassembly method; the nut pocket failure was found to be due to impurities in the casting material; and the through-bolt failures were due to insufficient initial torque and accompanying preload.

37. The Owners Group performed a stress analysis for both the bearing saddle and bearing caps.^{4/} A fatigue and fracture

^{4/} A detailed discussion of analyses performed is included in "Design Review of Engine Base and Bearing Caps for Transamerica Delaval DSRV-16 Diesel Engines," prepared by Failure Analysis Associates ("FaAA") for the Owners Group.

analysis was also performed for the bearing saddle. A "hole in a plate" model was used for the saddle. The loading consisted of main bearing loads derived from a journal orbit analysis, and load due to interference fit between the saddle and the main bearing. To represent the actual loading condition, the base and cap were modeled as an elastic foundation. Variation in the interference load due to differential expansion between the cast iron saddle-bearing cap assembly and aluminum bearings was considered.

38. The fatigue and fracture analysis performed to determine the fatigue life of the bearing saddle utilized maximum tensile stress and modified Goodman line failure criteria to assess the fatigue strength. Finish, size, temperature, and other parameters were considered to calculate the maximum alternating value of 7,960 psi and a factor of safety of 1.31. The Owners Group concluded that an adequate margin of safety against fatigue failure existed.

39. A similar model and similar procedure was used for the stress analysis of the bearing caps to calculate the alternating (17,700 psi) and the maximum positive principal stresses. The maximum stress of 10,030 psi was found to be well below the 60,000 psi yield-strength of the cap material. Based on a safety factor of 3.35, the Owners Group also

concluded that fatigue failure would not occur in the bearing cap.

40. For the through-bolts and nuts, the bolt, being the weakest of the two components, was reviewed by the Owners Group. The Owners Group approach was very conservative in that it assumed a very low value for the coefficient of friction. Stresses were calculated assuming axial loading of the bolts. Finite element analysis was used to determine the portion of the firing load carried by each bolt. The fatigue factor of safety for infinite life was calculated using the modified Goodman line failure criteria. Based on a factor of safety of 1.45, the Owners Group concluded that fatigue failure of the bolting would not occur if the preload torque specifications are met. To ensure that the cap would not move in the horizontal direction, the available friction force of the joint was compared with the lateral load of 218,000 lbs. The friction force of 389,000 lbs was found to be 1.78 times higher than the lateral load of 218,000 lbs. It was concluded that lateral movement of the joint would not occur.

41. The Owners Group also analyzed the bearing cap and nut fastener system. This system consists of top nut, bearing stud, bottom nut, bearing cap and pedestal. Strength of the nut, fastener behavior under preload, crankshaft vertical load,

and safety against fatigue failure of the stud were analyzed. The friction force at the joint and the crankshaft horizontal forces were compared to ensure that there would be no horizontal displacement of the bearing cap. Modified Goodman criteria was used to calculate the fatigue strength. Appropriate modifiers were used to calculate the material fatigue limit. From the factor of safety obtained (1.30), it was concluded that the fastener system is safe from fatigue failure. The friction force of 94,400 lbs was found to be 1.19 times the crankshaft horizontal force of 79,400 lbs. It was concluded that friction force is adequate to resist the cap movement.

42. SwRI reviewed the Owners Group report on the saddle-bearing cap assembly to verify its applicability to the DSRV-16-4 engines at PNPP. SwRI verified the accuracy and applicability of all of the assumptions made in the foregoing Owners Group analyses. The bolt loading values, material specifications, and material strength used for the analyses were appropriate and were used for SwRI's analysis.

43. One significant difference between the results obtained in the Owners Group analysis and that conducted by SwRI was observed. SwRI's analysis indicated that the interface between the cap and the saddle is much stronger than that calculated by the Owners Group. A stronger interface

means that there would be no lateral movement of the bearing cap under the influence of the crankshaft horizontal force. The difference occurred because SwRI's analysis also considered the presence of the dowel between the saddle and the bearing cap. It was concluded that the dowel must fail before any lateral movement of the bearing cap. Given the dowel's factors of safety of 7.0 in shear and 2.38 in bending, it was concluded that no lateral movement of the bearing caps will occur.

44. The Owners Group concluded that the saddle-bearing cap assembly is adequate for nuclear service. SwRI also concludes that the assembly is satisfactory for use at PNPP. In all cases the safety factors calculated are adequate to conclude that the saddle-bearing cap assembly has infinite life against fatigue failure.

45. To increase the factor of safety between the saddle and the bearing cap, the Owners Group recommended that the mating surfaces of the base and cap be thoroughly cleaned with solvent before first assembly and upon any reassembly as a precautionary measure to improve the frictional force resisting cap movement. SwRI recommends that the preload torque of both the bearing cap studs and the through-bolts be checked prior to engine operation. If the preload torque is less than that specified by TDI, all the studs should be retorqued.

46. PNPP has conducted a torque verification of the bolts and nuts. Proper maintenance procedures for engine disassembly have also been followed at PNPP.

D. Connecting Rod Bearing Shell

47. The babbitt-overlaid cast aluminum connecting rod bearing shells in the TDI standby diesel generator engines provide the oscillating/sliding surface between the crankpin and the connecting rod through formation of a hydrodynamic film. The cylinder firing pressure is transmitted through the connecting rod bearing shells to the crankpin. The force is thus converted into engine torque. In formulating the design review task description for this component, the Owners Group determined that the critical functional attributes of the bearing shells were that: 1) they have sufficient fatigue life and wear resistance to withstand the cylinder firing pressures; 2) that the bearing material be low friction and have corrosion-resistant properties; and 3) that the bearing design be such that during operation the peak oil film pressure, minimum oil film thickness and oil film temperature rise be within acceptable limits.

48. The connecting rod bearing shells were included among the sixteen Phase I components due to failures of the connecting rod upper shells in the Shoreham Nuclear Power Station

DSR-48 engines. Failures occurred due to cracks in the bearing shells after about 250 hours of full-load operation due to the large (1/4" x 45 degree) chamfer on the connecting rod bearing cap and bearing shells.^{5/}

49. A visual non-destructive examination of the bearing shell from a utility engine with 315 hours of full-load operation was performed in the Owners Group Phase I investigation. This particular engine was a DSRV-16-4 model with a 13-inch x 13-inch crankshaft, the same as on the PNPP engines. Dye penetrant inspection and measurement of wall thickness was part of the non-destructive examination. In only a few cases was edge loading, light to moderate scoring of overlay, and overlay removal evident. None of these had any effect on the bearing's ability to perform its intended function.

50. Chemical composition tests and tests for tensile properties of typical failure bearing shells from the Shoreham crankshaft were also performed. The material composition and the tensile properties were found to be in accordance with the TDI material specifications. It was concluded that the material was suitable for its intended application.

^{5/} The PNPP engines have the smaller (1/16" x 45 degree) chamfers.

51. Journal orbit analysis was used to determine the characteristics of the hydrodynamic oil film formed between the bearing and the crankpin. The peak oil film pressure in the DSRV-16-4 engine was found to be 25,800 psi.

52. Finite element stress analysis was performed to compare the magnitude, orientation and location of the largest tensile stress in the bearing shells. The maximum tensile stress in the bearing shells was found to occur at the inside diameter and had a value of 8,200 psi. Using the results of the finite element stress analysis, a fatigue analysis was performed. The fatigue life of the bearing shells in the DSRV-16-4 engine was estimated to be 38,000 hours at full-load operation. The finite element and fatigue analysis applied only to the fully-supported bearing shells which the PNPP engines have.^{6/}

53. Using a fracture mechanic analysis, the stress intensity factor range was used to compute an acceptance criterion for bearing shell castings of 0.050 inch (maximum discontinuity size).

^{6/} In the fully-supported assemblies, the chamfer in the connecting bore is 1/16" x 45 degrees. The fully-supported bearings eliminate the possibility of edge loading on the bearing shells.

54. The Owners Group analysis concluded that the connecting rod bearing shells used on DSRV-16-4 engines, with 13-inch x 13-inch crankshafts and connecting rod bores with 1/16" x 45 degrees chamfer (which fully support the bearing shell) have a fatigue life of 38,000 hours of full-load engine operation. This conclusion is verified by the operating history of the engines.7/

55. SwRI reviewed the Owners Group report on the connecting rod bearing shells to verify its applicability to the TDI diesel engines at PNPP.8/ Design parameters used by SwRI were in agreement with those used by the Owners Group. The assumption made by the Owners Group with regard to firing pressure was lower than that utilized by SwRI for PNPP, as discussed below.9/ Other assumptions were consistent.

56. SwRI performed an additional analysis to calculate the oil film thickness and oil film pressure accompanying an

7/ For example, no failure of bearing shells was observed after 1200 hours of operation of a Grand Gulf DSRV-16-4 engine.

8/ A detailed discussion of the Owners Group analysis is included in the FaAA report, "Design Review of Connecting Rod Bearing Shells for Transamerica Delaval Enterprise Engines," March, 1984.

9/ The SwRI approach would not have changed the conclusion of the Owners Group study that the connecting rod bearing shells are suitable for the PNPP engines and are safe from fatigue failure.

increased gas pressure. These calculations were performed by the Owners Group for a peak cylinder firing pressure of 1450 psig. The PNPP engine data indicated that the peak cylinder firing pressure can be as high as 1700 psig. For this higher cylinder pressure the minimum film pressure for the highest load was calculated by SwRI to be 23,314 psi, which is lower than the allowable pressure of 26,000 psi. It was concluded, therefore, that the increased peak cylinder firing pressure would not have a detrimental effect on the fatigue life of the bearing shells.

57. SwRI agreed with the Owners Group conclusion that the PNPP bearing shells have a fatigue life of at least 38,000 hours of full-load operation.

58. Based on the results of its analyses, the Owners Group concluded that the babbitt-overlayed cast aluminum connecting rod bearing shells were suitable for their intended purpose and that minor surface imperfections in the babbitt-overlay would not degrade the suitability of the bearing shells. Again, this analysis was for fully-supported bearing shells, which the PNPP engines have. The critical zone of the connecting rod bearing shells was determined to be a circular band on each end of the bearing, beginning 0.4 inch from the bearing end and extending 1.4 inches from the bearing end. The

Owners Group analysis resulted in a recommendation that the size of the voids in the casting in this critical zone not exceed 0.050 inch. In the non-critical areas of the bearing shell, and in the lightly-loaded lower bearing shell, the void size can be as large as 0.250 inch.

