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EVALUATION OF EMERGENCY DIESEL GENERATOR CRANKSHAFTS AT SHOREHAM AND GRAND GULF NUCLEAR POWER STATIONS

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The report is final, pending confirmatory reviews
required by FaAA's QA operating procedures.

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STATEMENT OF APPLICABILITY

This report addresses the structural integrity of the crankshafts in Transamerica Delaval Inc. DSR-48 engines at the Shoreham Nuclear Power Station and DSRV-16-4 engines at the Grand Gulf Nuclear Power Station. In view of possible differences in generators, flywheels, and engine operating conditions, the results may not necessarily apply to other engines of the same model. These plant-specific differences, where they exist, will be evaluated in separate reports.

EXECUTIVE SUMMARY

DSR-48 13-INCH BY 12-INCH CRANKSHAFTS AT SHOREHAM NUCLEAR POWER STATION

The structural integrity of the replacement 13-inch by 12-inch diameter crankshafts installed in the emergency diesel generators at the Shoreham Nuclear Power Station has been extensively evaluated by testing and analysis. Conventional analytical techniques typically utilized by the diesel engine industry show that 13-inch by 12-inch crankshafts comply with DEMA requirements. Angular displacements of the free end of the crankshaft, stress ranges in the most highly stressed crank pin fillets, and the range of output torque at the flywheel were measured at and above full-rated load. The torsiograph measurements of twist showed that the crankshafts meet the DEMA requirements. In addition, the strain gage measurements of maximum bending and torsional stress and calculations of maximum stress by a modal superposition analysis showed that the crankshafts have a factor of safety in fatigue of 1.48 without taking into account any benefit of shot peening the crank pin fillets. The factor of safety was determined from the measured endurance limit of the original 13-inch by 11-inch crankshafts that cracked in fatigue. The measured shaft response was in close agreement with that predicted by the modal superposition analysis.

The replacement crankshafts are suitable for unlimited operation at the rated load and speed in the emergency diesel generators at SNPS.

DSRV-16-4 13-INCH BY 13-INCH CRANKSHAFTS AT GRAND GULF NUCLEAR STATION

The structural integrity of the 13-inch by 13-inch diameter crankshafts installed in the emergency diesel generators at the Grand Gulf Nuclear Station has been evaluated by testing and analysis. Conventional analytical techniques typically utilized by the diesel engine industry show that 13-inch by 13-inch diameter crankshafts comply with DEMA requirements. Angular displacements of the free end of the crankshaft were measured at and above full-rated load at TDI. The torsiograph measurements of twist taken during factory tests showed that the crankshafts meet the DEMA requirements. The measured shaft response was in close agreement with that predicted by the modal superposition analysis.

The DSRV-16-4 crankshaft is sensitive to operating speed and the balance of cylinder firing. Torsiograph tests of several engines should be conducted to determine the range of crankshaft response permitted by TDI specified balance limits and the governor characteristics.

The oil holes in the main journals numbers 4, 6, and 8 are more critical in torsion than are the crankpin fillets and should be inspected.

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Task No. 03310A

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PART A:

REVIEW OF DSR-48
13-INCH BY 12-INCH CRANKSHAFT

1.0 INTRODUCTION TO REVIEW OF DSR-48 13-INCH BY 12-INCH CRANKSHAFT

As a result of fatigue damage in the crankshafts of three emergency diesel generator sets at Shoreham Nuclear Power Station, replacement crankshafts of current design have been installed. The principal difference is an increase in crankpin diameter from 11 inches to 12 inches. This report presents Failure Analysis Associates' findings on the adequacy of the replacement crankshafts in the emergency diesel engines at Shoreham Nuclear Power Station.

A detailed investigation of the orginal crankshaft, which attributed failure to high cycle torsional fatigue resulting from inadequate design, was previously conducted by Failure Analysis Associates (FaAA) [1-1]. An analysis of the replacement crankshafts, conducted prior to dynamic testing, was also performed by FaAA [1-2].

The installation of the replacement crankshaft was required to meet the recommendations of the Diesel Engine Manufacturers Association (DEMA). In Section 2.0, the torsional calculations of Transamerica Delaval Inc. (TDI) [1-3] and the torsiograph test results of Stone & Webster Engineering Corporation (SWEC) [1-4] are reviewed for compliance with the DEMA stress allowables.

In Section 3.0, a detailed analysis of the factor of safety against fatigue failure is performed. A torsional dynamic analysis is used to compute nominal torsional stresses at each crank throw. A three-dimensional finite element analysis of a quarter section of a crank throw is then performed to obtain the local stresses in the crankpin fillet. The computed stresses are compared to dynamic strain gage measurements to verify the models. In turn, the models are used to verify that strain gages have been placed in locations of maximum stress. Finally, the measured stresses are used to compute a factor of safety against fatigue failure for the replacement crankshafts. This is accomplished by comparing the measured stresses with the endurance limit for the replacement crankshafts.

Section 1 References

1-1 "Emergency Diesel Generator Crankshaft Failure Investigation, Shoreham Nuclear Power Station," Failure Analysis Associates Report No. FaAA-83-10-2.1, October 31, 1983.

1-2 "Analysis of the Replacement Crankshafts for Emergency Diesel Generators, Shoreham Nuclear Power Station," Failure Analysis Associates Report No. FaAA-83-10-2.2, October 31, 1983.

1-3 Yang, Roland, "Proposed Torsional and Lateral Critical Speed Analysis: Engine Numbers 74010/12 Delaval-Enterprise Engine Model DSR-48 3500 KW/4889 BHP at 450 RPM". Transamerica Delaval Inc., Engine and Compressor Division, Oakland, California, August 22, 1983.

1-4 Bercel, E., and Hall, J.R., "Field Test of Emergency Diesel Generator 103," Stone & Webster Engineering Corporation, March 1984.

