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EVALUATION OF DSR-48 EMERGENCY DIESEL GENERATOR CRANKSHAFTS AT RIVER BEND STATION

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1.0 INTRODUCTION AND SUMMARY

An evaluation of the 13-inch by 12-inch crankshafts installed in the emergency diesel generators at River Bend Station, RBS, was performed to determine the adequacy of the crankshafts for their intended service. This report summarizes the results of testing, analyses, and inspections performed on the crankshafts in the two Transamerica Delaval Inc., TDI, DSR-48 engines installed at RBS.

The TDI part number for the crankshafts at RBS is 03-310-05-AC. The plant ID for engine serial no. 74039 is 1EGS*EG1A (EG1A) and for engine serial no. 74040 is 1EGS*EG1B (EG1B). The forging and machining of the crankshafts was performed by Ellwood City Forge Corporation.

Significant testing and evaluation of 13-inch by 12-inch crankshafts installed in the TDI DSR-48 engines at Shoreham Nuclear Power Station, SNPS, has been performed. The part number for the crankshafts at SNPS is 03-310-05-AC. Fatigue damage was discovered in the original 13-inch by 11-inch crankshafts at SNPS and the replacement crankshafts installed are of the same design as those in use at RBS. The two crankshafts have identical crankshaft stiffness, inertia and fillet geometry. The main differences in the two torsional systems are that the engines have different flywheel and generator inertias and a different shaft stiffness between the flywheel and generator. Due to the similarities of the two torsional systems the response of the crankshafts are similar, and this similarity has been demonstrated by analysis and testing.

The testing on the crankshaft at SNPS consisted of dynamic strain gage measurements to determine maximum stresses, torsiograph testing and an endurance test run for 10^7 cycles at loads equal to or greater than 3300 kW. Dynamic torsional analysis and stress analysis have been performed on the SNPS crankshaft. In addition, the detailed investigation of the failure of the original 13-inch by 11-inch crankshaft at SNPS provides data for evaluation of the fatigue life of the crankshafts. To utilize the data available from the evaluation of the SNPS crankshafts, similarities and comparisons to the RBS crankshafts will be made throughout the report where appropriate.

The crankshaft is required to meet the recommendations of the Diesel Engine Manufacturers Association, DEMA. In Section 2.0, the RBS torsional design calculations prepared by TDI and the results of the torsiograph tests performed at RBS by Stone & Webster Engineering Corporation, SWEC, are reviewed for compliance with DEMA stress allowables. The relationship between torsional response and engine load is determined based on the results of the torsiograph test. The crankshafts at RBS meet the DEMA allowables for single order and combined order stresses at 450 rpm and 3500 kW.

The fatigue analysis to determine the safety margin of the crankshafts at RBS at 3130 kW is discussed in Section 3.0. Amendment 16 to the FSAR states 3130 kW as bounding the maximum emergency service loads. A dynamic torsional analysis of the crankshaft is performed to determine the stress levels in each cylinder of the crankshaft. This model is compared with the SWEC test data for the amplitudes of free-end vibration. Stress values at full load are determined over a range of ±5% of rated speed.

The fatigue endurance limit is established for the crankshafts at both RBS and SNPS by first obtaining the endurance limit for the failed crankshaft at SNPS from strain gage test data and a review of literature. The differences between the failed crankshaft and those in use at RBS and SNPS are then assessed and the endurance limit is then scaled to account for these differences.

Utilizing the SNPS dynamic strain gage test data, the factor of safety against fatigue failure is calculated to be 1.39. This value for the factor of safety is established with more confidence than usual on account of the dynamic strain gage testing of a similar crankshaft at SNPS, and on account of the endurance limit established from the failed crankshafts at SNPS. The variation of the factor of safety over the ±5% rated speed range is then reviewed. The reduction in the factor of safety at overspeed compared with rated speed is found to be small in comparison to the safety margin.

In Section 4.0, the RBS load that produces stress levels equal to those in the SNPS crankshaft during the 10^7 cycle endurance test is determined. The applicability of the SNPS endurance test results to the crankshafts at RBS is discussed.

The safety margin of 1.39 for the RBS crankshafts at 3130 kW demonstrates that the crankshafts are adequate for their intended service at loads not exceeding 3130 kW.

2.0 COMPLIANCE OF CRANKSHAFT WITH DEMA RECOMMENDATIONS

The purchase specifications for the diesel generator sets required that the recommendations of the Diesel Engine Manufacturers Association, DEMA [2-1], be followed. These recommendations state:

> In the case of constant speed units, such as generator sets, the objective is to insure that no harmful torsional vibratory stresses occur within five percent above and below rated speed.
>
> For crankshafts, connecting shafts, flange or coupling components, etc., made of conventional materials, torsional vibratory conditions shall generally be considered safe when they induce a superimposed stress of less than 5000 psi, created by a single order of vibration, or a superimposed stress of less than 7000 psi, created by the summation of the major orders of vibration which might come into phase periodically.

In March, 1975, Transamerica Delaval Inc. (TDI) performed a torsional critical speed analysis of the crankshafts at RBS [2-2]. References to TDI analysis in the body of this report all reference this effort. In Section 2.1, this analysis will be reviewed for compliance with the above allowable stresses. Where appropriate, data for SNPS will be included in the tables for comparison [2-3]. In September 1984 and again in May 1985, Stone & Webster Engineering Corporation, SWEC, conducted torsiograph tests on the crankshaft in EG1A at RBS [2-4, 2-5]. In Section 2.2, the test results will be compared with the above allowable stresses.

2.1 Review of TDI Torsional Critical Speed Analysis

Diesel generator torques due to dynamic response are usually calculated in two steps. First, the torsional mode shapes and natural frequencies of vibration are calculated. Second, the dynamic forced vibration response due to gas pressure and reciprocating inertia loading is calculated. TDI calculated the response at 100% of rated load of 3500 kW.

2.1.1 Natural Frequencies

The first step in a torsional critical speed analysis is to determine the natural frequencies of the crankshaft. The engine speed at which a given order resonates may then be calculated. The diesel generator is modeled as a system of lumped mass moments of inertia interconnected by torsional springs, as shown in Figure 2-1. The inertia and stiffness values are shown in Table 2.1.

It has long been standard practice in the diesel engine industry to solve this eigenvalue problem by the Holzer method [2-6]. This method has been used for at least 40 years [2-7], and thus is well established.

TDI used the Holzer method to calculate the system's first three natural frequencies, which are shown in Table 2.2. The first natural frequency was found to be 38.0 Hz, which produces 4th order resonance at 570 rpm.

2.1.2 Nominal Stresses

The second step in a torsional critical speed analysis is to determine the dynamic torsional response of the crankshaft due to gas pressure and reciprocating inertia loading. The 1st order is a harmonic which repeats once per revolution of the crankshaft. For a four-stroke engine, harmonics of order 0.5, 1.0, 1.5, 2.0, 2.5... exist. TDI performs this calculation for each order of vibration up to 12.0 separately. For each order, the applied torque at a cylinder due to gas pressure and reciprocating inertia is calculated. The values of this torque for each order are usually normalized by dividing by the piston area and throw radius. The normalized value for the nth order is referred to as T_n. The values of T_n for significant orders used by TDI are shown in Table 2.3. These values may be compared to those recommended by Lloyd's Register of Shipping, LRS [2-8]. It is found that TDI's values are higher than LRS's values for low orders and lower for high orders. However, the stress from the measured largest single order was found to be within 5% of that computed by TDI. The response is then calculated by one procedure if the harmonic is at resonance and by another if the harmonic is away from resonance.

At resonance, the torsional vibration amplitudes would increase indefinitely in the absence of damping. The solution is obtained by balancing the energy input with the energy loss due to damping. TDI used an empirical form of hysteresis damping due to friction. The purpose of this calculation is to ensure that the diesel generator could be brought up to operating speed without undergoing excessive stresses as critical speeds are passed. Observations have shown that excessive vibration during startup does not occur [2-4, 2-5].

Away from resonance, the torsional vibrations reach a steady-state level even without the aid of any damping. The magnitude of this response for each structural mode and loading order is calculated as the product of a dynamic amplification factor and an equivalent static equilibrium amplitude. The equivalent static equilibrium amplitude is computed using a modal load and modal stiffness [2-9] for the nth order harmonic and given mode shape. The nominal shear stress, τ, in the 12-inch pin of Crankpin No. 8 for each order is then calculated from the dynamic torque, T, using $\tau = Tr/J$, where r is the pin radius and J is the polar moment of inertia.

TDI calculated the response for the first three modes and plotted the results for only the first mode since higher modes produce much smaller stresses. The nominal shear stresses for the significant orders are shown in Table 2.4. It is seen that the largest single order stress of 3321 psi at rated load and speed for the 4th order is well below the 5000 psi DEMA allowable.

TDI does not calculate the associated phase angle with the response of each order, so that it is not possible to calculate the combined response. The measured combined response will be compared with the allowable in the next section.

2.2 Review of Stone & Webster Engineering Corporation Torsiograph Test

The purpose of the torsiograph test of the emergency diesel generator is to measure the angular displacements of the forward end of the crankshaft. These displacements are then used in conjunction with a dynamic torsional analysis of the crankshaft to confirm the results of the vibrational analysis and to assess the maximum stresses in the crankshaft. The torsiograph response may be decomposed into components corresponding to each order and the peak-to-peak response may also be obtained. The torsiograph test is usually performed in two stages. The first stage is performed without load at variable speed and is used to determine the location of critical speeds. Critical speeds may also be determined while operating at a fixed speed and observing the frequency content of the response. The second stage is performed at rated speed of 450 rpm with variable load, and is used to confirm the forced vibration calculations. These tests collect data during steady state operation. Data may also be obtained during transient (start-up, coast-down, misfiring) operations to assess the response of the crankshaft during these events.

Two torsiograph tests were performed on EG1A at RBS by SWEC. The first test, which was performed in September 1984, concentrated on collecting data during steady-state operation [2-4]. The second test, performed in May 1985, collected data during transient operation and repeated the variable load portion of the prior test to determine the response after the engine had been run in [2-5]. The operating history estimated from test engineers logs for EG1A and EG1B at the time of these tests are summarized in Tables 2.5 and 2.6. The results of these tests are discussed and compared with the results of the torsiograph test performed at SNPS [2-10] in the following sections.

The torsiograph response of the crankshaft at full load for each test is shown in Table 2.7. This comparison indicates that EG1A is better balanced now than at the time of the first torsiograph test. This is indicated by the lower half order response recorded during the second torsiograph test. The total response of the engine also shows a decrease. This decrease in response is typical of an engine after run-in. At the time of the second test EG1A had 120 hours of operation at 3500 kW or greater, 87 more than at the time of the first torsiograph test. The response of the engine during the second test is indicative of the future response of the engine. Torsiograph data presented in this section will be taken from the second torsiograph test.