59. To assure compliance with the acceptance criterion, the Owners Group recommended inspection of the bearing shells by radiography. PNPP has performed this inspection on its engines. All bearing shells presently installed in the engines meet the Owners Group criterion.

60. SwRI recommends that both the connecting rods and bearing shells be inspected to ensure that the chamfer on both sides of both of the parts does not exceed 0.062 inches x 45 degrees. PNPP has performed this inspection on its engines and determined that its engines have this smaller chamfer.

E. Emergency Diesel Generator Engine and Auxiliary
Module Wiring and Termination Qualification to
IEEE-383-1974

61. The TDI diesel generator engine and auxiliary module wiring and termination interconnect instrument, control, and power circuits on the diesel generator itself and within the control panels. In drafting the design review task description for this component, the Owners Group determined that the

critical functional attributes of the wiring and termination were that: 1) the conductors, insulation, and termination must be suitable for the specified amp rating; 2) the conductors and insulation must be flame retardant; and 3) the material and insulation rating must be appropriate for engine and generator applications.

62. The wiring and termination was included among the sixteen Phase I components because of a Service Information Memorandum ("SIM") issued by TDI informing utilites of two potentially defective engine-mounted cables that did not meet IEEE-383-1974. CEI has implemented the recommendations of this SIM and replaced its cables with cables meeting IEEE-383-1974 standards.

63. The Owners Group performed an analysis of both TDI-generic and PNPP-specific wiring and termination designs.^{10/} This analysis included a review of the circuit requirements. Included in this analysis was a determination of the wire insulation rating, type and rating of termination, voltage, maximum temperature, current flame retardancy requirements (IEEE-383-1974), and routing.

^{10/} A detailed discussion of this analysis is included in Stone & Webster Engineering Corporation, "Emergency Diesel Generator Auxiliary Module Control Wiring & Termination Qualification Review," July 1984, prepared for the Owners Group.

64. Satisfactory performance of the wiring and termination has been demonstrated in several utility engines with more than 750 hours of operation and over 100 starts each. This operating history confirms the reliability of the TDI-generic wiring and termination. Having implemented the SIM, PNPP now meets all of the requirements of the Owners Group review.

65. SwRI reviewed the Owners Group report on wiring and termination to verify its applicability to the PNPP diesel generators. The assumptions and methods of analysis used by the Owners Group were acceptable to SwRI.

66. Based on the results of its analysis, the Owners Group concluded that the PNPP wiring and termination were satisfactory. The Owners Group found that all wiring and termination was adequate to meet voltage and current requirements at the elevated engine block (82.2°C) or room ambient temperatures (60°C).^{11/} The Owners Group also found that the wiring met flame retardant requirements of IEEE 383-1974, that routing was acceptable and that shielded cable was provided where required.

67. SwRI agrees with the findings of the Owners Group and concurs with actions taken by PNPP, which included the conduct

^{11/} These findings are conservative for PNPP since PNPP's FSAR indicates that maximum room temperature will only be 49°C.

of an inspection and the implementation of the recommendations contained in TDI's SIM. No additional action is recommended.

F. Cylinder Head Stud

68. The cylinder head studs in the TDI standby diesel generators transmit cylinder firing pressure forces from the cylinder heads to the engine block and assure the required preload on the cylinder head gasket for combustion gas and water sealing. In drafting the design review task description for this component, the Owners Group determined that the cylinder head studs must have sufficient strength to withstand the necessary preload and cyclic firing pressure forces without preload relaxation or thread distortion. Further, the thread geometry of the head stud should provide an upper thread engagement which is sufficiently below adjacent cylinder liner landings to minimize stress concentration in this area.

69. The cylinder head studs were included among the Phase I components due to isolated failures resulting from insufficient preload in some non-nuclear applications. Two distinct head stud designs were evaluated by the Owners Group, the original, "straight" shank design, and a modified, "necked" shank type. CEI has replaced the "straight" shank type supplied with the engines at PNPP with the "necked" shank design.

70. A stress analysis was performed by the Owners Group on both head stud designs^{12/} The analysis included a determination of the applied stresses, the endurance limits, a fatigue life analysis, a thread distortion analysis, and a thermal stress evaluation of both stud designs.

71. SwRI reviewed the Owners Group report on the cylinder head studs to verify its applicability to the PNPP cylinder head studs. SwRI performed several analyses beyond those performed by the Owners Group for both stud types to further evaluate design concerns. SwRI investigated the potential for thread-stripping in the block and the effect on the stud if preload is lost.

72. The adequacy of the cylinder head studs to withstand the sum of the total peak combustion pressure force (419,700 lbf.) and the sum of the fire ring gasket and other seal compression forces (356,700 lbf.) was determined using standard engineering design calculations compared to a modified Goodman line failure criteria. This approach permitted both the mean ("straight" design: 46.3 ksi, "necked" design: 45.5 ksi) and alternating ("straight" design: 3.36 ksi, "necked" design: 2.62

^{12/} A detailed discussion of the analysis is included in Stone & Webster Engineering Corporation, "Emergency Diesel Generator Cylinder Head Stud Stress Analysis," April, 1984 and its Supplement, May, 1984, both prepared for the Owners Group.

ksi) stresses to be utilized. The Owners Group fatigue analysis yielded mean ("straight" design: 45.6 ksi, "necked" design: 44.8 ksi) and alternating ("straight" design: 3.8 ksi, "necked" design: 2.9 ksi) stresses as noted.

73. Thread distortion analysis was performed by comparing the principal stresses calculated from the biaxial stress field during nut tightening (to 3600 lbf.-ft) with the failure criteria of the distortion energy theory. Analyses were also conducted by the Owners Group to determine if thermal stresses were a consideration and whether creep would be a factor in causing loss of preload.

74. Satisfactory fatigue life for the cylinder head studs has been demonstrated by several utility engines accumulating more than 750 hours (10^7 cycles) of operation. This operating history verifies that failures due to fatigue, tightening, and loss of preload will not be a problem.

75. A review of results obtained by the Owners Group and those of SwRI did not indicate any significant differences. Based on the slightly different assumptions used by SwRI and the Owners Group, there was a difference in calculated cylinder head stud endurance limits (OG: 10.6 ksi, SwRI: 8.6 ksi) that yielded differences in safety factors for fatigue life of the "straight" shank design (OG: 1.35, SwRI: 1.29) and the "necked"

design (OG: 1.55, SwRI: 1.46). All of these safety factors are considered satisfactory. The safety factors during nut tightening are also satisfactory (OG: 1.2, SwRI: 1.5).

76. SwRI agrees with the Owners Group conclusion that thermal stresses would not be significant because the cylinder head stud and the head are of the same material, have the same coefficients of thermal expansion, and experience similar change in temperatures during engine operation. The temperature differences experienced during engine startup will not contribute significantly to the mean stud stress.

77. With maximum stud temperatures below 190°F and with the maximum operating stress less than half the yield stress, creep will not be a factor in reducing stud preload.

78. Based on these results, the Owners Group concluded that both the "straight" and "necked" studs were satisfactory for nuclear service. The "necked" shank design is somewhat more resistant to fatigue failure and less likely to lose its preload. The "necked" design top thread in the block is lower than that on the "straight" stud, which means that the "necked" design will produce lower stress levels near the liner landing area than the "straight" stud. The Owners Group recommended a quality revalidation review (material verification) on a sample basis to confirm that the cylinder head stud meets the

requirements of AISI 4140. This verification has been performed at PNPP.

79. SwRI concludes that either stud design is satisfactory for the PNPP engines. Neither will fail due to fatigue loading, thermal stresses, or creep when properly torqued. Because the "necked" design is less likely to lose its preload, has a higher safety factor under fatigue loading, and lowers the stresses in the block adjacent to the liner landing area, its adoption is recommended by SwRI. CEI has installed the "necked" design.

80. Both designs will fail due to loss of preload. However, preload will not be lost with proper maintenance (which includes retorquing during initial operation and retorquing during outages). SwRI recommends that the head stud nuts be retorqued periodically during initial engine operation until no movement is detected and thereafter at each outage. SwRI further recommends that care be taken in cleaning and lubricating of the stud and block threads in accordance with TDI specifications to assure maximum preload is attained.

G. Air Start Valve Capscrew

81. The air start valve capscrews in the TDI standby diesel generators provide the clamping force to hold the air start

valves in place on the cylinder heads. In drafting the design review task description for this component, the Owners Group determined that the critical functional attribute of the air start valve capscrews was that they have sufficient strength to withstand the necessary preload and reaction air loading without yielding (and resulting in loss of clamping load on the air start valves).

82. The air start valve capscrews were included among the Phase I components because one nuclear power plant engine was found to have an excessively long capscrew which prevented the air start valve from being properly seated. This was discovered during an inspection prompted by an SIM issued by TDI which recommended that users measure the length of their engine air start valve capscrews because some capscrews supplied in engines were too long.

83. A stress and dimensional analysis was performed by the Owners Group for the air start valve capscrew.^{13/} The analysis included a determination of minimum possible capscrew clearance (given the specified tolerances of the capscrew and its clamped parts), the applied stresses, a fatigue analysis and an analysis of stresses during tightening.

^{13/} A detailed discussion of this analysis is included in the Stone & Webster Engineering Corporation report, "Emergency Diesel Generator Air Start Valve Capscrew Dimensions and Stress Analysis," March 1984, prepared for the Owners Group.

84. Satisfactory fatigue and operational life has been demonstrated by several utility engines which have accumulated more than 750 hours (10^7 cycles) of operation and over 100 starts. This operating history verifies that capscrew fatigue and operational failures will not be a problem.

85. SwRI reviewed the Owners Group report to verify its applicability to the air start valve capscrews used in the engines at PNPP. Design parameters and assumptions used by SwRI were in agreement with those used by the Owners Group.

86. SwRI performed a number of analyses to evaluate the functional attributes of the capscrews. The maximum cyclic load on the capscrews was calculated at 3600 lbf. The adequacy of the capscrews to withstand this loading and the preload was determined using a series of standard engineering calculations culminating in the use of the modified Goodman line failure criteria which considers both mean (38.7 ksi) and alternating (2.7 ksi) stresses. The maximum biaxial stress was calculated for the capscrew during tightening to the specified torque (150 lbf.-ft). The principal stresses were compared to the failure criteria of the distortion energy theory. An analysis to determine the impact of the loss of preload on the capscrew was also performed.