2.0 COMPLIANCE OF CRANKSHAFT WITH DIESEL ENGINE MANUFACTURERS ASSOCIATION 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 August, 1983, Transamerica Delaval Inc. (TDI) performed a torsional critical speed analysis of the replacement crankshafts [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. The inappropriate T_n values employed in the original analysis of the 13-inch by 11-inch crankshaft were replaced with the correct values for this analysis. In January, 1984, Stone & Webster Engineering Corporation, SWEC, conducted a torsiograph test on a replacement crankshaft at Shoreham Nuclear Power Station [2-3]. 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 level 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 [2-4]. This method has been used for at least 40 years [2-5], 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 38.7 Hz, which produces 4th order resonance at 581 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 signifcant 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 [2-6]. It is found that TDI's values are higher than LRS's values for low orders and lower for high orders. However, the largest single order was measured to be within 5% of those computed using TDI's values of T_n. 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 [2-3]. Since the engine runs at 450 rpm and the 4th order critical speed is 580 rpm, the calculated response at resonance will not be further considered.

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 [2-7] 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 2980 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 cómpared with the allowable in the next section.

2.2 Review of Stone & Webster Engineering Corporation Torsiograph Test

Torsiograph tests are commonly used to confirm torsional vibrational calculations. The 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.

2.2.1 Natural Frequencies

The frequency content of the torsional vibration signal at 450 rpm showed a resonance at 38.6 Hz. This value is in excellent agreement with TDI's computed value of 38.7 Hz.

2.2.2 Nominal Stresses

The torsiograph provides the angular displacement response of the free end of the crankshaft. This displacement may be decomposed into components corresponding to each order. The peak-to-peak response may also be obtained.

The nominal shear stress, τ, in Crankpin No. 8 may be established from the amplitude of free-end vibration by assuming the shaft is vibrating in the first mode. The nominal shear stress is then found to be 9562 psi per degree of free-end vibration from the TDI analysis [2-2].

SWEC tabulated the single order and peak-to-peak response for both 3500 kW (100% of rated load) and for 3800 kW (109% of rated load). These values have been factored to obtain nominal shear stresses and are shown in Table 2.5. The results at 100% load show that the largest single order has a stress of 3108 psi which is well below the DEMA allowable of 5000 psi. The total stress of 6626 psi is also shown to be below the DEMA allowable of 7000 psi.

At 3800 kW the stresses of 3242 psi for a single order and 6875 psi for combined response are also lower than 5000 psi and 7000 psi respectively. At 3900 kW the corresponding stresses are 3[illegible]87 psi and 6958 psi by linear extrapolation. However, the 3900 kW level is a two-hour overload rating at which the engine is not required to operate continuously.

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

2.3 Nominal Stresses for Underspeed and Overspeed Conditions

Strict interpretation of DEMA regulations requires the consideration of torsional stresses at conditions other than operating speed. During normal standby diesel generator testing at SNPS the units are synchronized to the Long Island power distribution grid to simulate full load and two-hour overload conditions. At this time the frequency characteristics of the grid assure that the speed and associated frequency vary less than ± 1 Hz or ± 1.6% speed.

During other testing and potentially during a LOOP/LOCA event, the unit speed is controlled by the Woodward Governor. Testing at the SNPS site, during which step changes in load were produced by starting or stopping various pumps, revealed the largest variations in speed to be -3% to +2% associated with increasing or decreasing load respectively. These step changes were the same order of magnitude of those calculated to occur during a LOOP/LOCA. The time lag associated with the unit's ability to return to 450 rpm was likewise found to be less than 3 seconds.

Since speed variations associated with load step changes cannot be produced at full or two-hour overload power conditions due to grid connection, the modal superposition method was used to calculate the effects. The free-end vibration amplitude was first calculated by the modal superposition method, and then the nominal torsional stress was calculated to be 9562 psi per degree of free-end rotation [2-2]. The T_n values used in the modal superposition analysis were assumed to be equal to those obtained at 3500 kW and 450 rpm. The maximum nominal torsional stresses at 428 rpm (95% rated speed) and also at 473 rpm (105% rated speed) have been calculated to equal the DEMA limit of 7000 psi within ±3%, which reflects the uncertainty in the T_n values at these speeds. Thus, within the accuracy of the analysis, compliance with DEMA is obtained. Furthermore, the very small potential time during which such conditions could actually occur with the demonstrated performance of the governor precludes fatigue damage.

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, "Proposed Torsional and Lateral Critical Speed Analysis: Engine Numbers 74010/12 Delaval-Enterprise Engine Model DSR-48 3500 KW/4889 BHP at 450 RPM." Transamerica Delaval Inc., Engine and Compressor Division, Oakland, California, August 22, 1983.

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

2-4 Thomson, William T., Theory of Vibration with Applications. Second edition, Prentice-Hall, 1981.

2-5 Hartog, Den, Mechanical Vibrations. Third edition, McGraw-Hill, 1947.

2-6 Lloyd's Register of Shipping, Guidance Notes on Torsional Vibration Characteristics of Main and Auxillary Oil Engines.

2-7 Craig, Roy R., Jr., Structural Dynamics: An Introduction to Computer Methods. Wiley, 1981.

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	1100.1	276.8×10^6
Generator	2650.4	

TABLE 2.2

TORSIONAL NATURAL FREQUENCIES FROM TDI ANALYSIS

Mode	Natural Frequency (Hz)
1	38.7
2	92.9
3	116.7

TABLE 2.3

TORSIONAL LOADINGS FOR TDI ANALYSIS

Order	Torsional Loading, T_n (psi)
1.5	129.5
2.5	71.7
3.5	42.8
4.0	27.7
4.5	23.8
5.5	12.8

TABLE 2.4

SINGLE-ORDER NOMINAL SHEAR STRESSES FROM TDI ANALYSIS

Order	Amplitude of Nominal Shear Stress (psi)
1.5	1606
2.5	1064
3.5	452
4.0	2980
4.5	565
5.5	1080
DEMA Allowable for Single Order	5000

TABLE 2.5

NOMINAL SHEAR STRESSES CALCULATED FROM SWEC TORSIOGRAPH TEST

Order	Amplitude of free-end rotation (degrees)		Amplitude of Nominal Shear Stress (psi)*	
	At 3500 kW	At 3800 kW	At 3500 kW	At 3800 kW
1.5	0.171	0.187	1635	1788
2.5	0.130	0.140	1243	1339
3.5	0.058	0.061	555	584
4.0	0.325	0.339	3108	3242
4.5	0.064	0.067	612	643
5.5	0.127	0.136	1214	1300
DEMA Allowable for a Single Order			5000	5000
½ peak to peak	0.693	0.719	6626	6875
DEMA Allowable ½ peak to peak			7000	7000

* Amplitude of nominal shear stress is calculated to be 9562 psi per degree of free-end rotational amplitude.