2.2.1 Natural Frequencies

The frequency content of the torsional vibration signal indicated the torsional resonant frequency of the system was 38.1 Hz [2-4]. This value is in excellent agreement with TDI's computed value of 38.0 Hz.

2.2.2 Nominal Stresses at Full Load

The torsiograph response of the crankshaft at full load for significant orders along with the peak to peak amplitude is shown in Table 2.8. The amplitude of nominal shear stress may be estimated from the free-end vibration by assuming that the shaft is vibrating in its first mode. This assumption is typically used to estimate maximum stresses from torsiograph vibration data. However, a more accurate calculation is made in Section 3 in which the crankshaft stresses between each cylinder are determined and used in calculating the fatigue margin of the crankshaft. Assuming first mode response, the nominal shear stress in Crankpin No. 8 is 9351 psi per degree of free-end rotation. The amplitudes of nominal shear stress are presented in Table 2.8. The results indicate that the largest single order has a stress of 3348 psi which is well below the DEMA allowable of 5000 psi. The total stress of 6546 psi is also shown to be below the DEMA allowable of 7000 psi.

The measured response at 3500 kW is in general agreement with that calculated by TDI and shown in Table 2.4, although the measured values are somewhat higher than the calculated values.

A comparison of RBS and SNPS results indicates that the response is similar for the crankshafts as expected. The fourth order is the largest order for both crankshafts, and is slightly higher for RBS because its fourth order resonant speed is closer to the operating speed. The five and a half order is smaller at RBS because its resonant speed is further away from the operating speed than it is at SNPS. Even though the fifth order at RBS has a resonant speed of 455 rpm, its magnitude is still quite small and there are six other orders with greater magnitude. Thus, the fifth order response is not large enough to be of concern.

2.2.3 Nominal Stress at Reduced Loads

Torsiograph test data is typically obtained at about four or five different load levels. The relationship between torsional response and load may be obtained by plotting the torsiograph test data versus engine load.

The torsiograph test data for the variable load tests at RBS and SNPS are plotted in Figures 2-2 and 2-3. From these plots it was determined that a linear relationship between torsional response and load existed. This allows the determination of stresses at reduced loads based upon stresses at a 3500 kW load and the slope of the torsional response versus load curve. From the RBS data it was determined that there is a 7% change in torsional response for a 10% change in load. This relationship also holds for the SNPS torsiograph test data. To determine the torsional stress at a load L (kW), τ_L, from the torsional stress at 3500 kW, τ_{3500}, the following equation can be used

$$\tau_L = \tau_{3500} \left(0.3 + 0.7 \frac{L}{3500}\right)$$

Using this approach the stress levels at the 3130 kW load that produces stresses similar to those in the SNPS crankshaft during its endurance test were calculated. The values are tabulated in Table 2.9.

2.3 Conclusions

The following conclusions can be stated:

- The TDI Holzer calculations of the natural frequency of the crankshaft are in good agreement with test data.
- The TDI forced vibration calculations at 3500 kW are reasonably accurate and show that the single order stresses comply with DEMA.
- The torsiograph test data at 3500 kW shows that the combined order stresses comply with DEMA at the rated speed of 450 rpm.
- The torsiograph test data shows that there is a linear variation of crankshaft response with load.

Section 2 References

2-1 Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines. Diesel Engine Manufacturers Association, 6th ed., 1972.

2-2 Yang, Roland, "Torsional and Lateral Critical Speed Analysis: Engine Numbers 74039/42 Delaval-Enterprise Engine Model DSR-48 3500 KW/4889 BHP at 450 RPM for Stone & Webster Engineering Corp., Gulf States Utilities." Transamerica Delaval Inc., Engine and Compressor Division, Oakland, California, March 5, 1975, Revision - May 4, 1977.

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2-5 May, 1985, SWEC Torsiograph Test Report for River Bend Station.

2-6 Thomson, William T., Theory of Vibration with Applications. Second edition, Prentice-Hall, 1981.

2-7 Hartog, Den, Mechanical Vibrations. Third edition, McGraw-Hill, 1947.

2-8 Lloyd's Register of Shipping, Guidance Notes on Torsional Vibration Characteristics of Main and Auxiliary Oil Engines.

2-9 Craig, Roy R., Jr., Structural Dynamics: An Introduction to Computer Methods. Wiley, 1981.

2-10 Bercel, E., and Hall, J.R., "Field Test of Emergency Diesel Generator 103," Stone & Webster Engineering Corporation, April 1984.

TABLE 2.1

STIFFNESS AND INERTIAS FOR TDI HOLZER ANALYSIS

Inertia Location	Inertia (lb. ft. sec^2)	Stiffness (ft. lb./rad)
Front Gear	6.8	58.1×10^6
Cylinder No. 1	49.2	84.7×10^6
Cylinder No. 2	47.9	84.7×10^6
Cylinder No. 3	47.9	84.7×10^6
Cylinder No. 4	47.9	84.7×10^6
Cylinder No. 5	47.9	84.7×10^6
Cylinder No. 6	47.9	84.7×10^6
Cylinder No. 7	47.9	84.7×10^6
Cylinder No. 8	50.1	76.9×10^6
Flywheel	426.5 (1100.1)*	309.7×10^6 (276.8×10^6)
Generator	4976.1 (2650.4)	

* Where RBS and SNPS structural parameters differ, SNPS values are indicated in parentheses [2-3].

TABLE 2.2

TORSIONAL NATURAL FREQUENCIES FROM TDI ANALYSIS AT 3500 kW

Mode	Natural Frequency (Hz)	
	RBS [2-2]	SNPS [2-3]
1	38.0	38.7
2	107.0	92.9
3	146.5	116.7

TABLE 2.3

TORSIONAL LOADINGS FOR TDI ANALYSIS AT 3500 kW

Order	Torsional Loading, T_n (psi) [2-2]
1.5	129.5
2.5	71.7
3.5	42.8
4.0	27.7
4.5	23.8
5.0	17.4
5.5	12.8

TABLE 2.4

SINGLE-ORDER NOMINAL SHEAR STRESSES FROM TDI ANALYSIS AT 3500 kW

Order	Amplitude of Nominal Shear Stress (psi) RBS	SNPS
1.5	1599	1606
2.5	1069	1064
3.5	466	452
4.0	3321	2980
4.5	644	565
5.0	947	348
5.5	797	1080
DEMA Allowable for Single Order	5000	5000

TABLE 2.5

OPERATING HISTORY FOR EG1A

Load	Factory	7/15/84 Preservice Crankshaft Inspection	8/29/84 First Torsiograph Test	5/4/85 Second Torsiograph Test
>3500	4	Same as	4	4
3500	26	Factory	27.4	116
3130 < load ≤ 3500	--		--	67
2625 < load ≤ 3130	8		15.6	91
1750 < load < 2625	8		12.1	21
<1750	4		13.8	28
TOTAL	50	50	72.9	327

TABLE 2.6

OPERATING HISTORY FOR EG1B

Load	Factory	7/15/84 Preservice Crankshaft Inspection	8/29/84 First Torsiograph Test	5/4/85 Second Torsiograph Test
>3500	3	Same as	N/A	3
3500	32	Factory		93
3130 < load ≤ 3500	1			64
2625 < load ≤ 3130	--			61
1750 < load < 2625	4			24
<1750	1			30
TOTAL	41	41		275

TABLE 2.7

COMPARISON OF SWEC TORSIOGRAPH TEST RESULTS ON EG1A
AT 3500 kW LOAD

Order	Amplitude of Free End Rotation (degrees) from SWEC Torsiograph Data	
	1984 [2-4]	1985 [2-5]
0.5	0.128	0.093
1.0	0.002	0.004
1.5	0.174	0.177
2.0	0.001	0.001
2.5	0.132	0.132
3.0	0.001	0.001
3.5	0.055	0.056
4.0	0.365	0.358
4.5	0.071	0.069
5.0	0.080	0.068
5.5	0.086	0.086
6.0	0.007	0.007
Combined response $^1/_2$ peak to peak	0.757	0.700

TABLE 2.8

NOMINAL SHEAR STRESSES CALCULATED FROM SWEC TORSIOGRAPH TEST
AT 3500 kW LOAD

Order	Amplitude of Free-end Rotation (degrees)		Amplitude of Nominal Shear Stress (psi)*	
	RBS [2-5]	SNPS [2-10]	RBS	SNPS
0.5	0.093	0.056	870	535
1.0	0.004	0.005	37	48
1.5	0.177	0.171	1655	1635
2.0	0.001	0.001	9	10
2.5	0.132	0.130	1234	1243
3.0	0.001	0.001	9	10
3.5	0.056	0.058	524	555
4.0	0.358	0.325	3348	3108
4.5	0.069	0.064	645	612
5.0	0.068	0.034	636	325
5.5	0.086	0.127	804	1214
6.0	0.007	0.008	65	76
DEMA Allowabler for a Single Order			5000	5000
Combined Response $^1/_2$ peak to peak	0.700	0.693	6546	6626
DEMA Allowable for combined response $^1/_2$ peak to peak			7000	7000

* Amplitude of nominal shear stress is calculated to be 9351 psi per degree of free-end rotational amplitude for RBS [2-2] and 9562 psi per degree of free-end rotational amplitude for SNPS [2-3].