87. No significant difference in results between the Owners Group analysis and that of SwRI was observed. There were slight differences in mean stresses (OG: 37.5 ksi, SwRI: 38.7 ksi) and alternating stresses (OG: 1.6 ksi, SwRI: 2.7 ksi) that yielded different safety factors for fatigue life (OG: 2.04, SwRI: 1.6). Both of these safety factors are satisfactory.

88. The safety factors during capscrew tightening were also considered satisfactory (OG: 1.52, SwRI: 1.88). The safety factor with loss of preload, determined by SwRI, is 1.58. The Owners Group concluded that with capscrews meeting current TDI specifications, a minimum of 0.2 inches clearance exists with the worst case tolerance stack-up between the capscrew and the bottom of its tapped hole in the cylinder head. SwRI agrees with this calculation.

89. Based on these results the Owners Group concluded that the air start valve capscrew was satisfactory for nuclear service. The design provides adequate safety factors against fatigue failure and failure while tightening.

90. SwRI concludes, likewise, that the air start valve capscrew is of adequate design to resist failure by fatigue and during tightening. The capscrew will not fail due to loss of preload.

91. SwRI recommends retorquing at eight-hour intervals during initial engine operation, as specified by TDI and the Owners Group, until no further movement is detected. This retorquing procedure will assure retention of a tight seal as the copper gasket "sets". SwRI also recommends that care be taken in cleaning and lubricating the capscrew and head threads in accordance with TDI instructions, to assure that maximum preload is attained.

H. Push Rods

92. The intake and exhaust main push rods and the exhaust connecting (intermediate) push rods are among the sixteen Phase I components. The primary function of the push rods is to provide a portion of the linkage that transmits camshaft lobe motion to the intake and exhaust valves, thereby controlling the valve opening and closing cycle. Important functional attributes of the push rods are: 1) that the push rods have sufficient column strength to withstand compressive buckling; 2) that the ends have acceptable wear resistance; and 3) that the push rod tube and end fitting design minimize the possibility of design defects and be able to function without separation of the tube and end fittings.

93. Three basic designs are in use. These are: 1) forged head; 2) ball end; and 3) friction-welded end. The forged head

design consists of a tubular steel shaft fitted with hardened steel end pieces attached to the tube with four plug welds near the end of the tube. Cracking in the tube wall adjacent to the plug welds has been found in the forged head design. Concern about the shear capacity of the plug welds has also been raised.

94. The ball end design consists of a tubular steel shaft fillet-welded to a carbon steel ball at each end. The ball end design has exhibited cracking at the interface of the push rod tube and ball end fitting. These cracks have developed in the heat-affected zone of the high carbon chrome steel ball. Complete separation between the tubular steel shaft and the push rod ball has occurred upon disassembly. Failures of the forged head and ball end designs have not affected engine operability because the valve train lash has never been sufficient to allow the push rod end to escape from the tube.

95. The friction-welded design (the design used at PNPP) consists of a tubular steel shaft friction-welded to a solid steel plug on each end. No failures have been reported for the friction-welded push rods.

96. The Owners Group investigation of the push rods consisted of a metallurgical analysis of the fillet-welded ball end design, fatigue analyses and an analysis of the buckling

stability and wear resistance of the forged head and friction-welded designs.^{14/}

97. SwRI's review of the Owners Group effort consisted of a review of the push rod criticality status, a review of the push rod static and dynamic loading and a comparison of this loading with calculated critical buckling loads (i.e., the loads that the push rods could withstand without buckling). SwRI's analysis of critical buckling loads included the well-known Euler and short column equations used by the Owners Group. SwRI extended the analysis to include buckling under dynamic loading conditions.

98. The Owners Group performed a two-part experimental evaluation. The first part consisted of fatigue tests of forged head and friction-welded push rods. The fatigue tests were performed at loads in excess of those expected in service to a life of 10^7 cycles. The second portion of the experimental evaluation consisted of metallurgical examinations of forged head and friction-welded push rods for a comparison with specifications identified in the drawings. The metallurgical examination included: 1) a dimensional check of the push rod

^{14/} A description of the Owners Group review is contained in, "Design Review of Push Rods for Transamerica Delaval Diesel Generators", April, 1984, prepared by FaAA.

with the drawings; 2) a chemical evaluation of the ends and the tubular shaft; and 3) a microhardness profile of the ends and the tubular shaft.

99. The Owners Group analysis of push rod loading showed a combined static and dynamic maximum loading of 2,945 lbs. for the intake push rod, 7,787 lbs. for the exhaust main push rod and 13,040 lbs. for the exhaust intermediate push rod. Calculated critical loading at which buckling would be expected is lowest using the short column equation. For the main push rod the critical loading is 18,100 lbs. This gives safety factors of 6.1 and 2.3 for the intake and exhaust push rods respectively. For the exhaust intermediate push rod, the critical loading is 27,560 lbs. which provides a safety factor of 2.1.

100. The Owners Group fatigue crack analysis showed that, under cyclic loading, no potential fabrication crack is expected to propagate in either the main or intermediate friction-welded push rods. This conclusion was based on a very conservative model that assumed some net tension during cyclic loading. In reality, the push rods will always be loaded in compression which assures that fatigue crack growth will not occur.

101. The fatigue test performed by the Owners Group on the friction-welded intermediate push rod used a load cycle ranging

from zero to 1.25 times the maximum loading. Radiographic and fluorescent magnetic particle examinations performed before and after the tests showed no flaws. The end connections were sectioned and examined after the test. Neither flaws nor indications were found.

102. The results of the Owners Group metallurgical evaluation of the friction-welded design showed the push rod dimensions to be within drawing specifications. Chemical analyses showed that the chemistries fall within the material requirements specified on the drawings. The hardness values on the spherical portion of the end plug were in agreement with drawing specifications. Results of a metallurgical evaluation of the friction-welded design showed microstructures typical for the specified materials. No major discrepancies with the material properties specified on the drawings were observed.

103. SWRI's review of the push rod analysis performed by the Owners Group produced only minor areas of difference. SWRI's calculated values for dynamic loading were slightly higher (15% for intermediate rod and 8% for the main rod). This difference, when combined with static loading, yielded a combined increase of only 0.6% for the main and intermediate push rods, which still results in an acceptable safety factor.

104. SwRI's extension of the buckling analysis to include an evaluation of dynamic buckling showed that dynamic loading would have no effect on buckling.

105. From the analysis performed, the Owners Group has concluded that the friction-welded design is the most reliable of the three push rod designs. SwRI concurs with this selection of the friction-welded design for the PNPP engines.

106. The Owners Group recommended an inspection and sampling plan be developed to meet the following criteria:

- a) No surface cracks longer than 25% of the circumference (approximately one inch) should be allowed along the bond joint between the rod end and the tube. Those rods where the friction weld looks suspicious (i.e., obvious failure to weld, porous weld area, etc.) should be removed.
- b) Appropriate destructive testing techniques should be employed on a random sample of friction-welded design push rods to examine the interior section for lack of fusion which is identifiable by lack of "lips" on the interior of the tube.

107. The Owners Group report also states that any friction-welded push rods successfully completing 800 hours of in-service use at full load may be considered qualified for continued use.

108. Because push rod failure could result in an early engine shutdown, SwRI agrees with the recommendations of the Owners Group that an inspection and sampling plan should be implemented and applied to all push rods. An inspection of all of the push rods was performed prior to installing them in the PNPP engines to verify that they were of the friction-welded design.

I. Cylinder Heads

109. The cylinder heads in the TDI standby diesel generators provide a pressure-tight cap for the engine cylinders and provide passages and sealing for cooling water, lube oil, starting air, intake, and exhaust gases. In formulating the design review task description for this component, the Owners Group determined that the critical functional attributes of a cylinder head are that it must: 1) serve as a structural member with sufficient stiffness to react to the cylinder firing forces without leakage or damaging deformation; 2) maintain stresses below endurance limits; 3) withstand thermal and mechanical fatigue loading; and 4) be resistant to impact and corrosion damage.

110. The cylinder heads were included among the sixteen Phase I components because of head defects consisting of cracks in locations such as the fire deck, the exhaust and intake bridges, exhaust valve seats, and induction port. Cylinder heads cast before September 1980 were subject to core shift, inadequate control of solidification, and inadequate control of the stellite valve seat weld deposition process. Heads cast before October 1978 were not stress-relieved and were therefore subject to fatigue crack growth in thin sections and/or from fabrication-induced defects. Heads on the PNPP engines were cast prior to October 1978.

111. The cylinder heads at PNPP were removed and returned to TDI where they were inspected for cracks in thin sections. Several were rejected. Heads not rejected were given a stress-relief treatment before being reinstalled. All replacement heads were cast after September 1980. The inspection of and modification to the current cylinder heads installed at PNPP make them equivalent to those manufactured after September 1980.

112. A metallurgical analysis was performed by the Owners Group.^{15/} The analysis included: 1) examination of casting

^{15/} A detailed discussion of the analyses performed is included in FaAA report "Evaluation of Cylinder Heads of Transamerica Delaval, Inc. Series R-4 Diesel Engines," May 1984, prepared for the Owners Group.

shrinkage indications found in a cylinder head taken from a Comanche Peak engine; 2) examination of a pre-existing shrinkage void and of a crack in a stellite weld deposit, each found in a Grand Gulf head; and 3) the examination of a crack across the wall of the fuel injector port in a head from an engine at the Catawba Nuclear Power Station. A review by SWRI of the Owners Group metallurgical analysis confirmed that the analysis was performed using recognized and correct procedures.

113. An evaluation of thermal and pressure stresses was also performed by the Owners Group. This analysis included an initial determination of the transient and steady state temperature distribution in the fire deck, and then a thermal stress analysis and a pressure stress analysis. The calculations performed by the Owners Group for thermal and pressure stresses were reviewed for validity of assumptions and calculations.

114. Modeling of the cylinder head to fit existing well-established equations was performed by the Owners Group using the following assumptions:

- a) The fire deck is subjected to steady state mean stresses resulting from clamping the head by the studs and from thermal gradients.

- b) The fire deck is loaded in bending by peak gas pressure.
- c) The fire deck is idealized: 1) as circular flat plate with clamped outer boundary; 2) two parallel decks constrained to deflect together; and 3) a flat plate clamped at its boundaries having an average thickness of 3/4 inch.

SwRI used a more conservative approach, assuming that the fire deck was two parallel decks tied together at a number of locations.