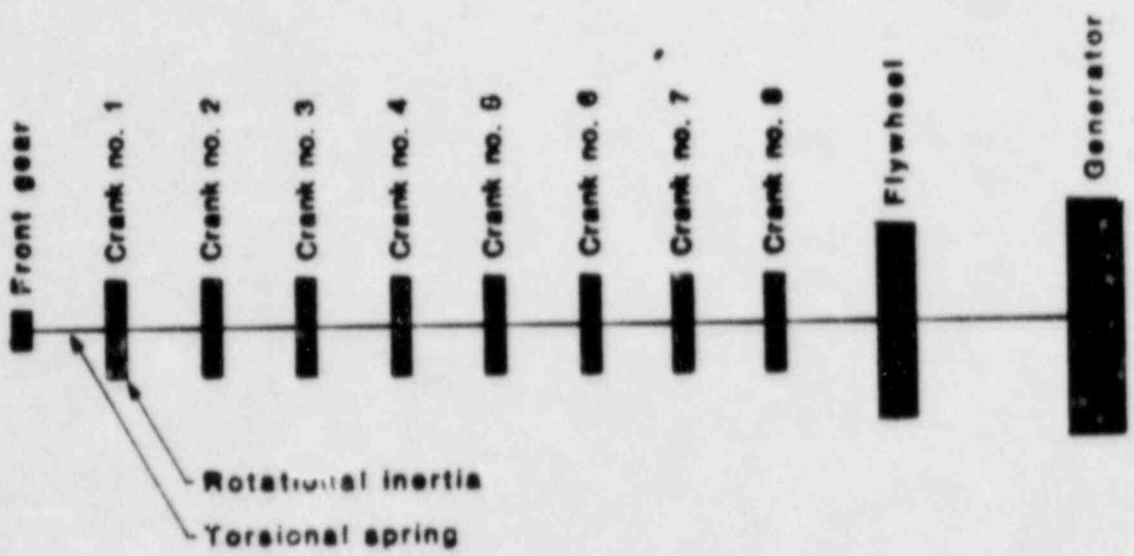


Figure 2-1. TDI dynamic model.

3.0 FATIGUE ANALYSIS OF CRANKSHAFT

In Section 2.0 it was found that the replacement crankshafts satisfy the DEMA nominal stress recommendations for both 3500 kW and 3900 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 quite 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 test data for the amplitudes of free end vibration, measured with the torsiograph, and for range of torque near the flywheel, measured with strain gages on the shaft.

Second, a finite element model of a one quarter crank throw is used to compute the local stresses in the fillet region. Torsional and gas pressure loading cases are considered. The results of this analysis are compared with strain gage test results, which were measured in the fillets of Crank Throw Nos. 5 and 7.

Third, the fatigue endurance limit is established for the replacement crankshaft by first obtaining the endurance limit for the failed crankshafts, and then assessing the differences between the failed and replacement 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 1.409×10^6 ft.-lb./radian based on generator specifications. This constant is set close to zero to represent SNPS emergency bus operation.

The first five torsional natural frequencies for the replacement crankshaft are shown in Table 3.1. The first natural frequency was found to be 2.93 Hz due to the connection to the grid. For operation on the SNPS emergency bus the first natural frequency is 0 Hz (rigid body mode). The other natural frequencies are in agreement with those computed by TDI and measured by SWEC.

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 model 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 <u>Harmonic Loading</u>

To calculate the harmonic loading on a crankshaft it is necessary to consider gas pressure, reciprocating inertia, and frictional loads. The gas pressure loading may be obtained from pressure versus crank angle data. This pressure was measured in the SWEC test [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 100% load 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. 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. In this case, because the pressure curve produced the correct power without friction, friction was not applied. The effects of pressure being too low and not applying friction are expected to largely cancel each other.

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 SWEC test report stated that the measured damping in the system was 2.6% [3-2].

The calculated amplitude of free-end displacement is compared to the SWEC test measurements 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.

The model also calculates the range of torque at each crank throw, which is shown along with the corresponding nominal shear stress ($\tau = Tr/J$) in Table 3.4. The computed torque range near the flywheel was found to be 312 ft-kips compared with the measured value of 357 ft-kips [3-2]. The strain gages were placed close to the flywheel hub, and thus were expected to give higher values. The apparent stress concentration factor is 1.14. The computed torque as a function of crank angle for each crank throw is shown in Figure 3-2. The computed and measured torques at the flywheel are shown as a function of crank angle in Figure 3-3.

3.2 Crankshaft Stress Analysis

3.2.1 Finite Element Model

The nominal crankshaft stress values calculated from the dynamic model are considerably less than the actual maximum stresses in the crankshaft. Those nominal values would prevail if the crankshaft were a long circular cylinder. Stresses in the real crankshaft are greatly influenced by its complex geometry and by stress concentrations, especially at the fillet radii between the main journal and web and the crankpin and web. In this section, maximum stresses and their location are determined with particular attention to the crankpin fillet.

The multi-throw crankshaft under investigation consists of a series of crankpins and main journals interconnected through webs. Typical structural dimensions of one throw are shown in Figure 3-4. The main journal is 13.0 inches in diameter; the crankpin is 12.0 inches in diameter with a web thickness of 4.5 inches. Fillet details are also shown in Figure 3-4.

The following material properties, corresponding to the AISI 1042 crankshaft steel, were used in the analyses:

Young's Modulus: $E = 30.0 \times 10^6$ psi

Poisson's Ratio: $\nu = 0.3$

A crankshaft throw is subjected to loads of two basic types: (1) torque transmitted through the throw, which is influenced by the output power level and by the torsional vibration response of the crankshaft and (2) connecting rod forces applied to the crankpin and reacted at bearing supports.