TABLE 2.9

RIVER BEND CRANKSHAFT STRESSES
AT 3130 kW

Order	Amplitude of Nominal Shear Stress (psi)
0.5	806
1.0	34
1.5	1533
2.0	8
2.5	1143
3.0	8
3.5	485
4.0	3100
4.5	597
5.0	589
5.5	745
6.0	60
Combined response $1/2$ Peak to Peak	6062

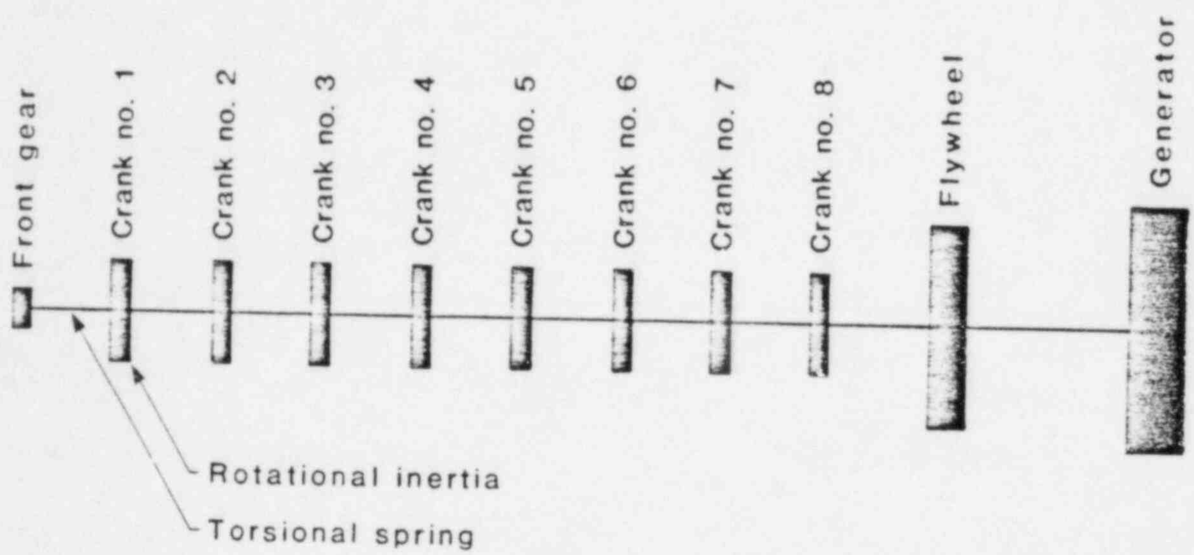

Figure 2-1. TDI dynamic model of crankshaft.

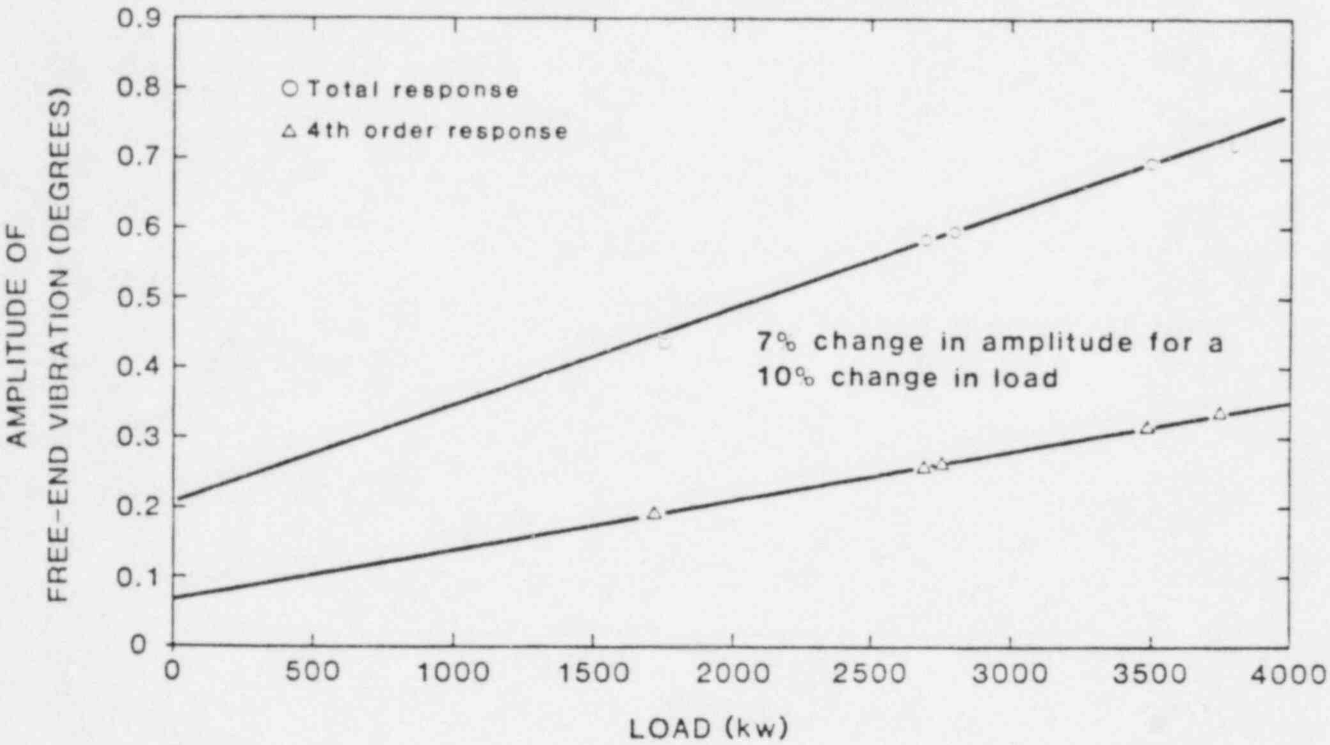

Figure 2-3. Shoreham torsiograph test data [2-10].

3.0 FATIGUE ANALYSIS OF CRANKSHAFT

In Section 2.0 it was found that the RBS crankshafts satisfy the DEMA nominal stress recommendations at rated speed for 3500 kW. The stresses for a single order were considerably below the 5000 psi that is recommended as an allowable. However, the stresses for combined orders were close to the 7000 psi that is recommended as an allowable. While the DEMA limits are believed to contain an intrinsic (though unspecified) safety margin, a fatigue analysis of the crankshaft was undertaken to determine the true margin.

First, a dynamic torsional analysis of the crankshaft is performed to determine the true range of torque at each crank throw. This model is compared with SWEC torsiograph test data for the amplitudes of free end vibration. The model may then be used to predict the nominal torsional stress over the speed range of 450 rpm ±5%.

Second, the maximum actual stresses in the crankpin fillet are determined from the dynamic strain gage test data from SNPS and the differences between the stresses at SNPS and RBS. The actual stresses in the oil holes are computed from the nominal torsional stress and the stress concentration factor.

Third, the fatigue endurance limit is established for the RBS 13-inch by 12-inch crankshafts by first obtaining the endurance limit for the failed 13-inch by 11-inch crankshafts at SNPS, and then assessing the differences between the two crankshafts. The endurance limit is compared with values provided in the literature.

Finally, a factor of safety against fatigue failure is computed.

3.1 Crankshaft Dynamic Torsional Analysis

3.1.1 Torsional Model

FaAA developed a dynamic torsional model of the crankshaft to overcome limitations in TDI's conventional forced vibration calculations. For instance, the TDI method does not compute the phase relationship between the various orders or modes, so it is not possible to compute the true summation. The actual maximum stress is a direct result of this summation. Furthermore, the TDI method always predicts maximum stress in Crankpin No. 8, which is generally true for a single order in the first mode but not true for the combined response of all orders and modes.

The dynamic model developed used the same idealized lumped inertia and torsional spring model as the TDI analysis (Figure 2-1 and Table 2.1) with one additional spring placed between the generator and ground to represent the effect of the grid on dynamic response during synchronous operation. This spring constant was found to be 2.053×10^6 ft.-lb./radian based on generator specifications. This constant is set close to zero to represent RBS emergency bus operation.

The first three torsional natural frequencies for the RBS crankshaft are shown in Table 3.1. The natural frequencies are in agreement with those computed by TDI and that measured by SWEC. There is also a rigid body mode which has a natural frequency of 3.0 Hz when the engine is connected to the grid, and is 0 Hz for RBS emergency bus operation.

When the diesel generator is running at a given speed and power level, the forced vibration problem is steady-state where both load and response repeat themselves every two revolutions of the crankshaft. To model the dynamic response, a modal superposition analysis [3-1] was used with harmonic load input. The calculation of the harmonic loads will be discussed in the next section.

3.1.2 Harmonic Loading

To calculate the harmonic loading on a crankshaft it is necessary to consider gas pressure, reciprocating inertia, and frictional loads. The gas pressure loading was obtained from pressure versus crank angle data. This pressure was measured at SNPS by SWEC [3-2]. The pressure was measured in Cylinder No. 7 by inserting a probe through the air start valve. A top dead center, TDC, mark for Cylinder No. 7 was simultaneously recorded by a probe on the flywheel. The pressure data at 3500 kW was reduced by FaAA to obtain the pressure curve shown in Figure 3-1.

The torque produced by this pressure may then be calculated as a function of crank angle. The mean value of this torque should be the torque required to produce 3500 kW divided by the mechanical efficiency. A mechanical efficiency of 1.0 was obtained, rather than the expected 0.88 to 0.90. The difference is probably explained by either the pressure measurements being too low or by the TDC being shifted. Peak pressures were measured in all the cylinders to ensure that all cylinders were balanced. Normally, the excess torque above that required to run the engine at 3500 kW is dissipated by friction.

The reciprocating mass of the connecting rod and piston was found to be approximately 820 lbs. This mass causes reciprocating inertia torque on the crankshaft. The effect of this torque was combined with the gas pressure torque.

The total torque was then decomposed into its sine and cosine harmonics corresponding to each order. These torque harmonics were used in the steady-state analysis. The magnitude of the torque harmonics are normalized by dividing by the piston area and throw radius. The resulting normalized torques for the most significant orders are shown in Table 3.2.

3.1.3 Comparison of Calculated Response With Test Data

The response due to the first 24 orders and all 11 modes is calculated using modal superposition with 2.5% of critical damping for each mode. The actual value of damping used has little effect on the response since the orders are not at resonance at 450 rpm.

The calculated amplitude of free-end displacement is compared to the SWEC test measurements at RBS [3-3] in Table 3.3. It is seen that the agreement is close for all significant orders. The vector summation listed represents half the maximum peak-to-peak displacement range. This confirms that the pressure versus crank angle data used provides a good representation of the pressure loading at RBS.

The model also calculates the range of torque at each crank throw, from which nominal shear stresses ($\tau = Tr/J$) are calculated and shown in Table 3.4. The maximum stress occurs between cylinder number 5 and cylinder number 6.

3.1.4 Calculation of Nominal Stresses at Offspeed Conditions

The dynamic torsional model of the crankshaft may be used to predict the stresses over the speed range 450 rpm ±5%. Over this speed range the maximum stress always occurs between cylinder number 5 and cylinder number 6. The maximum amplitudes of torsional stress are shown in Figure 3-2 and Table 3.5 for both RBS and SNPS at 3500 kW. It is seen that there is good agreement between the values computed by FaAA and those computed using Dr. Sarsten's computer program COMHOL2 [3-4]. At 450 rpm the stress at RBS is somewhat higher, 7357 psi, than that at SNPS, 7006 psi. At overspeed, 472.5 rpm, the stress at RBS is higher than SNPS since the fourth order critical speed is closer to the operating speed at RBS than it is at SNPS. On the other hand, at underspeed, 427.5 rpm, the stress at RBS is lower than SNPS because the five and a half order critical speed is further away at RBS than it is at SNPS.