115. Analytical results obtained by SwRI and the Owners Group differed only slightly due to the difference in assumptions made regarding the fire deck. The Owners Group analysis assumed the fire deck to be a flat plate, clamped at its edges. This model produced a bending stress of 115 ksi. The assumption that the deck was supported at the center resulted in calculations indicating that the central support would reduce the stress by one-half (i.e., to 58 ksi). SwRI's analysis showed that, using its assumption, the bending stress at the boundary would be more than one half that of a single plate. However, as the number of contact points increase (e.g., intake and exhaust ports, the injector port and water passages), the stress in the fire deck would approach one-half, as concluded by the Owners Group.

116. The Owners Group thermal stress analysis, reviewed by SwRI, showed that the contribution of thermal stress to high cycle fatigue is quite small (a 5.0 ksi increase in mean stress). The most significant thermal stress is the bending stress resulting from the temperature gradient in the deck thickness. Calculations predict a range of bending stresses of about 20 ksi between the exhaust valve ports and 30 ksi between intake valve ports, both below the yield stress of the material.

117. Pressure stress analysis showed a maximum bending of 58 ksi due to pressure loads with a mean and amplitude stress of 29 ksi. The Owners Group calculations for combined pressure and thermal stress showed a maximum of 28 ksi and a minimum stress of -30 ksi, both below the yield strength of the material.

118. Analysis of the head design by the Owners Group showed that stresses resulting from combustion pressure loads can exceed the material yield stress. However, the combined stresses resulting from pressure and thermal loading are reduced to a safe level because the thermal and gas pressure bending stresses partially cancel out. SwRI concurs that combined stresses will have a cancelling effect during steady-state running. During an emergency start, several seconds are

required for thermal loading to develop. In this time period, the pressure loading may approach its design value and consequently the full benefit of combining thermal and pressure stresses may not be realized. These stress calculations were made using a number of assumptions to render a very complex thermal and structural problem tractable to hand solution. A consequence of the numerous assumptions is that the computed stresses can only be considered as an approximation of the stress rather than actual values. The analysis does serve to identify an item for which periodic inspections are necessary. No field failures of this kind were documented in the Owners Group component database or the Owners Group report (which include the review of several utility engines with more than 100 starts and 750 hours (10^7 cycles) of operation).

119. The inspection, replacement of some heads, and additional stress-relief conducted at PNPP for the cylinder heads assure that they will be adequate for service. This inspection will be supplemented by periodic checks for water leaks, according to the preventative maintenance recommendations contained in the Perry Maintenance Matrix provided by the Owners Group in the PNPP DR/QR Report. A barring-over or blow-over procedure will detect water leakage that could occur if head cracks develop. Early detection of leaks will negate any harmful effects on the engines.

J. Piston Skirt

120. The piston skirt in the TDI diesel generators transmits the cylinder gas pressure force on the piston crown to the connecting rod. Also, the piston skirt supports the piston crown and guides the connecting rod into the engine's cylinder. The side thrust developed due to the obliquity of the rod is transferred to the piston skirt. The piston skirt provides a sliding friction surface against the stationary cylinder liner. In the TDI diesel generators, the piston crown is bolted to the piston skirt. The skirt is, therefore, required to carry the bolt preload. In preparing the design review task description for this component, the Owners Group considered the important functional attribute of the piston skirt to be that it have sufficient strength to withstand cyclic loading without fatigue cracking.

121. The piston skirts were included among the sixteen Phase I components due to failures indicated by cracks in the skirt-to-crown stud attachment bosses in 23 out of 24 pistons in a DSR-48 engine at the Shoreham Nuclear Power Station. Each of the failed piston skirts were of the AF design. The one piston skirt which did not show any sign of failure was of the AN design.

122. The Owners Group performed a detailed analysis of the AF and AE type piston skirts. Basically, the AF and AE piston skirts are similar in design with identical loading and functional attributes. The major difference between the two designs is in their fabrication thermal history and the configuration of the stud bosses inside the skirt where the washers on the crown attachment stud meet the skirt. Because of the design similarities, the Owners Group analysis used the data/test results in an interchangeable manner whenever applicable. Originally, the piston skirts used on the PNPP DSRV-16-4 engines were of the AH type. These piston skirts have since been replaced with the much superior AE design.

123. In the Owners Group analysis, a non-destructive dye-penetrant test was performed on the failed AF piston.^{16/} The analysis found that all of the cracks were similarly located and oriented on the spot-faced boss. Eddy current tests were used to confirm the presence of the cracks. A destructive examination was also conducted and the cracks were opened up. Examination of the fractured surfaces indicated that they were fatigue cracks. This was confirmed by examination in a scanning electron microscope.

^{16/} A discussion of the analysis performed by FaAA for the Owners Group is contained in "Investigation of Types AF and AE Piston Skirts," May 23, 1984.

124. Small specimens from the area near the fatigue-cracked bosses were properly examined to reveal microstructure. Due to the difference in the heat treatments/cooling rates, the microstructures for AF and AE piston skirts were different. However, the differences were found to be consistent with the difference in the heat treatment of the two piston skirts. The amount of ferrite in the AE piston skirts was only half that of the AF piston skirts.^{17/} A sample of the piston skirt material was subjected to chemical analyses. The chemical composition of the test specimen was found to be as per the design specifications (ductile iron ASTM-8536 grade 100-70-03). Material hardness and the mechanical properties of the specimen were also found to be in agreement with the design specifications and the heat treatment. Using a dissection technique, residual stress measurements were made on both AF and AE piston skirts. The residual stress near the ridges and boss region on the AF piston skirts was found to be over 11 times that of the AE piston skirts. The most undesirable residual stresses in AE piston skirt material were low due to the increased cooling rate imposed after normalizing the AE skirts.

^{17/} The lower ferrite makes the AE piston skirts structurally stronger.

125. Experimental stress analysis was performed on both AF and AE piston skirts. A stress coat technique was used to identify regions of peak stress. These regions were then concentrated upon in the strain gauge tests. Appropriate test set-up was used to simulate the engine loading on the piston crown. The test pressure of 2,000 psig applied on the piston crown was well above peak combustion gas working pressure on the piston crown. The strain gauges on the peak stress area of the AE piston skirts provided information about the pressure required to close the gap between the outer ring and the crown. This information was then used in another set of experiments in which the skirt loading was maximized. During the warming up period, the loading on the skirt is the highest, when there is a gap between the skirt and the crown's outer ring. Calculations were performed to calculate the stresses in the high stress region. The stress at 1,627 psig working pressure was estimated to be -68.7 ksi. Peak stresses in AE skirts were found to be generally lower than corresponding AF values. In addition to gas loading, stresses due to bolt preload were evaluated by finite element model and measured on AE piston skirts. Since the bolt preload was found to have no affect on the stresses in the critical area of the stud boss region, it was omitted from the overall finite element analysis of the skirt.

126. Stresses and displacement under gas pressure and inertial loads were calculated for the AE piston skirt using ANSYS finite element computer program. Two models, a full model of the skirt and a local model of the crown stud boss, were prepared. In addition to these skirt models, a finite element model of the crown was developed for placing on the global skirt model. The Owners Group developed additional local models to better evaluate the magnitude and location of the peak stresses in the stress gradient wherever required.

127. Interaction between the crown and skirt was analyzed with the help of the models and the information obtained through the experimental stress analysis. The analysis indicated that the gap between the skirt and the crown closes uniformly. Uniform gap closing means an even distribution of load and therefore uniform stresses throughout the load-bearing areas of the piston skirt.

128. The peak stress magnitudes computed by finite element analysis were compared with the experimental results. The Owners Group analysis found generally good agreement between the experimental and finite element results. The average experimental values on AE skirts were 28 times below the finite element values. The differences were attributed to assumptions such as a rigid wrist pin in finite element analysis. The

experimental and finite element stress analysis results were then appropriately used with the fatigue and fracture properties of the material to analyze the possibility of crack initiation in the piston skirts. A modified Goodman diagram was plotted for two different yield strengths (measured and specified). The Owners Group finite element analysis predicted that although fatigue cracks may possibly initiate in the stud boss region of the AE piston skirt, these cracks will not necessarily grow, because they would propagate into a region of decreasing stresses. The experimental results, however, predicted that cracks will not initiate in AE piston skirts.

129. A crack propagation analysis was undertaken to analyze whether the cracks that are predicted (by finite element analysis) to possibly initiate in the stud boss region of AE piston skirts could substantially grow. Crack growth calculations were performed using the cyclic stresses which included cold and steady-state running and the range of gap (between crown and skirt) sizes from 0.007 to 0.011 inches. Cracks of depths up to 0.48 inches were considered for the analysis. The result of this analysis was a prediction that, in the absence of significant residual stresses, the cracks will not propagate. Very low residual stresses are present in the stud boss area of AE piston skirts, as previously discussed.

130. Based on the favorable service history,^{18/} results of inspections of engine-operated AE piston skirts, and the results of experimental stress measurements and fatigue and fracture mechanics analyses, the Owners Group concluded that the AE piston skirts are adequate for unlimited life under full load operations.

131. No additional analyses were performed by SwRI. The analyses, results and conclusions of the Owners Group were found to be appropriate and applicable to TDI diesel engine piston skirts at PNPP. No specific recommendations were made by the Owners Group. SwRI agrees with the Owners Group analysis and therefore also makes no recommendations.

K. Cylinder Block and Liner

132. The cylinder block in the TDI standby diesel generators makes up the central framework for the engine. In formulating the design review task description for this component, the Owners Group determined that the critical functional attributes of the block are that it provide: 1) mounting support for the cylinder heads; 2) support for the cylinder liner and camshaft; 3) passages for engine coolant; and 4) reaction

^{18/} AE piston skirts have accumulated in excess of 6000 hours without failure. See Board Notification 84-152, August 29, 1984.

to the loads of the cylinder firing pressure. The cylinder liner forms the walls of the combustion chamber. Functional attributes of the liner are: 1) ability to contain the high temperature and high pressure combustion gasses (along with the cylinder head and piston); 2) ability to provide a guide for the piston; and 3) ability to withstand reactive side forces without excessive wear or scuffing.