Linear elastic analyses were performed using the computer program MARC, K.1-1 Version, from MARC Analysis Research Corporation. Generation of the geometric input data and the post-processing graphics was performed using PATRAN-G and PATMAR developed by PDA Engineering. One throw of the crankshaft was analyzed by applying a static unit twist on the main journal. Since all throws are geometrically identical, a single model with appropriate loads and boundary conditions can be used to represent approximately any throw. Three existing planes of local symmetry were employed in the analysis to keep the finite element model to a feasible size without compromising the accuracy of the results. These planes of symmetry are shown schematically in Figure 3-5 along with the coordinate system. The first plan of symmetry is the vertical plane passing through mid sections of the crankpin, the web, and the main journal. The second and third planes of symmetry are orthogonal to the first at the mid distance between two adjacent webs in the crankpin and the main journal, respectively. Thus, only the portion of the crankpin, the web, and the main journal contained within these planes of symmetry was modeled.

This model uses eight node, three-dimensional, isoparametric brick elements with linear interpolation, capable of modeling an arbitrarily distorted cube. Each node in an element has three translational degrees of freedom. Because the state of stress in the vicinity of the fillet is of greatest interest and stress gradients are highest there, a finer mesh was used in this region. Figure 3-6 shows the three-dimensional model, with node and element numbers omitted for clarity, along with the coordinate system adopted in the model.

As adjustment to account for mesh refinement was obtained by comparing finite element stresses for a step shaft with data reported from Peterson [3-3]. It was found that a factor of 1.08 needs to be applied to the finite element stresses.

3.2.2 Stresses Due to Torsional Loading

There is no set of boundary conditions that can be applied to the model that will represent exactly the physical crankshaft under torsional loading. Therefore, two separate sets of boundary conditions (Table 3.5) were analyzed. Boundary conditions for Case 1 represent antisymmetric behavior of the main journal to torsional loading in the axial (x) direction and those for Case 2, symmetric behavior. For both cases, transmitted torque was simulated by applying a unit rotation about the axis of the main journal in the third plane of symmetry of Figure 3-5.

Figure 3-7 illustrates the relative crank throw orientations best approximated by each of the two boundary condition cases. Those crankpin fillets adjacent to a throw on the same side of the main journal and in the same plane are best represented by Case 1 boundary conditions. Those adjacent to a throw on the opposite side of the main journal and in the same plane are more closely approximated by the Case 2 boundary conditions. Stresses in fillets not represented by either of these situations (i.e., adjacent to a throw not in the same plane) will fall between the two cases considered.

Stresses obtained from applying the unit torsional rotation were scaled to represent maximum positive and negative torques of 251,600 and -144,600 ft.-lbs. at Cylinder No. 5. These stresses were then scaled by a factor of 1.08 to account for the slight finite element underprediction of the fillet stresses, due to the size of the elements used.

From the eight element integration points, stresses were extrapolated to the surface. For Case 1, Figure 3-8 shows the circumferential variation, and Figure 3-9 shows the axial variation of maximum principal stress for both the peak-positive and peak-negative torque conditions. Figures 3-10 and 3-11 show similar variation for Case 2.

All stress values that have been presented are for the positive z side of the crankshaft, as viewed in Figure 3-4. In the crankpin fillet, this has also been designated as the 0° to 180° portion.

3.2.3 Stresses Due to Gas Pressure Loading

Near TDC the pressure in a cylinder causes a high vertical load on the crankpin. This load may be calculated from the pressure loading and reciprocating inertial loading. The pressure load is calculated from the area of the piston and peak pressure of 1680 psi. The reciprocating inertial load is obtained from the 820 lbs. of reciprocating weight and peak acceleration of 74.1 g. The reciprocating inertial load subtracts from the pressure load at TDC.

The pressure loading was applied to the model as a distributed load on the topmost three lines of noded points on the crankpin. Two types of boundary conditions were applied. In both cases symmetry planes 1 and 2 (see Figure 3-5) were modeled by symmetric boundary conditions. In the first case the third plane was modeled as a fixed support, and in the second case it was modeled as a pinned support. The actual moment in the main journal is greater than zero (pinned support) but less than the fixed-end moment (fixed support). The moment in the main journal may be estimated by treating the crankshaft as a continuous beam with simple supports at the main bearing locations. From this analysis it was determined that the moment in the main journal was 0.63 times the fixed-end moment. Since stresses for the fixed-end case were very small, the stresses due to the vertical loading were calculated as 0.37 times the stresses for the simply supported case. The maximum stress occurs in the 180° location and was found to be 15.5 ksi. The distribution of stress around the crankpin is shown in Figure 3.12.

3.2.4 Comparison of Stresses with Test Data

The SWEC test [3-2] recorded data from strain gages in the fillets of Crank Throw Nos. 5 and 7. These gages were placed in the locations where stresses are a maximum due to torsional loading. The measured stresses are compared with those calculated by the finite element model in Tables 3.6 and

3.7. Good agreement is found between the test data and computed results. The maximum principal stress range of 44.9 ksi was measured in Crank Throw No. 5 and it is bounded by the two finite element results of Case 1 and Case 2. At Cra k Throw No. 7, the measured stress is slightly higher than the computed stress from Cases 1 and 2.

The finite element stresses for vertical loading are in agreement with stresses measured in TDI's static test [3-4] on an inline 6 cylinder 13-inch by 11-inch crankshaft. TDI determined the maximum stress due to vertical loading, after factoring for the difference in crankpin area, to be 16.3 ksi at the 180° location. At this location, the torsional stresses are less than half of their maximum values. Also, at the location of maximum torsional stresses, the vertical bending stresses are measured to be 7.8 ksi and computed to be 8.1 ksi. At the No. 5 location, the transmitted torque is quite low during firing (see Figure 3-2), and thus, the highest stresses are not affected by vertical bending stresses.

3.3 Crankshaft Fatigue Failure Margin

The factor of safety against fatigue failure in the replacement (12-inch crankpins) crankshafts is calculated in this section. The stress levels in the replacement crankshafts are computed from strain gage test data. The endurance limit is first established for the failed crankshafts (11-inch crankpins) from strain gage test data. This endurance limit is then scaled to account for the higher ultimate tensile strength of the replacement crankshaft. The effect of shot peening the replacement crankshafts provides an additional margin against fatigue failure.