3.1.5 Calculation of Nominal Stresses at Reduced Loads

In Section 2.0 it was shown that the torsional vibration of the crankshaft is linearly related to the engine load. Torsiograph data for both RBS and SNPS showed that this relationship leads to a 7% change in vibration amplitude for a 10% change in engine load. In addition, the dynamic strain gage test at SNPS [3-2] showed that the relationship between crankpin fillet stresses and load is also linear. Figure 3-3 indicates that the crankpin measured stress data shows an $8^1/_2$% change in stress for a 10% change in engine load.

This linear relationship may now be used to compute stresses at loads below 3500 kW. The stresses in Figure 3-4 and Table 3.6 are computed for a load of 3130 kW at RBS and 3300 kW at SNPS. The load of 3300 kW at SNPS corresponds to the load used for the 10^7 cycle entrance test on DG103 at

SNPS. The load of 3130 kW produces stresses at RBS that are comparable to stresses at 3300 kW at SNPS. These reduced stresses are calculated assuming a 7% stress reduction for a 10% load reduction. Using 7% leads to a comparison which gives relatively higher stresses for RBS than if $8^1/_2$% were used.

The stresses at 3130 kW computed in the above manner will be used to compute the fatigue margin of the crankshaft.

3.2 Calculation of Fillet and Oil Hole Stresses

The fillet of crankpin number 5 of a 13-inch by 12-inch crankshaft at SNPS was instrumented with strain gages and tested under operational conditions at 450 rpm and 3500 kW [3-2]. The fillet geometry of the SNPS crankshafts is the same as that of the RBS crankshafts as specified in TDI drawing 03-310-05-AC. Figure 3-5 shows typical structural dimensions for the crankshafts.

A dynamic model of the crankshaft confirms that crankpin number 5 undergoes the greatest stress. Three-dimensional finite element models of a quarter crank throw were used to determine the location of highest stress, both within the fillet and around the crankpin [3-5]. The strain gage rosette was placed in this highest stressed location.

The strain gage results from the SNPS test at 3500 kW are shown in Table 3.7. The calculation of principal stresses from the strain gage results are also shown.

To account for the simultaneous effects of shear and bending, the stress state is represented by an equivalent stress using Sine's method [3-6]. For a biaxial stress state, the equivalent alternating stress, S_{qa}, and equivalent mean stress, S_{qm}, are given by:

$$S_{qa} = (S_{a_1}^2 - S_{a_1}S_{a_2} + S_{a_2}^2)^{1/2}$$

and $$S_{qm} = S_{m_1} + S_{m_2}$$

where S_{a_1} and S_{a_2} are the alternating components of principal stress, and S_{m_1} and S_{m_1} are the mean components of principal stress. From the SNPS test at 3500 kW [3-2], the equivalent alternating stress, S_{qa}, and equivalent mean stress, S_{qm}, on crankpin number 5 were calculated to be:

$$S_{qa} = 24.6 \text{ ksi}$$
$$S_{qm} = 4.8 \text{ ksi}$$

Equivalent stresses, S_{qa} and S_{qm}, are those alternating and mean uniaxial stresses that can be expected to give the same life as the given multiaxial stresses.

To determine the actual fillet stresses at 3130 kW in the RBS crankshafts the value of S_{qa} needs to be factored by the differences in nominal torsional stress. Thus, the equivalent alternating stress at RBS at 3130 kW is given by

$$S_{qa} = 24.6 \left(\frac{6813}{7006}\right) = 23.9 \text{ ksi}$$

since the nominal stress at this load is 6813 psi and the nominal torsional stress at SNPS at 3500 kW is 7006 psi.

The stress in the oil holes may be calculated from the nominal torsional stress using a stress concentration factor of 4.16. This factor is based on theoretical solutions for a plate with a hole in shear [3-7], and a finite element model to account for the blend radius.

The nominal torsional stress in the main journals is computed from its adjacent crankpin by accounting for the difference in the radius of the journal. The maximum oil hole stress is determined to be in main journal number 6 and is 22.3 ksi.

3.3 Endurance Limit for Crankshafts

The endurance limit for the RBS crankshafts was based on an analysis of the failed 13-inch by 11-inch crankshafts at SNPS and confirmed by a review of the literature.

The number 5 crankpin fillet of a 13-inch by 11-inch crankshaft at SNPS, which had a fatigue crack, was instrumented [3-8]. The strain gage data and resulting principal stresses for this crankshaft at 3500 kW are shown in Table 3.8.

The equivalent alternating stress, S_{qa}, and equivalent mean stress, S_{qm}, were calculated to be:

$$S_{qa} = 33.7 \text{ ksi}$$
$$S_{qm} = 10.9 \text{ ksi}$$

At the time of the fatigue failure in DG102 the crankshaft had experienced at least 273 hours at equal to or greater than 3500 kW. Counting one stress cycle for every two revolutions of the crankshaft, this leads to about 4×10^6 cycles. The operating history for the three SNPS crankshafts are shown in Table 3.9.

A review of S-N data for material similar to the crankshaft indicates that a typical S-N curve is represented by the notched wrought 1040 data shown in Figure 3-6 [3-10]. This data was used to determine the shape of the S-N curve. The curve was then shifted on the vertical scale to match the data from the full scale crankshaft failures at SNPS. Using the shape of the curve in Figure 3-6, the 10^7 cycle endurance limit for this mean stress was determined to be 32.4 ksi. The ultimate tensile strength for these crankshafts averaged 96 ksi.

The crankshafts at RBS have a minimum ultimate tensile strength of 94 ksi. The endurance limit scales approximately linearly with ultimate tensile strength. On this basis, the endurance limit for the RBS 13-inch by 12-inch crankshafts is calculated from the SNPS 13-inch by 11-inch crankshafts and is shown on the Goodman diagram [3-11] in Figure 3-7.

Additionally, a review of the literature was conducted to find other fatigue data for large diameter crankshafts. Nishihara and Fukui [3-12] reported full scale fatigue test results for a large diameter crankshaft of a similar material, with an ultimate tensile strength of 94 ksi and found an endurance limit of 35 ksi. This is in excellent agreement with the value of 35.7 ksi at zero mean stress calculated by FaAA.

3.4 Factor of Safety Against Fatigue Failure

The factor of safety against fatigue failure for the RBS crankshafts at 3130 kW is computed to be 1.39 from the Goodman diagram in Figure 3-7. The stress point corresponding to crankshafts at SNPS at 3300 kW is also shown on the Figure.

The factor of safety computed above is for operation at 450 rpm. At offspeed conditions the ractor of safety may be adjusted to account for the difference in nominal torsional stresses. Thus, at 5% overspeed, the factor of safety would be approximately 9% less since nominal torsional stress is 9% greater. However, continuous operation at offspeed conditions is prevented by the control of the governor.

3.5 Conclusions

The following conclusions may be stated:

- The nominal torsional stresses in the RBS crankshafts at 3130 kW are approximately equal to those at SNPS at 3300 kW over the speed range. The stresses in the crankshafts at RBS are lower at underspeed and higher at overspeed than those at SNPS.
- The actual maximum crankpin fillet stresses at RBS at 3130 kW may be determined by scaling the strain gage test data from SNPS at 3500 kW.
- The endurance limit for the RBS crankshaft may be determined from the fatigue data from the 13-inch by 11-inch crankshafts at SNPS. This data is consistent with other fatigue data for large diameter crankshafts.
- The factor of safety against fatigue cracking of the RBS crankshafts at 3130 kW is 1.39. This factor of safety provides sufficient margin for safe operation at loads up to 3130 kW.

Section 3 References

3-1 Timoshenko, S., D.H. Young, and W. Weaver, Jr., Vibration Problems in Engineering. Fourth edition, Wiley, 1974.

3-2 Bercel, E., and Hall, J.R., "Field Test of Emergency Diesel Generator 103," Stone & Webster Engineering Corporation, April 1984.

3-3 May, 1985, SWEC Torsiograph Test Report for River Bend Station.

3-4 Nervik, N.R., and A. Sarsten, "COMHOL2 User's Manual," Division of Internal Combustion Engines and Marine Engineering, The Norwegian Institute of Technology.

3-5 "Evaluation of Emergency Diesel Generator Crankshafts at Shoreham and Grand Gulf Nuclear Power Stations," Failure Analysis Associates Report No. FaAA-84-3-16, May 22, 1984.

3-6 Fuchs, H.O., and Stephens, R.I., Metal Fatigue in Engineering. Wiley, 1980.

3-7 Peterson, R.E., Stress Concentration Factor. Wiley & Sons, New York, 1974.

3-8 Bercel, E., and Hall, J.R., "Field Test of Emergency Diesel Generator 101," Stone & Webster Engineering Corporation, October 1983.

3-9 "Emergency Diesel Generator Crankshaft Failure Investigation Shoreham Nuclear Power Station," Failure Analysis Associates Report No. FaAA-83-10-2.1, October 31, 1983.

3-10 Structural Alloys Handbook, Volume 1, Battelle's Columbus Laboratories, page 45, 1981 Edition.

3-11 Collins, J.A., Failure of Materials in Mechanical Design. Wiley, 1981.

3-12 Nishihara, M., and Fukui, Y., "Fatigue Properties of Full Scale Forged and Cast Steel Crankshafts," Transactions of the Institute of Marine Engineering. Series B on Component Design for Highly Pressure-Charged Diesel Engines, London, January 1976.