133. The cylinder blocks were included among the Phase I components because of block top cracking exhibited in blocks in the Shoreham Nuclear Power Station engines and other TDI engines in nuclear and non-nuclear applications. Four types of cracks were identified in the Shoreham engine blocks: 1) ligament (from cylinder liner counter bore to head stud counterbore; 2) stud-to-stud; 3) stud-to-end of block; and 4) circumferential cracks at the liner counterbore. All of these cracks connected with the block-top surface and could be detected by surface inspection. All three engines at Shoreham had ligament-type cracks. At Shoreham, one block (from Emergency Diesel Generator ("EDG") 103) was found to have a Widmanstaetten graphite microstructure, discussed below. This block was found to also have stud-to-stud and stud-to-end cracks. No other nuclear engine blocks have been found to have substandard material. Shoreham is the only location where stud-to-stud and stud-to-end, and circumferential cracks have been

found. No instance has been reported where cylinder block cracks have resulted in failure of an R-4 or RV-4 engine. Cam gallery cracks were also found on the Shoreham cylinder blocks. However, the Owners Group has stated that these cracks are unique to the in-line R-4 engines and not of concern with the V-type engines installed at PNPP.

134. Since manufacture of the blocks, block to liner clearances have been verified and the liner vertical protrusion above the block top (proudness) has been reduced. PNPP has advised SwRI that the liner proudness has been reduced to the specification currently used by TDI.

135. The Owners Group analysis included the following studies: 1) review of block-to-liner cold clearance; 2) materials evaluation; 3) block and liner loading; 4) block top stress analysis; and 5) fatigue and fracture analysis of block cracks.^{19/} SwRI reviewed the Owners Group report with attention being directed towards the methods of analysis used, findings and interpretation of findings.

^{19/} A detailed discussion of the analysis performed is contained in the Owners Group report, "Design Review of TDI R-4 and RV-4 Series Emergency Diesel Generator Cylinder Blocks and Liners," prepared by FaAA.

136. The Shoreham engine blocks analyzed by the Owners Group had operated a significant amount of time at or above the nameplate rating of 3500 kW. As part of its engine requalification testing Shoreham operated each engine 100 hours at or above the design load. During engine disassembly and inspection, the aforementioned ligament cracks were found. After inspection, unit EDG102 was operated through 100 starts to loads greater than 50% nameplate load and was then reinspected. Eddy current examination showed no discernable extension of ligament cracks. In general, cracks were detected and measured using visual, liquid penetrant, and eddy-current inspection techniques.

137. A cylinder block strain gage test was conducted by the Owners Group on unit EDG103 following a 100-hour full power endurance test. Measurements of strain were recorded during preload while the cylinder head stud nuts were being tightened. Strain measurements were also recorded during steady operation at 0, 873, 1500, 2000, 2500, 3500 (full load) and 3830 kW. To measure effects of thermal loading, strain measurements were taken during a slow start, a quick start to full load, and a start during a loss of offsite power/loss of coolant accident ("LOOP/LOCA") stimulation. Strain measurements during quick starts did not exceed the peak steady-state values at each power level. As a means of studying the effect of different

load components on the block top stress state, contributions from bolt preload, thermal loading and cylinder pressure were obtained from the test data. The results were used in conjunction with a scale factor developed by the finite element analysis to estimate conservatively the mean and alternating stresses at the crack initiation sites of ligament and stud-to-stud cracks.

138. A metallurgical analysis was conducted by the Owners Group. The analysis consisted of an investigation of the microstructure, composition and mechanical properties of four TDI R-4 cylinder blocks from three engines at Shoreham. The four cylinder blocks investigated were EDG101, EDG102, EDG103 (original) and EDG103 (replacement). It was established that the microstructure of EDG101, EDG102, and EDG103 (replacement) was normal for grey cast iron but the EDG103 (original) block was characterized by an abnormal graphite distribution (degenerant Widmanstaetten graphite). It is generally accepted that the mechanical properties of grey cast iron with the Widmanstaetten ferrite are lower than normal. Higher than normal amounts of the tramp elements (lead and antimony), were present in the EDG103 (original) blocks.

139. Uniaxial tensile tests and smooth bar axial fatigue tests were performed on representative material for the EDG103

original block and other material with a normal microstructure. Fatigue crack growth rate tests were also performed using compact tension specimens. These tests demonstrated that the tensile strength and endurance limit for the EDG103 original material were significantly below normal. The cyclic stress tests on the compact tension also showed a lower than normal resistance to fatigue crack growth for the EDG103 original block material.

140. The study described in the Owners Group report was complete in scope and adequate data was presented to support the observations made. The study clearly showed that the original, cracked EDG103 cylinder block was characterized by an abnormal microstructure and that the mechanical properties of that material were substandard.

141. The Owners Group fracture and fatigue life evaluation produced a cumulative damage analysis which takes into account a cumulative Fatigue Damage Index ("FDI"). This index accounts for hours of operation at each power level and the corresponding mean stress and cyclic stress driving the crack at each power level. The index quantifies the effect of differing fatigue crack growth rates of different materials. This allows comparison of the test period experience on the original Shoreham EDG103 block, with its documented degraded fatigue

resistance, to the expected behavior of other type cylinder blocks having the fatigue resistance characteristic of typical grey cast iron under required test and postulated LOOP/LOCA conditions. Application of the cumulative damage analysis can be used to set future engine operation limits. Safe operation can be assured during a LOOP/LOCA based on Shoreham benchmark operations in combination with past operation of the engine provided proper procedures are followed as referred to in the Owners Group report.20/

142. The Owners Group analysis of circumferential cracks shows these cracks to be the result of loads applied to the cylinder liner landing. Liner landing pressure is controlled mainly by the interference of the liner collar (proudness) above the block top. PNPS engine blocks have been reworked to reduce this interference to the current TDI specification. The Owners Group finite element analysis shows that stress perpendicular to the plane of each postulated crack was found to decrease with distance from the corner and to become fully compressive at a depth of less than 0.5 inch. The original EDG103 block, with inferior material properties, developed cracks less than 3/8 inch deep after operation in excess of 1000 hours. It

20/ This may involve non-destructive examination, and material properties evaluations.

is concluded that circumferential cracking in blocks of nominal dimension and material properties will slow down and arrest and will not impair the operation of the TDI diesel generators.

143. SwRI has reviewed the Owners Group stress analysis and agrees with the assumptions, methods, procedures and results. Recommendations set forth by the Owners Group and concurred with by SwRI are the following:

- . Periodic inspections are necessary to demonstrate that each cylinder head block is capable of meeting its intended function as a component in a diesel generator in nuclear standby service.
- . All blocks should be metallurgically evaluated to verify that the microstructure is characteristic of typical grey cast iron.
- . Cylinder blocks that are inspected and found to be free of ligament cracks can operate without additional inspections for combinations of load and time that produce less than the excess cumulative damage index that has been demonstrated by its operation at the time of the latest block top inspection. Blocks of engines that have operated without block top inspection or for a time beyond the last inspection in excess of the allowable fatigue damage index should conservatively be assumed to have cracked ligaments.
- . For blocks with known or assumed ligament cracks, absence of detectable stud-to-stud or stud-to-end cracks between the heads should be established before returning the engine to emergency standby after any operation in excess of 50% nameplate load. Any stud-to-stud or stud-to-end crack indications must be evaluated with detailed inspection to assure that they extend less than 1.5 inches from the block top before returning the engine to emergency standby after any operation in excess of 50% nameplate load. It is necessary to evaluate the microstructure to ensure typical cast iron.

The above recommendations are being implemented at PNPP.

L. Turbocharger

144. The turbochargers on the TDI standby diesel generators utilize exhaust gas energy to drive the turbine which drives a compressor on a common shaft to pressurize the engine intake manifold. This forces more air into the combustion chamber, permitting more fuel to be introduced which results in higher combustion pressures and net engine output than that possible with natural aspiration. The TDI inline DSRV-4 series engines use a single turbocharger while the DSRV-4 series engines use two turbochargers, one for each bank. The PNPP DSRV-16 engines use two Elliott Model 90 G turbochargers each.

145. In drafting the design review task description for this component, the Owners Group determined that the critical functional attributes were that: 1) the turbocharger components have adequate strength and fatigue resistance to react to loads imposed by flowing engine exhaust gases; 2) the turbine and components have the ability to withstand a high temperature corrosive environment; 3) the lubrication system have the ability to supply sufficient oil to the bearings quickly after start up and continuously while running, and maintain adequate discharge oil temperatures; 4) the cooling system must provide sufficient water flow and pressure to maintain adequate

component temperatures; 5) the external piping must not transmit excessive thermal loads to the turbocharger casings; and 6) the turbocharger must be sized to avoid surge during operation.

146. The turbochargers were included among the Phase I components because of thrust bearing, nozzle vane, nozzle ring capscrew and washer, and nozzle ring failures on TDI nuclear standby diesel engines. Non-nuclear TDI engine applications also show failures due to the above problems and other causes that are addressed in the functional attributes.

147. The Owners Group performed analyses^{21/} to:

1) determine the loads on the thrust bearing and the nozzle ring assembly from the aerodynamic forces of the exhaust and induction air gases and the preload on the nozzle ring capscrews; 2) determine the load carrying capability of the thrust and radial bearings under both transient (startup) and steady-state conditions and the lubrication and cooling requirements in terms of the type of oil, oil flow rate, oil temperatures and oil pressure; 3) compare the loads imposed on the thrust bearing with the bearing load carrying capability

^{21/} A detailed discussion is included in the FaAA report "Design Review of Elliott Model 90G Turbocharger Used on Transamerica Delaval DSR-48 and DSRV-16 Emergency Diesel Generator Sets", prepared for the Owners Group.

and determine if the thrust bearing design is adequate under all operating modes; 4) determine the dynamic response of the rotating mass under startup, normal operation and overload conditions and compare this to the rotor systems natural modes and imbalance specifications to ascertain whether deflection and subsequent thrust bearing performance degradation or vibration would be a problem.

148. In addition to these analyses, the Owners Group performed inspections of thrust bearings from nuclear engines that had experienced a series of starts and some operating time. The thrust bearings from an engine that used the drip pre-lubrication system only experienced measurable wear on the order of 0.002 inches with 67 starts and 85 operating hours. The thrust bearing from an engine with a before and after (b & a) lubrication system, activated manually for two minutes prior to start, and with 32 starts and 275 operating hours, experienced no measurable wear.

149. The Owners Group concluded from its analyses that the thrust bearing would provide adequate service if a prelubrication system with a b & a pump and with a drip provision was installed and used during non-emergency starts. The Owners Group found that both the thrust and radial bearings were of adequate design, given proper lubrication, to operate

at the loads introduced by the exhaust and induction air gases during transient, normal operating and overload conditions. The Owners Group recommended that the thrust bearing be inspected at each 5-year overhaul interval or after 40 emergency (without the use of b & a pump) starts. The recommendation for inspection after 40 starts is based on findings from nuclear engines that have undergone 100 to 300 engine starts without the b & a pump (but with drip prelubrication) before thrust bearing failure. The Owners Group also recommends, based on this same data, that the drip prelubrication system, with at least 0.1 gallons per hour (gph) oil flow, be retained for emergency starts.