3.3.1 Stresses in Replacement Crankshafts

The replacement crankshaft was instrumented with strain gages in the fillet locations of Crankpin Nos. 5 and 7 and tested under operational conditions at 3500 kW (100% rated load) and 450 rpm (100% rated speed). The highest stresses were measured in Crankpin No. 5. A dynamic model of the crankshaft confirms that this pin undergoes the greatest range of torque. Three-dimensional finite element models of a quarter crank throw show that the strain gage rosette was placed in the location of highest stress, both within the fillet and around the crankpin. The following strains were measured at 3500 kW:

Strain Gage	Maximum	Minimum
5-1 (Compression)	-195με	288με
5-2 (Bending)	695με	-410με
5-3 (Tension)	737με	-610με

To account for the simultaneous effects of shear and bending, the stress state is represented by equivalent stresses using Sine's method [3-5]. 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_2} are the mean components of principal stress. From the test report [3-2], the equivalent alternating stress, S_{qa}, and equivalent mean stress, S_{qm}, on Crankpin No. 5 were calculated to be:

S_{qa} = 24.6 ksi
S_{qm} = 4.8 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.

3.3.2 Endurance Limit for Original 13-Inch by 11-Inch Crankshaft

The original 13-inch by 11-inch crankshaft was instrumented with strain gages in the fillet location of Crankpin No. 5. This fillet had previously experienced a fatigue crack during performance testing. After the test, the three-dimensional finite element models of a quarter section of a crank throw showed that the strain gage location was placed close to the location of maximum stress. The measured stress range is used to establish the endurance limit in this analysis as a conservative assumption, although the actual maximum stress range is revealed by the finite element model to be about 15% higher at a nearby location. From the test report [3-6], the following strains were measured at 3500 kW:

Strain Gage	Maximum	Minimum
5-1 (Tension)	1118$\mu\epsilon$	-707$\mu\epsilon$
5-2 (Bending)	773$\mu\epsilon$	-459$\mu\epsilon$
5-3 (Compression)	-389$\mu\epsilon$	266$\mu\epsilon$

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

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

From the test logs, it was determined that the shaft had experienced 273 hours at equal to or greater than 100% load, or about 4×10^6 cycles. By using Miner's rule and typical slopes of S-N curves, it was determined that the endurance limit for this mean stress was 32.4 ksi. The ultimate tensile strength for these crankshafts averaged 96 ksi. A line representing this endurance limit is shown on the Goodman diagram [3-7] in Figure 3-13.

This line is bounded by two lines showing the endurance limit for full scale crankshafts based on other test data [3-8].

3.3.3 Endurance Limit for Replacement Crankshafts

The replacement crankshafts have a minimum tested ultimate tensile strength of 103 ksi. The endurance limit scales linearly with ultimate tensile strength. On this basis, the endurance limit for the replacement crankshafts is shown in Figure 3-13.

The fillet regions of the replacement crankshafts have been shot peened. The effect of shot peening will produce increases in fatigue endurance limit [3-9].

3.3.4 Factor of Safety Against Fatigue Failure

The factor of safety against fatigue failure of the replacement crankshafts is 1.48 when the effect of shot peening is not considered.

At 3800 kW, the strain gage test data [3-2] on the replacement crankshaft shows that the stress level is 4% greater than it is at 3500 kW. At 3900 kW it would be about 5% greater than it is at 3500 kW. Thus, there is an adequate safety margin against fatigue failure at the specified diesel generator set two-hour-per-24-hour period rating of 3900 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 Peterson, R.E., Stress Concentration Factor. Wiley & Sons, New York, 1974.

3-4 "R-48 Crank Crankshaft Stress Analysis," Transamerica Delaval Inc. Report No. CR-01-1983.

3-5 Fuchs, H.O., and Stephens, R.I., Metal Fatigue in Engineering. Wiley, 1980.

3-6 Bercel, E., and Hall, J.R., "Field Test of Emergency Diesel Generator 101," Stone & Webster Engineering Corporation, October 1983.

3-7 Collins, J.A., Failure of Materials in Mechanical Design. Wiley, 1981.

3-8 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.

3-9 Burrell, N.K., "Controlled Shot Peening to Produce Residual Compressive Stress and Improved Fatigue Life," Proceedings of a Conference on Residual Stress for Designers and Metallurgists, American Society for Metals, April 1980.

TABLE 3.1

NATURAL FREQUENCIES FOR DSR-48
13-INCH BY 12-INCH CRANKSHAFT

Mode	Natural Frequency (Hz)
1	2.93*
2	38.73
3	92.94
4	116.67
5	184.33

* For SNPS emergency bus operation the natural frequency of the first mode is zero (i.e., rigid body mode), and the natural frequencies of the higher modes are not significantly altered.

TABLE 3.2

TORSIONAL LOADING FOR FaAA ANALYSIS

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
5.5	15.5

TABLE 3.3

FREE-END VIBRATION AT 100% LOAD FOR
DSR-48 13-INCH BY 12-INCH CRANKSHAFT

Order	Amplitude of Vibration (degrees)	
	FaAA Analysis	SWEC Test [3-2]
0.5	0.065	0.056
1.0	0.001	0.005
1.5	0.177	0.171
2.0	0.000	0.001
2.5	0.142	0.130
3.0	0.001	0.001
3.5	0.061	0.058
4.0	0.340	0.325
4.5	0.069	0.064
5.0	0.031	0.034
5.5	0.122	0.127
6.0	0.014	0.008
6.5	0.014	0.016
7.0	0.002	0.002
7.5	0.001	--
8.0	0.015	--
Vector Summation	0.662	0.693

TABLE 3.4

TORQUE RANGE AT 100% LOAD FOR DSR-48 13-INCH BY 12-INCH CRANKSHAFT

Location	Torque Range (ft. lbs.)		Amplitude of Nominal Shear Stress (psi)	
	4th Order	Total	4th Order	Total
Between Cylinder No. 1 and Cylinder No. 2	36.6×10^3	167.1×10^3	648	2955
Between Cylinder No. 2 and Cylinder No. 3	69.0×10^3	184.5×10^3	1220	3263
Between Cylinder No. 3 and Cylinder No. 4	100.0×10^3	271.1×10^3	1768	4794
Between Cylinder No. 4 and Cylinder No. 5	129.0×10^3	309.8×10^3	2282	5478
Between Cylinder No. 5 and Cylinder No. 6	155.6×10^3	396.2×10^3	2752	7006
Between Cylinder No. 6 and Cylinder No. 7	178.8×10^3	327.3×10^3	3162	5788
Between Cylinder No. 7 and Cylinder No. 8	198.6×10^3	329.7×10^3	3512	5830
Between Cylinder No. 8 and Flywheel	214.2×10^3	311.8×10^3*	3792	5514

* SWEC test [3-2] computed the torque range to be 357.1×10^3 ft.-lb. This indicates a stress concentration factor of 1.145 due to the proximity of the gage to the flywheel hub.