TABLE 3.1

NATURAL FREQUENCIES FOR DSR-48
13-INCH BY 12-INCH CRANKSHAFTS AT RBS

Mode	Natural Frequency (Hz)
1	38.0
2	107.0
3	146.5

TABLE 3.2

TORSIONAL LOADING FOR FaAA ANALYSIS AT 3500 kW

Order	Torsional Loading, T_n (psi)
1.5	112.0
2.5	77.0
3.5	48.0
4.0	33.0
4.5	26.2
5.0	21.2
5.5	15.5

TABLE 3.3

FREE-END VIBRATION LOAD FOR DSR-48
13-INCH BY 12-INCH CRANKSHAFT AT RBS AT 3500 kW

Order	Amplitude of Vibration (degrees)	
	FaAA Analysis	SWEC Test [3-3]
0.5	0.070	0.093
1.0	0.001	0.004
1.5	0.180	0.177
2.0	0.001	0.001
2.5	0.145	0.132
3.0	0.001	0.001
3.5	0.064	0.056
4.0	0.397	0.358
4.5	0.079	0.069
5.0	0.048	0.068
5.5	0.094	0.086
6.0	0.013	0.007
6.5	0.013	--
7.0	0.002	--
7.5	0.001	--
8.0	0.014	--
Vector Summation	0.733	0.700

TABLE 3.4

TORSIONAL STRESSES FOR DSR-48 13-INCH BY 12-INCH CRANKSHAFT AT RBS AT 3500 kW

Location	Amplitude of Nominal Shear Stress (psi)	
	4th Order	Total
Between Cylinder No. 1 and Cylinder No. 2	716	3068
Between Cylinder No. 2 and Cylinder No. 3	1347	3293
Between Cylinder No. 3 and Cylinder No. 4	1951	4842
Between Cylinder No. 4 and Cylinder No. 5	2516	5666
Between Cylinder No. 5 and Cylinder No. 6	3030	7357
Between Cylinder No. 6 and Cylinder No. 7	3484	6191
Between Cylinder No. 7 and Cylinder No. 8	3867	5977
Between Cylinder No. 8 and Flywheel	4176	5903

TABLE 3.5

COMPARISON OF SHOREHAM AND RIVER BEND CRANKSHAFT
STRESSES AT 3500 kW

Engine Speed (rpm)		Max Amplitude of Nominal Torsional Stress (psi)			
		SNPS		RBS	
FaAA	Sarsten	FaAA	Sarsten	FaAA	Sarsten
427.5	428.0	7203	7050	7098	7077
436.0	437.0	7072	--	6999	7074
445.0	444.0	6962	--	7169	7206
450.0	450.0	7006	7090	7357	7427
455.0	454.0	7137	--	7574	7606
464.0	464.0	7500	--	7779	7893
472.5	473.0	7694	7850	8059	8172

TABLE 3-6

COMPARISON OF SHOREHAM AND RIVER BEND CRANKSHAFTS STRESSES AT REDUCED LOADS

Engine Speed (rpm)	Max Amplitude of Nominal Torsional Stress (psi)	
	SNPS at 3300 kW	RBS at 3130 kW
427.5	6915	6573
436.0	6789	6481
445.0	6684	6638
450.0	6726	6813
455.0	6852	7014
464.0	7200	7203
472.5	7386	7463

TABLE 3.7

STRAIN GAGE RESULTS FROM SNPS TEST AT 3500 kW
WITH 13-INCH BY 12-INCH CRANKSHAFTS [3-2]

Strain Gage	Maximum	Minimum
5-1 (Compression)	-195 με	288 με
5-2 (Bending)	695 με	-410 με
5-3 (Tension)	737 με	-610 με
Major Principal Stress	26.2 ksi	4.9 ksi
Minor Principal Stress	-2.9 ksi	-18.7 ksi

TABLE 3.8

STRAIN GAGE RESULTS FROM SNPS TEST AT 3500 kW
WITH 13-INCH BY 11-INCH CRANKSHAFTS [3-8]

Strain Gage	Maximum	Minimum
5-1 (Compression)	1118 $\mu\varepsilon$	-707 $\mu\varepsilon$
5-2 (Bending)	773 $\mu\varepsilon$	-459 $\mu\varepsilon$
5-3 (Tension)	-389 $\mu\varepsilon$	266 $\mu\varepsilon$
Major Principal Stress	35.4 ksi	12.5 ksi
Minor Principal Stress	-4.2 ksi	-22.0 ksi

TABLE 3.9

OPERATING SUMMARY FOR SNPS EMERGENCY DIESEL GENERATORS WITH 13-INCH BY 11-INCH CRANKSHAFTS [3-9]

	DG101 S/N 74010	DG102 S/N 74011	DG103 S/N 74012
TDI Factory Test Hours	128 *	30	40
110%	16	19	20
100%	180	254	249
> 75%	260	323	427
> 50%	51	75	77
Unknown (< 100%)	11	17	5
Total hours	646	718	818
Number of Starts*	536	266	236
Hot Restarts	5	5	2

* Engine S/N 74010 was used to qualify the R-48 product line for nuclear standby service and, therefore, experienced more factory test hours, including 300 starts.

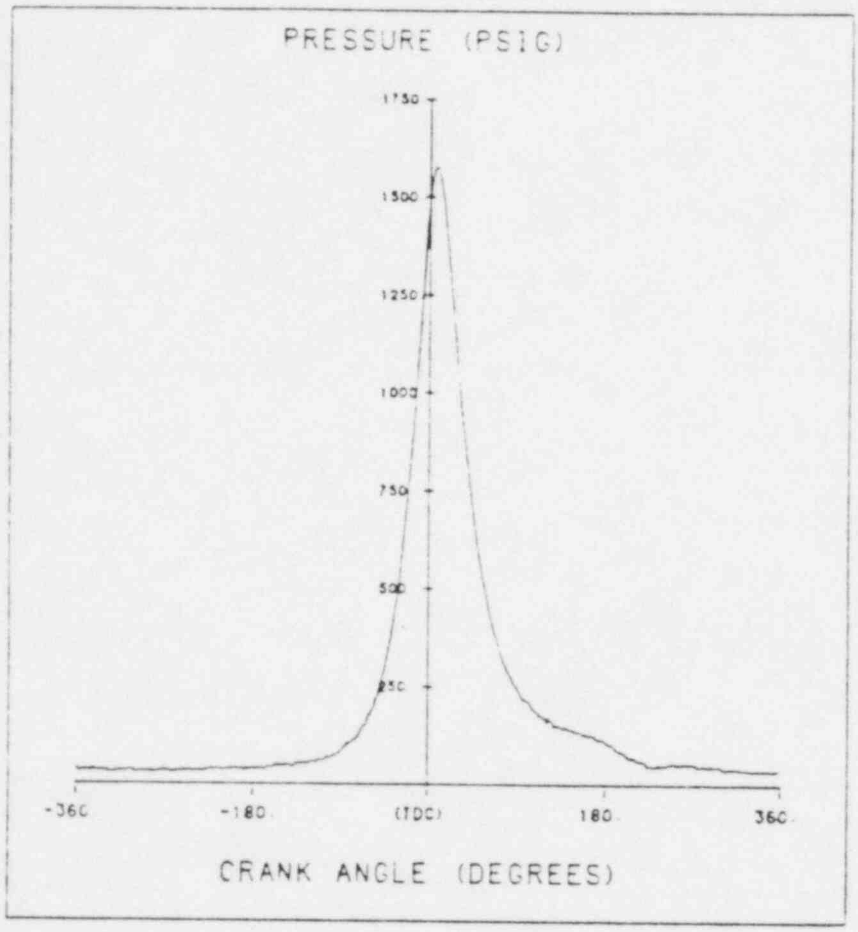

Figure 3-1. Measured pressure versus crank angle for SNPS at 3500 kw load.

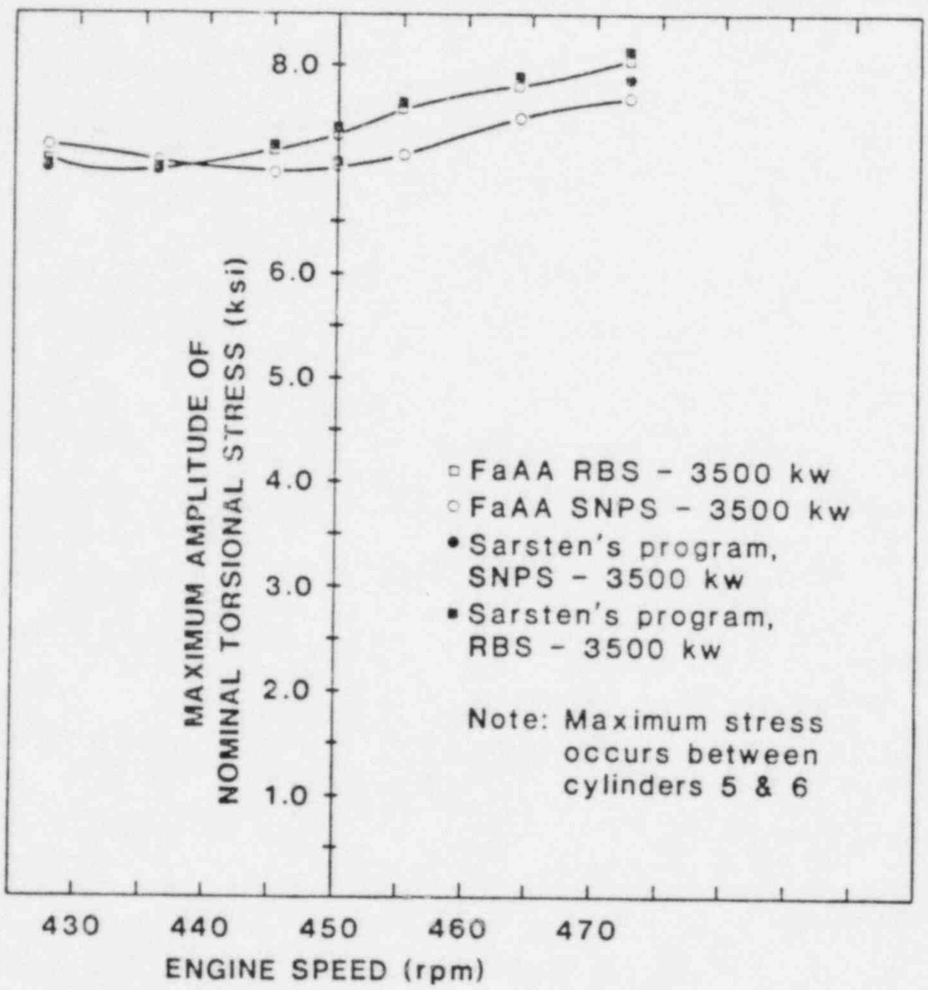

Figure 3-2. Comparison of RBS and SNPS nominal stress over the speed range of 450 rpm ±5%.