150. The Owners Group found no problems with regard to the dynamic performance of the rotor assembly. While the turbocharger passes through five critical speeds during startup, all of these are heavily damped and are passed through quickly. There are no critical speeds near the normal and overload turbocharger speeds of approximately 16,500 rpm. Analysis shows that rotor imbalance to 1.0 oz-in (the maximum allowable by the manufacturer is 0.06 oz-in), would not impair operation of the turbocharger. Thrust bearing deflection due to rotor shaft bending will not occur. Operation of the turbocharger is sufficiently removed from the surge line on the compressor performance map to preclude reverse loading damage.

151. The Owners Group reported on the nozzle ring capscrew, washer and vane failures in a supplement to the original report. They investigated the failures of these components at several nuclear power plants. Their analysis concluded that the nozzle vane failures were due to high cycle fatigue introduced by engine vibration or exhaust gas pulsations. The Owners Group report also concluded that while isolated vane failures may occur, they have not resulted in any degradation in diesel generator performance or a shutdown among the nuclear engines investigated.^{22/} No cause for the cracked washer was found.^{23/} The failure of the capscrews is attributed to improper preload. The single ring failure is attributed to probable impact from a broken vane. The Owners Group recommended that all turbochargers be inspected for nozzle vane damage and that all capscrews be properly torqued to the 18 to 22 lbf-ft specified by Elliott. In addition, the Owners Group recommended that the exhaust temperatures be monitored and corrective actions taken if they exceed the values recommended by TDI.

^{22/} Maintenance performed during every outage would permit identification of loose or missing vanes or capscrews.

^{23/} Failure of a washer would not cause any degradation of diesel generator performance.

152. SwRI reviewed the Owners Group analysis to confirm its applicability to the PNPP engines. The methodologies and assumptions were examined in detail to determine if they conformed to accepted engineering practice and supported the Owners Group conclusions. SwRI is in agreement with the Owners Group results and conclusions. SwRI would, however, add to the Owners Group recommendations an inspection to assure an adequate lubrication supply to the thrust bearings. Based on an independent analysis of the thrust bearing, SwRI recommends that PNPP: 1) assure that the oil supply to the turbocharger is filtered to 10 microns, as specified by Elliott; 2) perform spectrochemical analysis of the engine oil, as recommended by the Owners Group, to detect the copper level which would indicate degradation of the thrust bearing; 3) confirm that the b & a pump provides at least 0.5 gallons per minute flow; 4) confirm that the oil pressure to the turbocharger is between 25-35 psig and that the pressure, 10 seconds after start, is at least 10 psig; 5) that the oil inlet temperature is less than 180°F and that the outlet temperature is less than 215°F; 6) that the coolant outlet temperature is less than 190°F and that the coolant temperature rise is less than 30°F.

153. Given application and use of the drip and b & a prelubrication systems, compliance with the above mentioned turbocharger coolant and oil parameters, and adherence to the

Owners Group maintenance recommendations, SwRI feels that the Elliott Model 90G turbochargers will perform satisfactorily on the PNPP engines.

Jacket Water Pumps

154. The jacket water pump, taking suction from the jacket standpipe, delivers coolant (treated water) at the required pressure and flow rate to the engine jacket water header. In so doing, the pump provides the coolant circulation needed to cool the engine cylinder assemblies, exhaust manifold, turbocharger, intercooler, and engine lube oil cooler. A water pump failure would result in diminished or complete loss of coolant flow with subsequent overheating of critical engine systems. With loss of coolant and/or high coolant temperatures, the engine would eventually shut-down.

155. The particular water pump design used on the DRSV-4 engine does not have a history of failures. However, the jacket water pump used on the TDI standby generators equipped with the in-line diesel engine, model DSR-4, has a documented history of shaft failure in nuclear service. These failures occurred at the Shoreham facility, and resulted in the Owners Group conducting a design review of both pumps even though they differ in several areas.^{24/} Besides the difference in size of

^{24/} A detailed discussion of the analysis is included in Stone and Webster Engineering Corporation, "Emergency Diesel Genera-

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the pumps, other notable differences are: 1) the impeller diameter and impeller material; 2) the method of securing the impeller to the pump shaft; and 3) the shaft coupling mechanism used to interface the pump to the engine. These differences are important in that they limit the applicability of design evaluation and failure history to a particular pump. In other words, conclusions derived from analysis of the DSR-4 pump failure do not directly apply to the DSRV-4 pump.

156. In formulating the design review task description for the jacket water pump(s), the Owners Group determined that the critical functional attributes consist of the following:

1) that the pump drive gear and shaft coupling be adequate to transmit the combined torsional loading, and 2) that the pump shaft deliver the required torque to the impeller given the fluctuating nature of the input torque to the drive gear. To ensure that pump design was indeed capable of performing these critical functions, the design review for the DSRV-4 pump focused on the torque transmitting components. The analyses included a torsional analysis to determine maximum alternating torque (1595 lbf-in) and gear tooth loading (863 lbf), an

(Continued)

tor Engine Drive Jacket Water Pump Design Review," April 1983, and its supplement, June 1984.

analysis to determine mean impeller torque (2426 lbf-in) at rated speed and flow, and a stress analysis of the shaft coupling and impeller-to-shaft interface using the calculated torque values. The results of the analyses indicated more than adequate design margin in the components (design factors greater than three in all cases).

157. It should also be noted that the Owners Group report for the DSRV-4 pump reflects the use of two different impeller diameters - either 12.1 inches or 10.75 inches. This variability is due to differences in plant specification and the consequential coolant flow requirement for the standby generators. In addressing the impact of the difference in diesel impeller diameter on the review, the Owners Group used a conservative approach - one based on analyzing the system with the larger impeller. This situation represented "the worst case" in terms of shaft and coupling loading. Consequently, the stress levels in those pumps with the smaller diameter (such as the 10.75 inch impeller at PNPP) would actually be less than calculated in the review.

158. Based on their findings, the Owners Group concluded that the pump design was adequate for nuclear service. The results did show, however, that certain aspects of the design had limitations, but that these were overcome by redundancy in

component function. The drive fit of the impeller onto the shaft taper (at a specified torque of 80 ft-lbs) was determined to be adequate to resist the torsional loading. The key and keyway, however, were of sufficient strength to transmit the load. The actual ratio of working stress to the material yield strengths was less than 1/3. For this case, it was assumed that the drive fit would help prevent the key from "rocking" and wearing both itself and the keyway. Furthermore, the analysis did not reflect any rotational influence of the impeller on the degree of interference at operating speed.

159. The drive fit of the external spline on its shaft taper (at specified nut torque of 120 ft-lbs) was also found to be inadequate for transmitting the total torque; it too required the key to transmit the load. The analysis also indicated that, because of the locking feature of the nut, it was possible (during assembly) to have an installation torque higher than specified. Accordingly, the Owners Group recommended that the installation procedure reflect both a minimum torque of 120 ft-lbs and a maximum torque of 660 ft-lbs. Stresses at the maximum torque were calculated to be approximately 2/3 of the shaft yield strength.

160. SwRI reviewed the Owners Group report on the DSRV-4 jacket water pump to verify its applicability to the pumps at

PNPP. The Owners Group methodology was found to be consistent with accepted engineering practice. In conducting the review, SwRI took into consideration the actual diameter of the PNPP pump impellers which are 10.75 inches in diameter. Additionally, SwRI considered the impeller rotation and the effects of the stresses due to centrifugal force on the drive fit. The concern was that the impeller might become loose on the shaft taper and accelerate keyway wear.

161. SwRI concluded that the Owners Group review was applicable to the PNPP installation. The smaller impeller diameter had little effect on the torsional calculations as the differences in assumed mass moment of inertia were small (OG: 665 lbm-in², SwRI: 616 lbm-in²). The effect of impeller rotation on the drive fit at the impeller end was more significant. Further assessment of this assumption indicated that thermal expansion of the shaft would compensate for the rotational effects and maintain adequate interference with the impeller hub to prevent accelerated wear of the keyway. SwRI recommends, however, that the impeller keyway be inspected at each outage to detect any key and keyway wear.

Crankshaft

162. The crankshaft in TDI standby diesel generator engines converts reciprocating motion, component inertial forces and gas pressure piston forces to rotary motion and torque at the output flange. In formulating the design review task description for this component, the Owners Group determined that the critical functional attributes of the crankshaft were that it have sufficient structural stiffness to maintain acceptable stresses in the crankpin web and main journal areas, maintain system natural frequencies sufficiently removed from engine operating speeds, and that the design be able to withstand normal main bearing misalignments. Also, the main bearings and crankpin areas must be large enough to maintain proper lubricating oil film pressure but small enough to prevent endwear of the bearing sleeves. The material of the crankshaft and the surface finish should be sufficient to resist fatigue crack initiation.

163. The 13" x 13" DSRV-16-4 engine crankshaft was included among the Phase I components due to three crankshaft failures in non-nuclear applications. There have been no failures of 13" x 13" crankshafts on DSRV-4 engines in nuclear service. The failures were attributed to torsional fatigue cracks initiated in the oil holes of main journal numbers 6 or 8. Only the

smaller 11" x 13" crankshafts on the three Shoreham Nuclear Power Station DSR-48 engines failed.

164. The Owners Group analysis was performed for the 13" x 13" crankshafts in the emergency diesel engines at Grand Gulf Nuclear Power Station. The engine types and the crankshafts of PNPP are of identical design to those at Grand Gulf.

165. The crankshaft is required to meet the recommendations of the Diesel Engine Manufacturers Association ("DEMA"). The Owners Group reviewed the design calculations and torsionograph test results of TDI for compliance with the DEMA recommendations. The DEMA recommendations state:

a. In the case of constant speed units, such as generator sets, the objective is to insure that no harmful torsional vibratory stresses occur within five percent above and below rated speed.

b. For crankshafts, connecting shafts, flange or coupling components, etc., made of conventional materials, torsional vibratory conditions shall generally be considered safe when they induce a superimposed stress of less than 5000 psi, created by a single order of vibration, or a superimposed stress of less than 7000 psi, created by the summation of the major orders of vibration which might come into phase periodically.