TABLE 3.5

DISPLACEMENT BOUNDARY CONDITIONS FOR TORSIONAL LOADING
(REFER TO FIGURE 3-5)

Case 1

Symmetry Plane	**Nodal Degrees of Freedom**		
	X	**Y**	**Z**
1	Fixed	Fixed	Free
2	Free	Fixed	Fixed
3	Free	Prescribed*	Prescribed*

Case 2

Symmetry Plane	**Nodal Degrees of Freedom**		
	X	**Y**	**Z**
1	Fixed	Fixed	Free
2	Free	Fixed	Fixed
3	Fixed	Prescribed*	Prescribed*

* Prescribed displacements were used on symmetry plane 3 to simulate torsional load on the main journal.

Table 3.6

COMPARISON BETWEEN FINITE ELEMENT MODEL TORSIONAL LOADING RESULTS AND TEST RESULTS FOR PIN NUMBER 5

	Peak Positive Torque[1] Principal Stresses (ksi)		Peak Negative Torque[2] Principal Stresses (ksi)		Range of Principal Stress (ksi)	Range of Equivalent Stress (ksi)
	σ_1	σ_2	σ_1	σ_2		
Finite Element Case 1	20.7	-18.0	10.4	-11.9	32.6	52.9
Finite Element Case 2	29.2	1.2	-0.7	-16.8	46.0	45.1
Strain Gage [4]	26.2	-2.9	4.9	-18.7	44.9	49.3

1 Peak positive torque = 251.6×10^3 ft.-lb.

2 Peak negative torque = -144.6×10^3 ft.-lb.

Table 3.7

COMPARISON BETWEEN FINITE ELEMENT MODEL TORSIONAL LOADING RESULTS AND TEST RESULTS FOR PIN NUMBER 7

	Peak Positive Torque[1] Principal Stresses (ksi)		Peak Negative Torque[2] Principal Stresses (ksi)		Range of Principal Stress (ksi)	Range of Equivalent Stress (ksi)
	σ_1	σ_2	σ_1	σ_2		
Finite Element Case 1	18.7	-16.3	7.3	-8.3	27.6	43.9
Finite Element Case 2	26.5	1.1	-0.5	-11.8	38.3	37.5
Strain Gage [4]	23.4	-8.9	2.8	-14.1	37.5	44.5

[1] Peak positive torque = 251.6×10^3 ft.-lb.

[2] Peak negative torque = -144.6×10^3 ft.-lb.

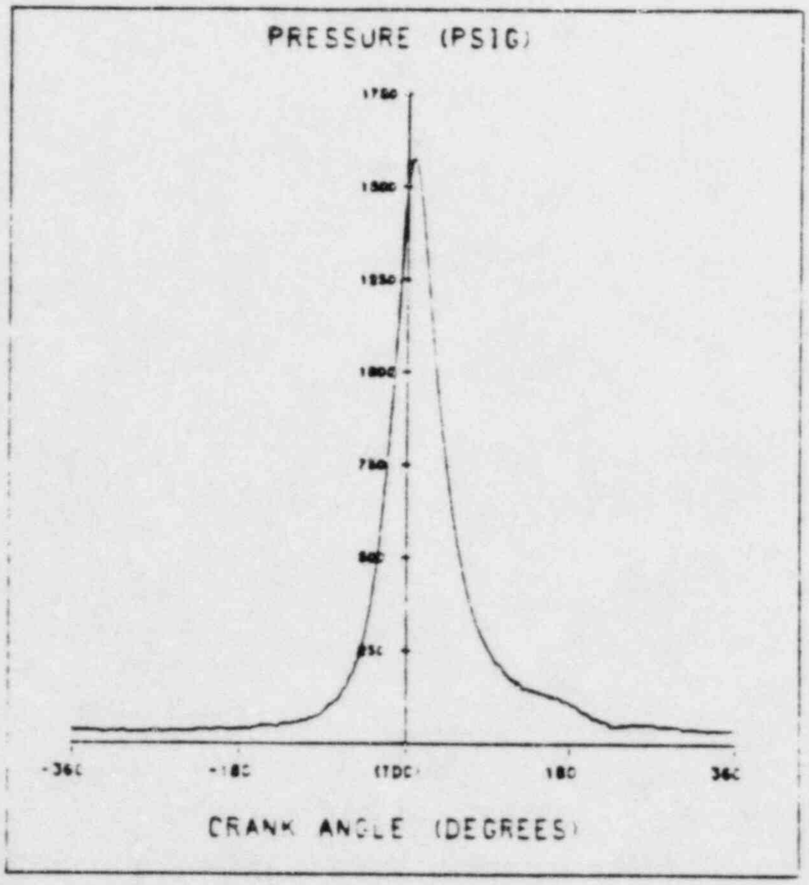
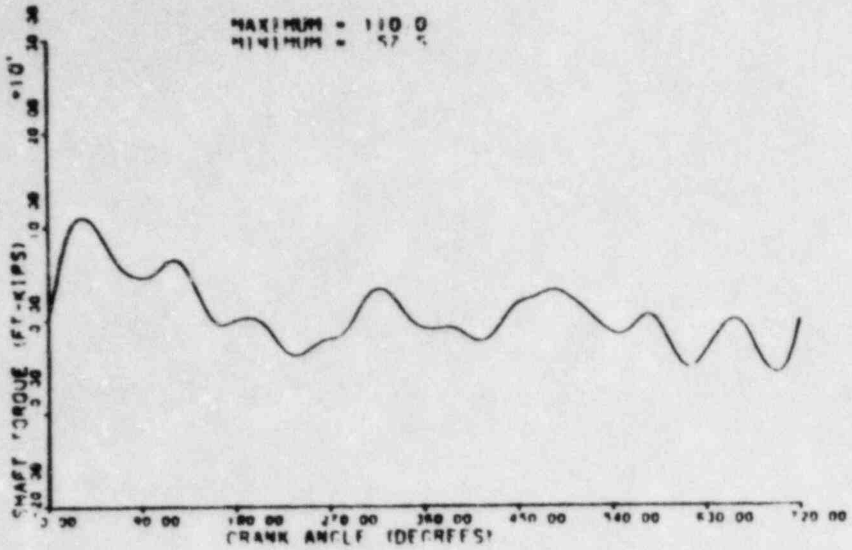
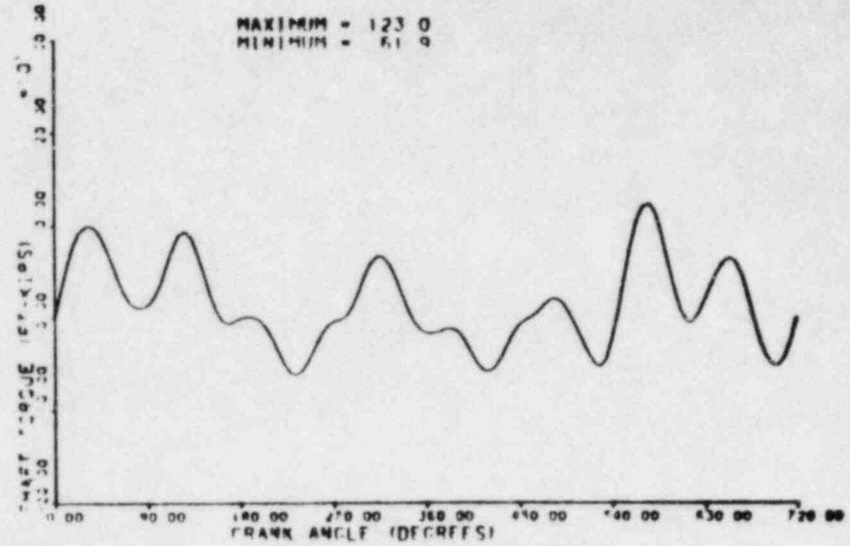