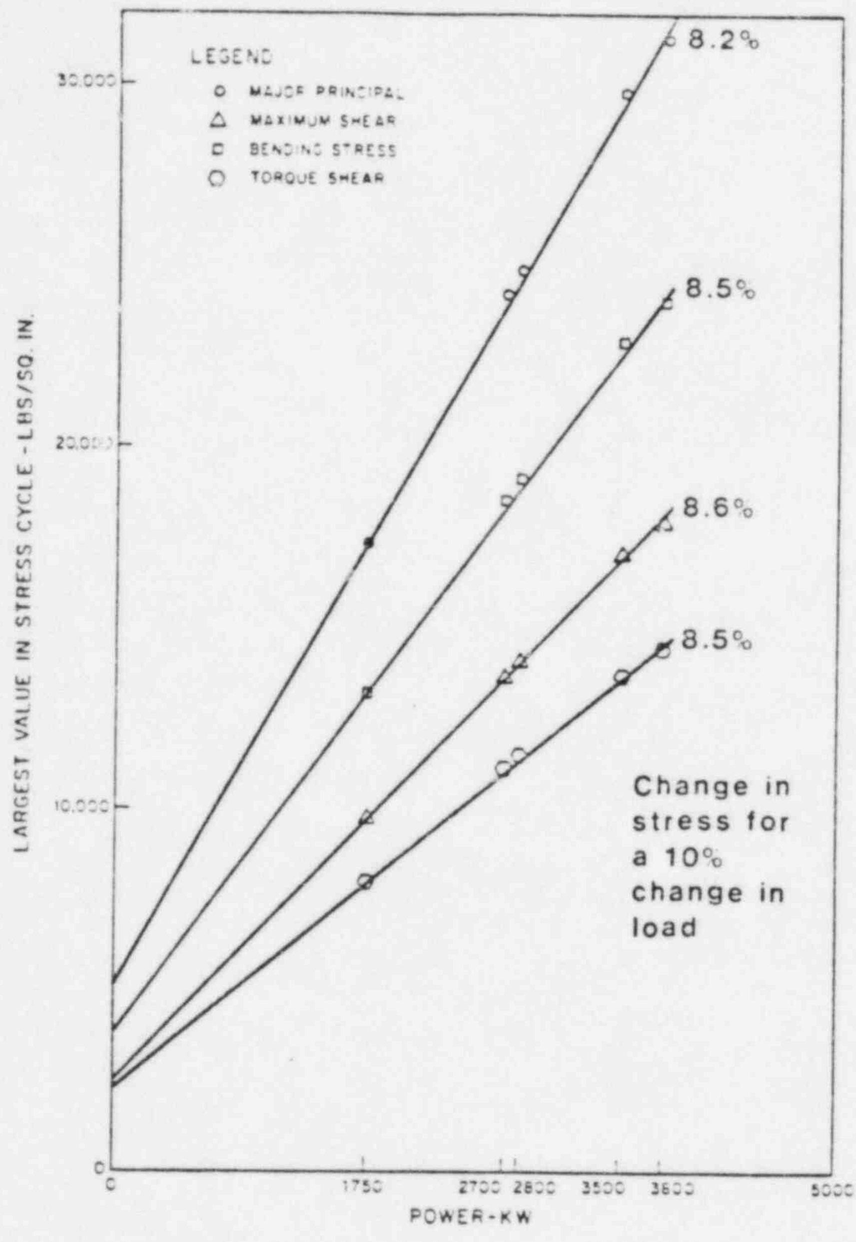

Figure 3-3. Measured stresses as a function of load for SNPS.

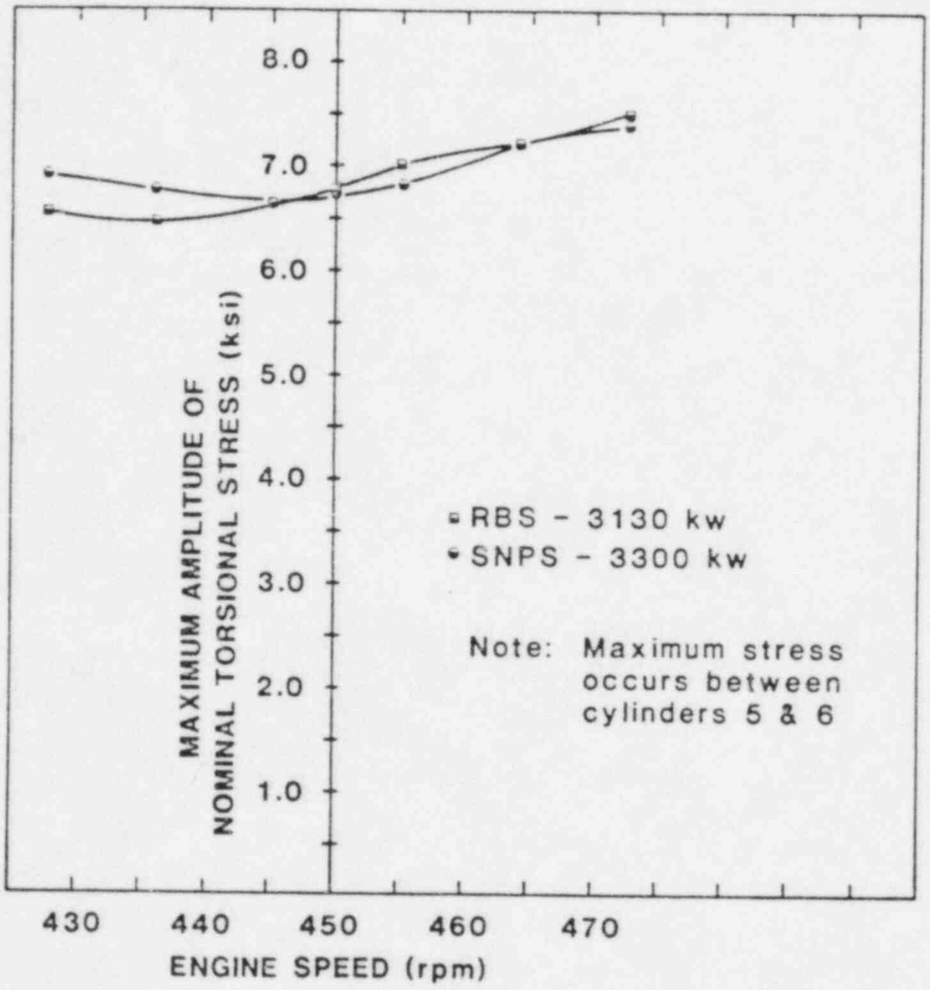

Figure 3-4. Comparison of RBS and SNPS nominal torsional stresses at reduced loads over the speed range of 450 rpm ±5%.

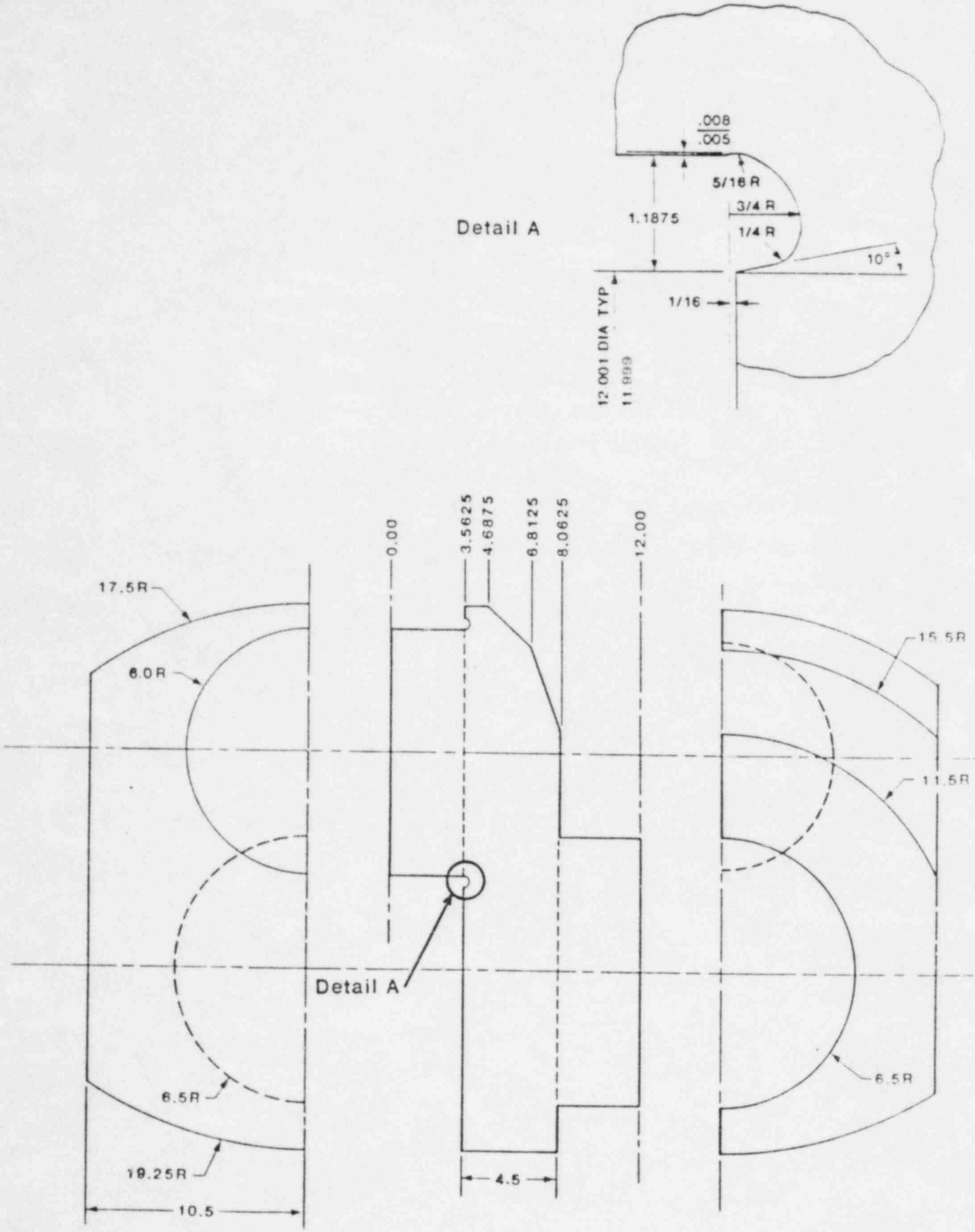

Figure 3-5. Typical structural dimensions of the RBS crankshafts.