166. Diesel generator torques due to dynamic response are calculated in two steps. In the first step, TDI determined the natural frequency of the crankshaft. TDI used the well-established Holzer method to calculate the system's first three natural frequencies. The first natural frequency of Grand Gulf's crankshaft was found to be 28.8 Hz which produces 4th order resonance at 432 rpm of the engine. In the case of the PNPP engine the corresponding figures are 29.23 Hz and 438 engine rpm due to the unique PNPP generator rotor and flywheel mass moments of inertia. In the second step of torsional critical stress analysis TDI determined the dynamic torsional response of the crankshaft due to gas pressure and inertia loading. In the case of Grand Gulf, TDI performed the calculations for each order of vibration up to the 12th order. In the case of the PNPP engines, TDI provided the normalized values of the torsional loading for significant orders. These values were compared by the Owners Group with those recommended by Lloyd's Register of Shipping. The largest single order was measured to be within 4% of those computed using TDI's value of normalized torsional loading. TDI used an empirical form of calculation to ensure that the diesel generator can be brought up to operating speed without undergoing excessive stresses as critical speeds are passed. In the case of the PNPP engine, the 4th order critical speed of 438 rpm is important. The Owners Group

analysis concluded that in this particular type of design (V-16 engine with articulated connecting rods) the 4th order loading from one bank cancels the other, which significantly reduces the excitation. However, the excitation is sensitive to the balance between the two banks.

167. The nominal shear stresses for the significant orders were calculated. The largest single order nominal shear stress of 1956 psi was found to be well below the 5000 psi allowable by DEMA. The corresponding stress level in PNPP engines was calculated to be 1780 psi. The torsigraphic tests performed on the Grand Gulf engine by TDI were reviewed by the Owners Group.^{25/} On the Grand Gulf engine the first natural frequency was found to be 28.7 Hz which compared very well with TDI's computed value of 28.8 Hz. The TDI analysis established that the nominal shear stress is 8540 psi per degree of free-end vibration for the Grand Gulf engine. The Owners Group calculated a vector summation of orders 1 through 8 for the PNPP crankshaft of 0.583 degree amplitude which would yield 4979 psi according to SwRI. This is less than the 7000 psi maximum DEMA allowable stress. The largest single order stress of 2028 psi was found to be less than the DEMA allowable stress of 5000

^{25/} A torsigraphic test will be performed at PNPP during the engine pre-operational period. See Applicants' Direct Testimony of Edward C. Christiansen on Issue No. 16 at ¶ 42.

psi. The corresponding figure for the PNPP engine was 2118 psi - also less than the DEMA recommendation. A good agreement was found between the TDI calculated stresses and the measured stresses.

168. The Owners Group verified data to make sure that the crankshaft stresses are within the DEMA allowable for a speed range of 440 to 450 rpm $\pm 5\%$. For the PNPP engine at 7000 kw power rating and 95% of rated speed the maximum nominal shear stress was found to be 2281 psi. At 7000 kW power rating and 105% of rated speed the maximum nominal stress was found to be 3296 psi. The amplitude of both these stresses is well below the DEMA allowable stress of 5000 psi. The Owners Group analysis has emphasized the importance of adequate engine balance and the necessity of not operating the engine below 440 rpm except during startup and shutdown. The Owners Group analysis concludes that the balance specifications provided by TDI may be adequate, however, it has cautioned against a possible conclusion that all engines will respond identically to this balance. The torsigraph test planned for the PNPP engines will determine whether these engines are balanced satisfactorily.

169. The Owners Group performed a dynamic torsional analysis of the crankshaft to determine the true range of torque at each crankthrow. The Owners Group developed a torsional model

of the crankshaft to supplement TDI's conventional forced vibration calculations. In order to compute the true summation of the stresses due to various orders, the model included computations for phase relationships between the various orders. The first three natural frequencies of the PNPP crankshaft were calculated. The natural frequencies were found to be in agreement with those computed by TDI.

170. The Owners Group analysis considered the harmonic loading on the crankshaft. Gas pressure, inertia forces and frictional loading were considered for the harmonic loading calculations. The dynamics of a V-16 engine are such that the 4th order load components from the left and right banks almost cancel. In practice, however, due to various reasons such as manufacturing tolerances and individual cylinder timing, the balance is not complete. To simulate the unbalance the Owners Group applied one degree delay in the right bank cylinder. In the absence of torsigraphic test data, the calculated amplitudes are not compared with the measured amplitudes. The model was used to calculate the range of torques at each crankthrow. The stress level was found to be highest between cylinders 3 and 4, 5 and 6, 7 and 8. The highest nominal shear stress amplitude of 5335 psi was found to be lower than the 7000 psi allowable by DEMA. For the PNPP crankshaft, the critical speed was found to be 438.4 rpm. The Owners Group performed

calculations to determine the amplitude of free-end vibration and the associated nominal shear stress at 7000 kW power and at a crankshaft critical speed of 438.4 rpm. The highest nominal shear stress amplitude of 5128.8 psi was found to be lower than the 7000 psi allowable by DEMA.

171. The Owners Group performed a modal superposition analysis of the crankshaft for PNPP. Pressure loading was obtained from the dynamic test at Shoreham. This analysis calculates the nominal shear stresses at each crankpin and main journal location. The maximum amplitude of nominal stress was found to be 5335 psi between cylinder numbers 5 and 6 for a load at 7000 kW. The nominal stresses were found to satisfy the DEMA requirement, and are less than 5000 psi for a single order, and less than 7000 psi for combined orders.

172. The Owners Group analysis concluded that the TDI calculations are appropriate and show that the crankshaft stresses are below DEMA recommendations for a single order. The torsigraph tests performed on a DSRV-16-4 diesel generator set at the Duke Power Catawba Nuclear Generating Station showed the peak-to-peak crankshaft stresses to be within the DEMA recommendations.

173. The Owners Group analysis is appropriate and applicable to the PNPP engine. SwRI did not find it necessary to

perform any additional analyses. Considering the similarities of the crankshaft design and the similarities of TDI's and the Owners Group results, SwRI has concluded that the PNPP crankshafts are adequate for their intended service provided all of the Owners Group recommendations are followed. SwRI recommends that a torsigraph test be performed and the results compared with both the Owners Group analysis and TDI's calculations. The results of the torsigraph test should demonstrate that the crankshaft stresses meet DEMA standards and that they are close to the TDI and Owners Group values. Individual cylinder timing should be adjusted to accomplish engine balance and bring the stress values in line with those of TDI and the Owners Group. The setpoint of the governor is above the critical speed of 438 rpm.

Connecting Rods

174. The connecting rods used in the TDI standby diesel generators installed at PNPP transmit engine cylinder firing forces from the pistons to the crankshaft such that the reciprocating motion of the pistons induces shaft rotation and output torque. By virtue of the V-cylinder configuration, the mechanism required to perform this function consists of two connecting rods arranged in a master-articulated manner. This arrangement requires more sophistication in bearing design and

assembly methodology than that found with in-line type engines. Such complexity makes the design analysis more involved and requires that there be substantial operating experience and/or experimental data to confirm the design integrity.

175. Initially, the connecting rod assembly was produced with 1-7/8 inch diameter bolts to secure the main assembly joint (master rod box to link rod box). Operating experience in non-nuclear applications indicated that the design was vulnerable to failure by one of two mechanisms -- either by loss of bolt preload resulting in bolt failure or by the fatigue failure of the master rod box with cracks initiating at the thread roots. To compensate for the loss of bolt preload, the installation torque specification for the joint bolts was increased. Corrective action for the fatigue of the master rod box resulted in a design revision with 1-1/2 inch diameter bolts, torqued to 1700 lbf-ft to provide the same clamping force obtained with the 1-7/8 inch bolt configuration torqued to 2600 lbf-ft. This change in bolt diameter provided a greater material section and consequently reduced the stress levels in the master rod box. PNPP is supplied with the 1-1/2 inch bolts.

176. In light of the failure history compiled with non-nuclear application of the connecting rod and the fact that

a rod failure would require immediate shut-down of the engine, the Owners Group included the connecting rod in the Phase I components. The scope of the review included engines with both types of rod assemblies (i.e., those having the 1-7/8 inch bolts as well as those having 1-1/2 inch bolts). In formulating the design review task description for the connecting rod assembly, the Owners Group determined that the system must provide three functional attributes in order for it to be acceptable for nuclear standby usage: 1) it must be of sufficient buckling strength and fatigue resistance to withstand the expected firing forces and inertial loads; 2) it must be of sufficient dimensional uniformity so as not to distort lubrication performance and/or the clamping effectiveness of mating surfaces; and 3) it must utilize fasteners and torque specifications that support the combined loading without fatigue, preload relaxation, or severe thread distortion.

177. The design review conducted by the Owners Group to verify the functional attributes focused on three aspects:^{26/} 1) a detailed physical examination of failed components to verify failure mechanisms; 2) finite element stress analysis to predict performance under those engine operating conditions

^{26/} A detailed description of this analysis is included in FaAA report "Design Review of Connecting Rods for Transamerica Delaval DSRV-4 Series Diesel Generators," August 1984, prepared for the Owners Group.

dictated by nuclear service, and 3) comparison of the results from 1) and 2) above with experimental data and operating history to establish credibility.

178. The analysis conducted by the Owners Group indicated that both rod assembly designs were acceptable for nuclear service. In the case of the 1-7/8 inch bolt configuration, the fatigue sensitivity exhibited (in terms of actual failures) was confirmed through physical examination of failed parts. Supporting analytical documentation quantified the failure process in terms of the maximum operating stress level for the 1-7/8 inch bolts (mean: 11 ksi, alternating: 4.94 ksi) and for the 1-1/2 inch bolts (mean: 8.82 ksi, alternating: 4.51 ksi) and its relationship to the modified Goodman line failure criteria. The Goodman line was defined by the ultimate strength of the connecting rod material (AISI 4142) of 115 ksi and a calculated endurance limit at the bolt hole location of 5.213 ksi. Comparison with the Goodman criteria showed that the material would indeed have a finite life. This observation was further supported with calculations of crack growth rate, which showed that the effective stress intensity factor range was 5.27 ksi in. which is within the fatigue threshold range of this material of from 5.0 to 7.5 ksi in.