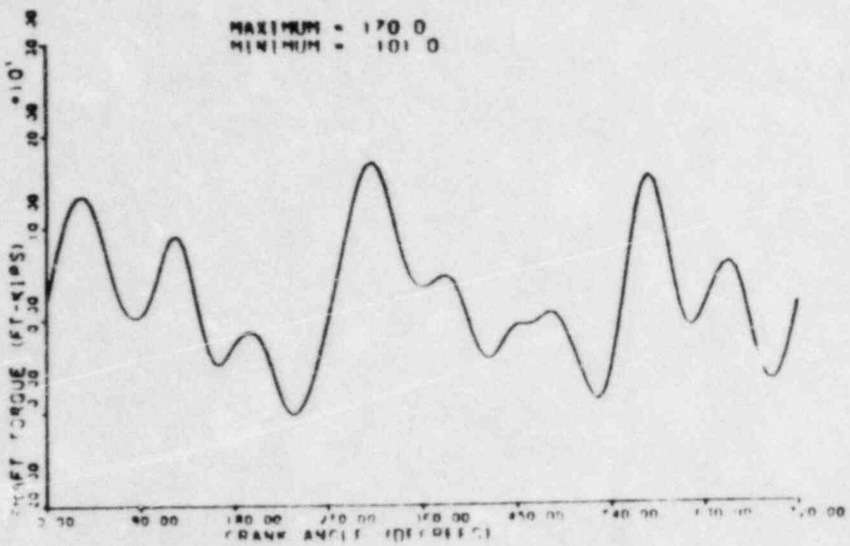
Figure 3-1. Measured pressure versus crank angle at 100% load.



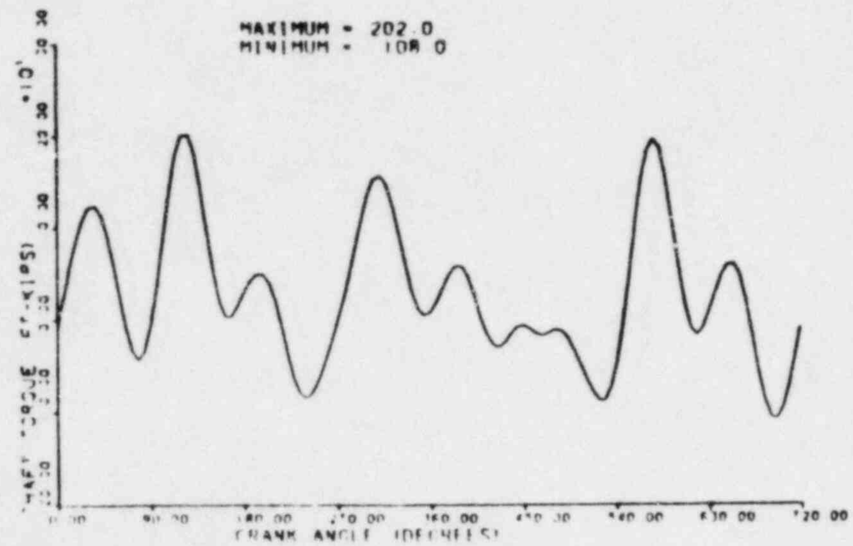
a) Cylinder 1 to 2.



b) Cylinder 2 to 3.



c) Cylinder 3 to 4.



d) Cylinder 4 to 5.

Figure 3-2. Dynamic model torsional response at 100% load for Shoreham 13-inch by 12-inch crankshaft.