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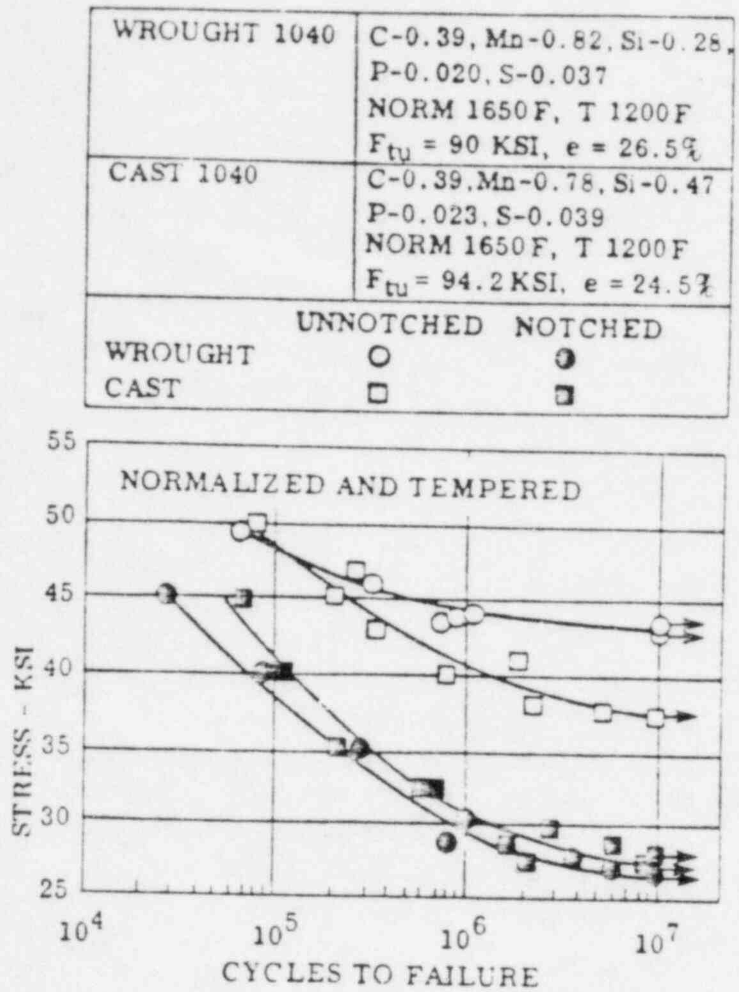

Figure 3-6. Endurance limit for 1040 steel.

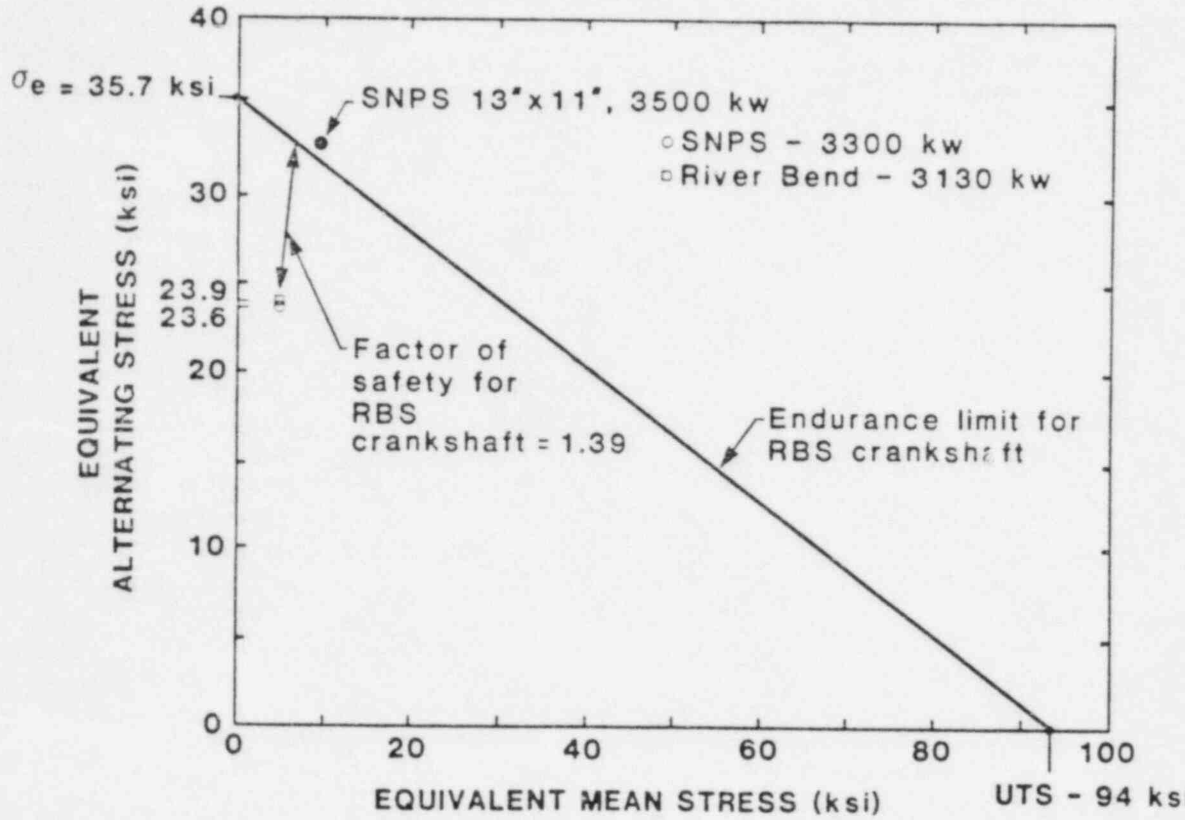

Figure 3-7. Goodman diagram for River Bend crankshafts.

4.0 CRANKSHAFT ENDURANCE TESTING

The NRC Safety Evaluation Report on the TDI Diesel Generator Owners Group Program Plan, [4-1] provides that where TDI diesels are to be operated at brake mean effective pressures exceeding 185 psig, the applicant should submit information demonstrating that components, which are of the same design, have operated successfully for at least ten million load cycles which meet or exceed the maximum emergency service load requirements for the subject engines. The following comparison can be made with an endurance test performed on one of the SNPS crankshafts. Alternatively, a design may be qualified by testing a prototype to 10^7 cycles at the rated load where additional data is available to demonstrate an adequate fatigue margin. In this case the endurance test is being used to provide additional assurance of the adequacy of the design even though the test by itself would not represent sufficient assurance.

A 10^7 cycle endurance test was run on a DSR-48 engine crankshaft with 13-inch main journals and 12-inch crankpin journals at a load of 3300 kW* at SNPS. The crankpin fillet and oil hole geometry is identical for the crankshafts at RBS and those at SNPS. The crankshafts were made according to TDI drawing number 03-310-05-AC.

While the crankshafts are of the same design for the two stations, the flywheel and generator are somewhat different. This difference leads to a small change in the torsional stresses for the crankshafts as discussed in previous sections of this report. To account for these design differences the load at RBS that produces the same torsional stress as a load of 3300 kW at SNPS may be calculated. This calculation is based on the linearity of stress with load which has been established in Sections 2.0 and 3.0. Based on a 7% change in stress for a 10% change in load, a load of 3071 kW at RBS produces the same stress as a load of 3300 kW at SNPS, as shown in Figure 4-1.

* The endurance test has been stated to be nominally at 3300 kW. However, Dr. Pischinger (FEV) performed a cumulative damage analysis of the crankshaft based on the actual loads experienced during the 745 hour endurance test and found the load to be equivalent to 3505 kW [4-2].

The material specifications for the crankshafts at RBS and SNPS were the same. Crankshafts at both plants were required to meet ABS Grade 4 which has a minimum ultimate tensile strength, UTS, of 83000 psi. The UTS measurements for RBS [4-3, 4-4] and SNPS [4-5, 4-6, 4-7] are shown in Table 4.1. The lowest measurement at RBS is 94000 psi and at SNPS is 100800 psi. The crankshafts at both stations meet the minimum requirements of ABS Grade 4 and have UTS values within the expected range for this material. The lowest measurement at RBS is about 7% lower than the lowest measurement at SNPS.

The fatigue limit for these crankshafts is approximately proportional to the UTS of the material. Thus, the factor of safety in the SNPS crankshafts at 3300 kW is approximately 7% higher than in the RBS crankshafts at 3071 kW. However, this effect is small in comparison to the factor of safety in the RBS crankshafts.

The crankpin fillets at SNPS are shot peened whereas those at RBS are not. While no credit was taken for the effect of shot peening in computing fatigue margins, it is possible to estimate the maximum effect that it could have had on the crankshaft endurance test. The stresses in the most highly stressed main journal oil hole is computed to be about 7% less than that of the most highly stressed crankpin fillet. Thus, if shot peening has more than a 7% effect on the crankpin fillets, the oil holes will govern crankshaft life. This 7% effect is quite small when compared to the safety margin.

The endurance testing at SNPS along with RBS actual experience are plotted against the crankshaft S-N curve in Figure 4-2. It is seen that for the RBS crankshafts at 3071 kW, the stresses are significantly below the actual SNPS endurance test of 3500 kW. The margin between the RBS stresses at 3071 kW and the S-N curve for the RBS crankshaft corresponding to the lowest UTS measurement is adequate to ensure reliable operation of the RBS crankshafts for their intended service.

4.1 Conclusions

The following conclusions may be stated:

- The crankshafts at RBS are geometrically identical to those at SNPS
- At 450 rpm the torsional stresses at RBS at 3071 kW are equal to those at SNPS at 3300 kW
- Although the lowest UTS at RBS is lower than that at SNPS, the material specifications are the same and the material meets the requirements of ABS Grade 4.
- The advantage of shot peening is limited to about 7% since the oil holes are not shot peened.

Section 4 References

4-1 U.S. NRC Safety Evaluation Report, Transamerica Delaval, Inc. Diesel Generator Owners Group Program Plan, August 13, 1984.

4-2 U.S. NRC Atomic Safety and Licensing Board hearing on Shoreham Nuclear Power Station, Docket No. 50-322-OL, Tr. 28419-20.

4-3 American Bureau of Shipping, Report on Castings and Forgings for the crankshaft in engine 74039, 4-6-80.

4-4 American Bureau of Shipping, Report on Castings and Forgings for the crankshaft in engine 74040, 31-7-80.

4-5 Krupp Stahl, Schmiede Und Bearbeitung for the crankshaft in engine 74010, 17-10-83.

4-6 American Bureau of Shipping, Report on Castings or Forgings for the crankshaft in engine 74011, February 4, 1983.

4-7 American Bureau of Shipping, Report on Castings or Forgings for the crankshaft in engine 74012, February 4, 1983.

TABLE 4.1

COMPARISON OF RBS AND SNPS ULTIMATE TENSILE STRENGTHS

Station	Designation	Engine Serial No.	Ultimate Tensile Strength (psi)
RBS	EG1A	74039	94000 97000
RBS	EG1B	74040	99500 101000
SNPS	DG101	74010	100800 100800
SNPS	DG102	74011	101800 106500
SNPS	DG103*	74012	100800 106500
ABS Grade 4 Requirement			83000

* SNPS 10^7 cycle endurance test at loads equal to or greater than 3300 kW was performed on this engine.