179. The methodology outlined above was also applied to the 1-1/2 inch bolt configuration. The results, however, showed that the 1-1/2 inch design was not vulnerable to fatigue failure under the same engine load conditions. Application of the same Goodman line failure criteria yielded a safety factor of 1.08 for the 1-1/2 inch bolts. While a safety factor of this magnitude is low, field experience with connecting rods with 1-1/2 inch bolts has been favorable.

180. For both designs, the Owners Group recommended that each rod assembly be inspected with nondestructive techniques to confirm the absence of flaws. In the case of the 1-7/8 inch bolt system, the inspection should be performed at intervals of 200 engine hours. For the 1-1/2 inch bolt system, such as that used at PNPP, the inspection need only be done initially to confirm the absence of pre-existing flaws. Beyond these inspections, the rod assemblies need only be checked for proper bolt torque at installation.

181. SwRI reviewed the Owners Group report on the connecting rod assembly to verify its applicability to the connecting rod assemblies in place at PNPP. The methodology and assumptions were examined in detail to ensure consistency with accepted engineering practice and to ascertain that the findings did indeed support the conclusions. No exceptions or

unresolved differences regarding the report and its conclusions were found. Furthermore, SwRI concludes that the Owners Group findings with respect to the 1-1/2 inch rod configuration do apply to the PNPP diesel generator units. The rod assemblies are suitable for their intended application provided the maintenance recommendations of the Owners Group are observed.

These recommendations are that: 1) the bolt holes be initially inspected to verify the absence of pre-existing flaws, 2) the protocol for torquing the rod bolts reflect the importance of cleaning the threaded surfaces and using a thread lubricant as specified in the TDI instruction manual, 3) the bolt torque be checked at each outage to ensure that bolt preload has not been lost, and 4) a comprehensive inspection of the connecting rods be conducted at each engine overhaul interval.

IV. CONCLUSIONS AS TO THE TDI DIESEL GENERATORS' RELIABILITY

182. In order for the PNPP TDI diesel generators to perform reliably, they must meet the following requirements:

(1) they must start on demand, attain rated speed, and accept load within the specified time; and (2) they must have the capacity to supply power continuously to the equipment needed to maintain the plant in a safe condition. SwRI has concluded that the sixteen Phase I components in the PNPP engines are of satisfactory design and will perform their intended

safety-related function based on a review of the Owners Group reports and backup material, analytical analysis of various component attributes, and discussions with Owners Group and PNPP personnel. Based on its investigation, SwRI concludes that the PNPP engines will permit the diesel generators to perform reliably as defined by the requirements listed above.

183. Achievement of reliable diesel generator performance also requires that PNPP implement the recommendations of the Owners Group and SwRI with regard to inspection and maintenance. Initial inspections recommended by the Owners Group and SwRI prior to diesel generator operation have been performed at PNPP. Other inspections and maintenance actions required periodically during the operational life of the diesels have been delineated in the Owners Group Maintenance Matrix for the PNPP engines which is included in the PNPP DR/QR Report. Other inspection and maintenance actions recommended by SwRI, as outlined in the preceding paragraphs, will also be implemented at PNPP.

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- o Section Manager, Automotive Research Department at SwRI (1967-74)
- o Senior Research Engineer, Automotive Research Department at SwRI (1962-67)
- o Propulsion Engineer, Ling-Temco-Vought, Inc. (1961-62)
- o Test Engineer, Ling-Temco-Vought, Inc. (1958-62)

Education:

B.S. in mechanical engineering, Rice University, 1956
M.S. in mechanical engineering, Southern Methodist University, 1962

Professional Qualifications:

Registered Professional Engineer in the State of Texas.

Professional Associations:

- o SAE (formerly Society of Automotive Engineers). Positions held with SAE: Advanced Power Plant Committee, Diesel Engine Committee, Chairman of South Texas Section.
- o Texas Society of Professional Engineers.

Publications:

"An Unthrottled Gaseous Fuel Conversion of a Two-Stroke Diesel Engine," by T. E. Ritter and C. D. Wood, SAE No. 750159, February 1975.

"Design and Development of a Lightweight Winch Drawn Bulldozer," by T. E. Ritter, C. D. Wood and J. W. Colburn, Jr., International Society for Terrain-Vehicle Systems, Proceedings of the ISTVS Conference, Vol. III, June 1975.

"Mechanisms of Dust Erosion," by C. D. Wood and Park Espenschade, SAE Transactions, Vol. 73, 1965.

"REDSOD—Techniques and Performance," by C. D. Wood, SAE No. 730850, September 1973.

"An Analog Heat Release Computer for Engine Combustion Evaluation," by R. A. McFarland and C. D. Wood, SAE No. 760553, 1976.

"Improvement of Automobile Fuel Economy," by C. W. Coon and C. D. Wood, SAE No. 740969, October 1974.

"Unthrottled Open-Chamber Stratified Charge Engines," by C. D. Wood, SAE No. 780341, February 1978.

"An Experimental Investigation of an Open-Chamber Stratified Charge Engine," by C. D. Wood, presented at the Central States Section, 1978 Spring Technical Meeting, The Combustion Institute, March 1978.

"Performance of a Stratified Charge Engine," by C. D. Wood, SAE No. 790434, February 1979.

"Performance of Coal Slurry Fuel in a Diesel Engine," by K. Tataiah and C. D. Wood, SAE No. 800329, February 1980.

"Direct Injected Methanol Fueling of Two-Stroke Locomotive Engine," by C. D. Wood and J. O. Stormont, SAE No. 800328, February 1980.

"A Simplified NO_x Model for Spark Ignition Engines," by C. D. Wood, ASME No. 80-DGP-43, February 1980.

"Alternative Fuels in Diesel Engines - A Review," by C. D. Wood, Reprinted from SP-480 - "Alternate Fuels," February 1981.

"Experience on the Use of Fuel-Air Explosions for Controlled Earth-moving," by C. D. Wood, A. R. Nye, R. B. Melton and D. L. Craft, SAE No. 710160, January 1971.

"Erosion of Metals by the High Speed Impact of Dust Particles," by C. D. Wood, Institute of Environmental Sciences' 1966 Annual Technical Meeting Proceedings, 1966.

"Vehicle Diagnostic Systems - The State of the Art," by Robert N. Hambright and Charles D. Wood, ASAE Paper No. 73-144, presented at the 1973 Annual Meeting of the American Society of Agricultural Engineers, June 1973.

"Alternate Forms of Energy" by Conan Furber and Charles Wood presented at the Railway Fuel and Operating Officers Association, Technical Conference, Chicago, Illinois, September 22-24, 1980.

"Trends in Diesel Power," Oral Presentation by Charles Wood to the 33rd International Convention of the Association of Diesel Specialists in Houston, Texas, September 24, 1980.

Inventor or Co-Inventor of the Following U.S. Patents:

- 3,572,273 Apparatus for Breaking a Layer of Ice on a Body of Water by Repetitive Combustion Explosions, C. D. Wood
- 3,600,116 Air-Control System for Apparatus Displacing Material by Combustive Explosions, C. D. Wood, J. M. Clark, and A. R. Nye.
- 3,680,287 Air Filter, C. D. Wood, S. A. Olsen, and J. C. Potter.
- 3,744,018 Method of an Apparatus for Producing a Repetitive Seismic Impulse, C. D. Wood.
- 3,750,837 Explosive Seismic Energy Source and Quick Release Valve, C. D. Wood.
- 3,752,240 Method of an Apparatus for Proving an Impact to a Vehicle-Carried Penetrating Tool, C. D. Wood and John M. Clark.
- 3,756,416 Apparatus Having a Filter Panel Disposed Across a Fluid Passageway, C. D. Wood.
- 3,787,144 Explosive Pumping and Dredging Method and Apparatus, C. D. Wood.
- 3,801,346 Method for Applying Particulate Coating to a Work Piece, C. D. Wood, R. B. Melton, J. M. Clark, R. J. Mathis, and W. D. Weatherford.
- 3,839,848 Method and Apparatus for Cleaning Air, C. D. Wood and J. M. Clark.
- 3,880,568 Combustion Method and Apparatus for Generating Repetitive Explosions, C. D. Wood, J. W. Colburn, and R. B. Melton.

- 3,915,381 Method and Apparatus for Applying Particulate Coating Material to a Work Piece, C. D. Wood, J. M. Clar, R. B. Melton, R. J. Mathis, and W. D. Weatherford.
- 3,918,937 Particulate Lead Trap System, C. D. Wood, J. G. Holloway, H. W. Barch, and M. B. Treuhaft.
- 3,918,944 Lead Trap, C. D. Wood and M. B. Treuhaft.
- 3,924,897 Earth Ripper Employing Repetitive Explosions, C. D. Wood, R. J. Mathis, A. R. Nye, and J. W. Colburn.
- 4,011,886 Sleeve Valve, C. D. Wood
- 4,194,574 Draft Power Sensor, H. S. Benson and C. D. Wood

RELATED CORRESPONDENCE

DOCKETED
USNRC

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UNITED STATES OF AMERICA
NUCLEAR REGULATORY COMMISSION

OFFICE OF SECRETARY
DOCKETING & SERVICE
BRANCH

BEFORE THE ATOMIC SAFETY AND LICENSING BOARD

In the Matter of

THE CLEVELAND ELECTRIC
ILLUMINATING COMPANY, ET AL.

(Perry Nuclear Power Plant,
Units 1 and 2)

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Docket Nos. 50-440
50-441

CERTIFICATE OF SERVICE

This is to certify that copies of the foregoing APPLICANTS' DIRECT TESTIMONY OF EDWARD C. CHRISTIANSEN ON ISSUE NO. 16, APPLICANTS' DIRECT TESTIMONY OF JOHN C. KAMMEYER ON ISSUE NO. 16, and APPLICANTS' DIRECT TESTIMONY OF CHARLES D. WOOD III ON ISSUE NO. 16 were served by deposit in the United States Mail, first class, postage prepaid, this 25th day of March, 1985, to all those on the attached Service List, except for those parties identified by a single asterisk, who were served by hand-delivery and those parties identified by a double asterisk, who were served by express mail.

Rose Ann Sullivan
Rose Ann Sullivan

UNITED STATES OF AMERICA
NUCLEAR REGULATORY COMMISSION

BEFORE THE ATOMIC SAFETY AND LICENSING BOARD

In the Matter of)	
)	
THE CLEVELAND ELECTRIC)	Docket Nos. 50-440
ILLUMINATING COMPANY, <u>ET AL.</u>)	50-441
)	
(Perry Nuclear Power Plant,)	
Units 1 and 2))	

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