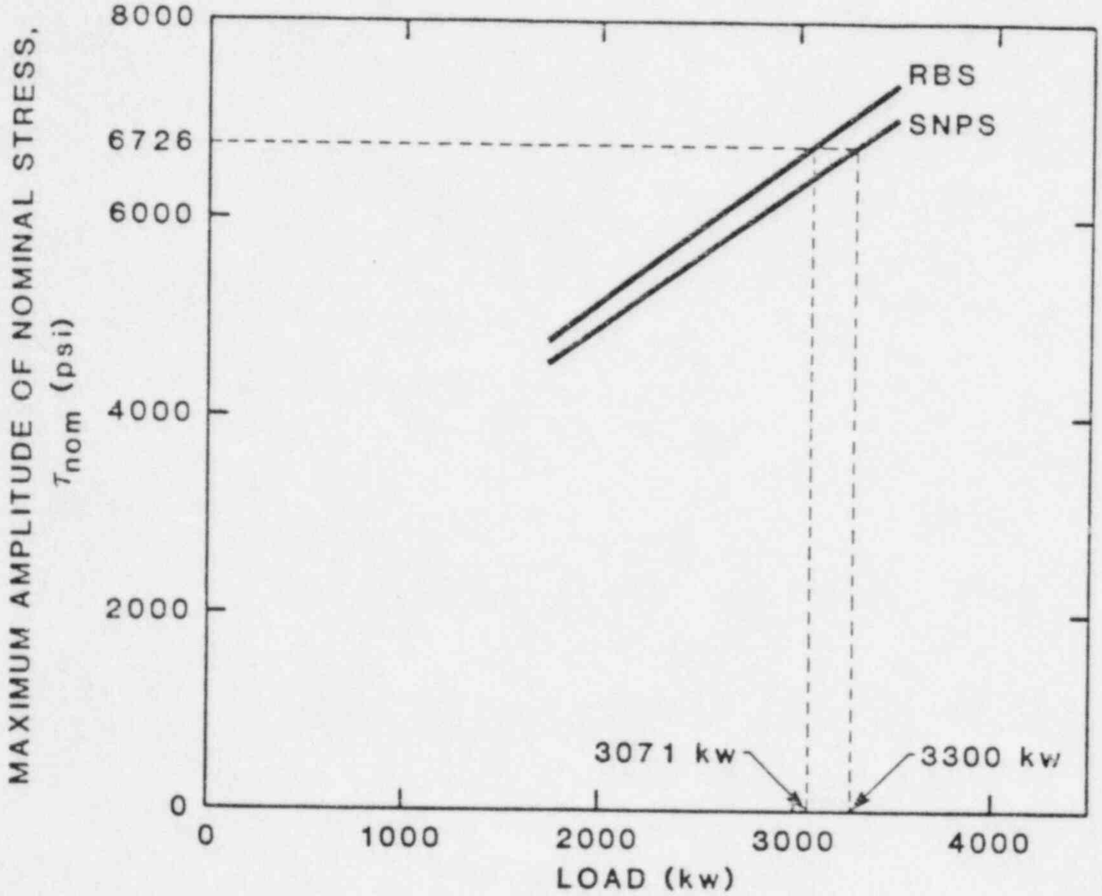

Figure 4-1. Loads at RBS and SNPS that result in equal crankshaft stresses.

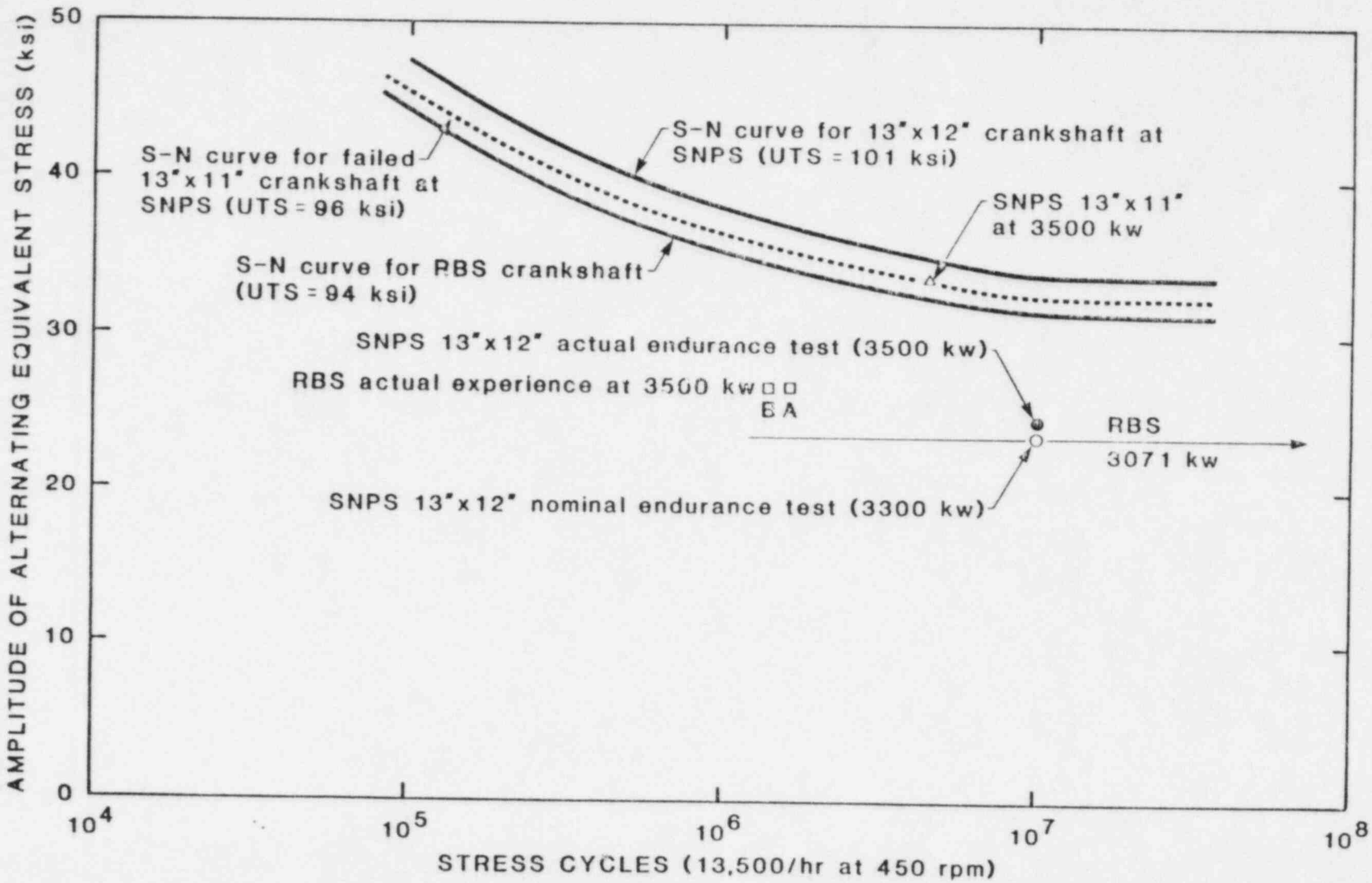

Figure 4-2. Comparison of RBS and SNPS crankshaft stresses and experience with S-N curve.

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5.0 CRANKSHAFT FILLET AND OIL HOLE INSPECTIONS

Crankpin journal and main journal oil holes and crankpin fillets on diesel engines EG1A and EG1B at RBS were examined to identify any crack indications. The inspections were performed by FaAA personnel in July 1984. The operating history of engines EG1A and EG1B prior to these inspections were summarized in Tables 2.5 and 2.6. The results of the inspections are discussed below. The recommended acceptance criteria for eddy current inspections are given in References 5-1 and 5-2.

5.1 Crankpin Fillet Eddy Current Examination

Eddy current examinations were performed on the crankpin fillets on cylinders No. 5 governor end and cylinders no. 6 through 8 governor and generator ends on both engines [5-3]. FaAA NDE 11.6 Revision 0 eddy current test procedure was used. The eddy current recording criterion was any crack-like indication exceeding 25 percent of the signal obtained from the $^1/_{16}$ inch long by $^1/_{32}$ inch deep simulated crack in FaAA crack standard PA07395-831228. No recordable indications were observed.

5.2 Crankshaft Oil Hole Liquid Penetrant Examination

Liquid penetrant examinations were performed on the main journal oil hole radius of Journals No. 7 through No. 9 and on the crankpin oil hole radius of crankpins No. 5 through No. 8 in both diesel engines [5-4]. FaAA NDE 6.4 Revision 0 liquid penetrant examination was used. No linear (crack-like) indications were observed.

5.3 Crankshaft Oil Hole Eddy Current Examination

Eddy-current examinations were performed on main journal oil holes in journals No. 7 through No. 9 and crankpin journal oil holes in crankpins No. 5 through No. 9 in both engines [5-5]. The crankshaft oil holes were inspected to a depth of 3.0 inches. The acceptance criteria for the individual oil holes is specified in Reference 5-2. Main Journal No. 10 was not inspected since it is not a through-depth oil hole.

FaAA NDE 11.3 Revision 0 eddy current procedure was used. The crack reference (FaAA PA07395-74073) is a simulated oil hole machined in a segment of crankshaft and containing a simulated crack of 0.040 inch long by 0.020 inch deep.

Eddy current indications were observed exceeding the recording threshold at various depths in all of the oil holes inspected. An interpretation of the results of this test was made utilizing the most recent FaAA eddy current recording criteria for oil holes (NDE 11.3, Revision 2). Using this procedure only 5 of the 28 oil holes inspected had recordable indication [5-2]. The recommended oil hole acceptance criteria is detailed by cylinder location and broad ranges of depth from the journal surface (0" to 1", 1" to 2", and 2" to 3"). Further evaluation of the allowable notch sizes for the five oil holes at the depth at which the threshold readings were exceeded was made [5-5]. The results indicated that for these oil holes the readings are within the allowable criteria.

5.4 Conclusions

The following conclusions can be stated:

- The eddy current inspection of the fillets on EG1A and EG1B showed no recordable indications were present.
- The results of the liquid penetrant inspections of the main journal and crankpin journal oil hole radii indicated that no crack-like indications were observed.
- A review of the results of the eddy current inspection of the main journal oil holes on EG1A and EG1B indicates that all readings are within the acceptable criteria.

References

5-1 Memorandum to John C. Kammeyer, DR/QR Program Manager from Paul R. Johnston, Crankshaft Design Review Task Leader, dated 7/26/84, re: Oil Hole Inspections for DSR-48 Crankshafts at River Bend.

5-2 Memorandum to Duane Johnson, FaAA NDE Manager, from Paul R. Johnston, Crankshaft Design Review Task Leader, dated 7/9/84, re: Inspection of DSR-48 Crankshafts of River Bend.

5-3 FaAA Eddy Current Examination 1 Report Nos. 840710-101, 840710-201, 840711-202, 840711-203, dated July, 1984.

5-4 FaAA Liquid Penetrant Examination Report Nos. 840715-603, 840715-604 dated July, 1984.

5-5 Hand Calculation "Review of NDE Crankshaft Inspection Reports," FaAA-SP-85-5-10.