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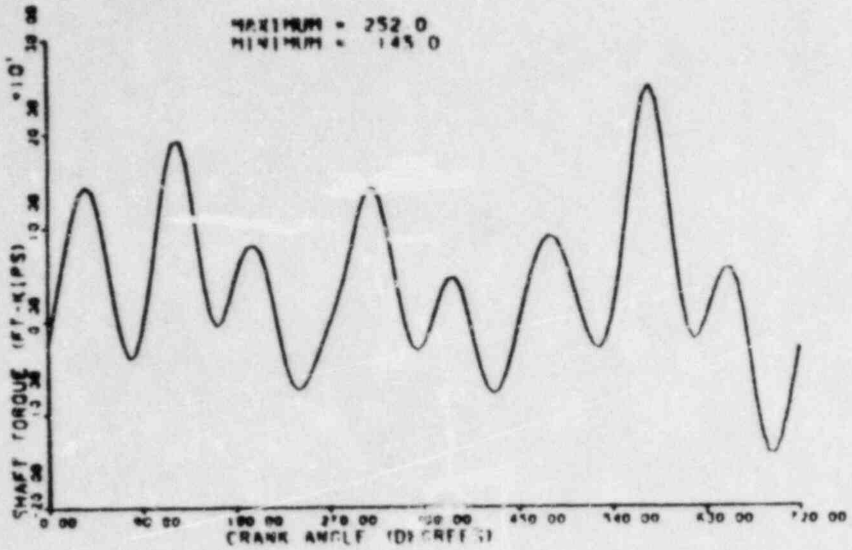
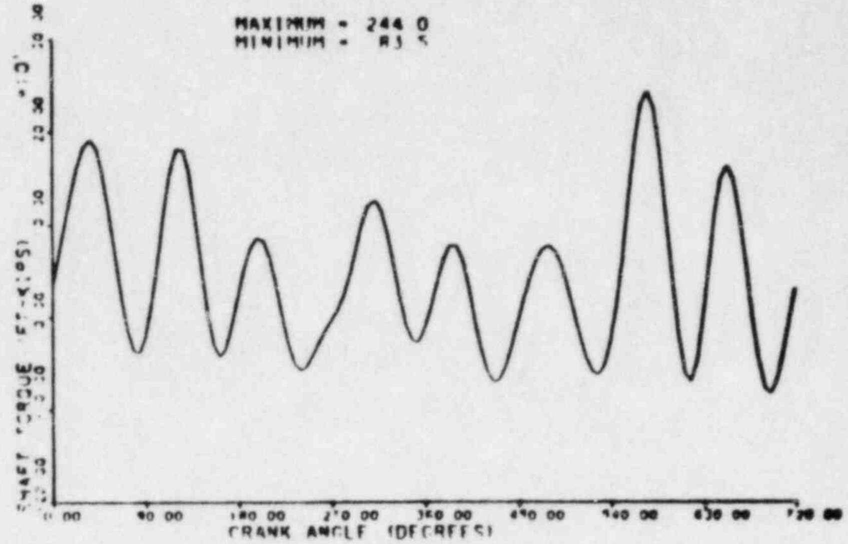
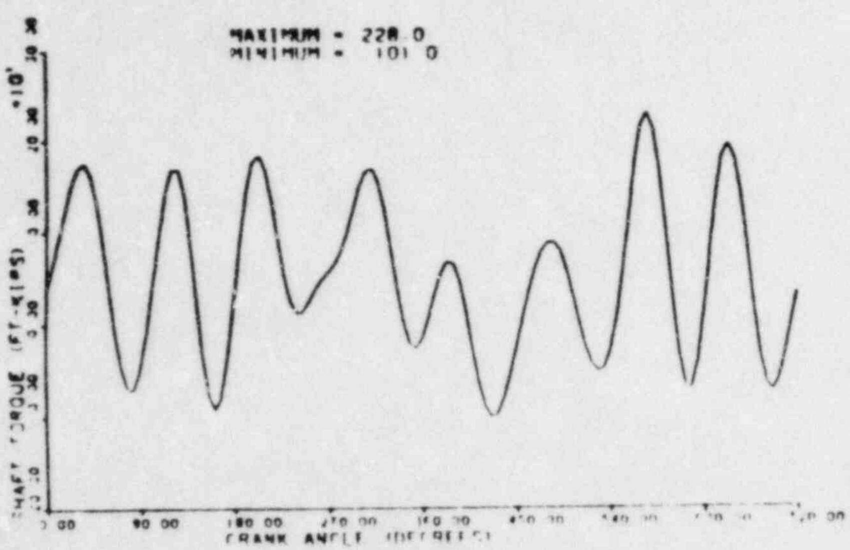


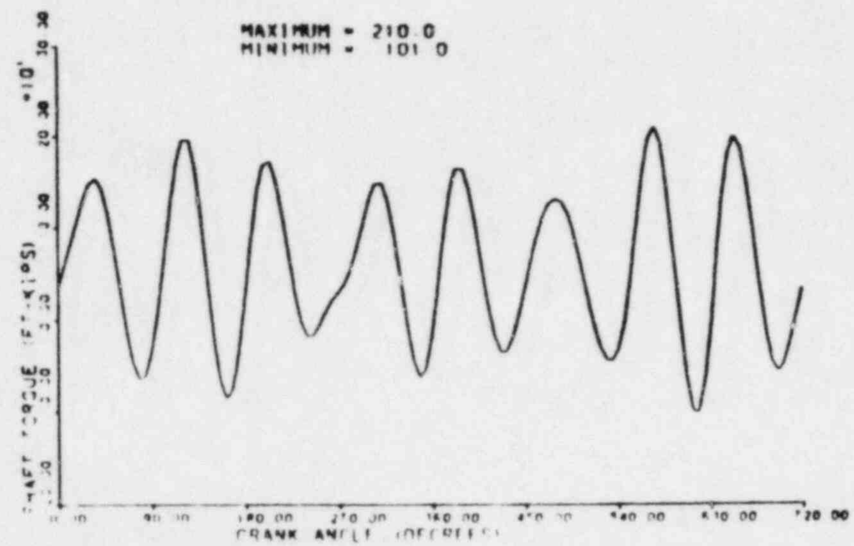
e) Cylinder 5 to 6.



f) Cylinder 6 to 7.



g) Cylinder 7 to 8.



h) Cylinder 8 to flywheel.

Figure 3-2 (continued).

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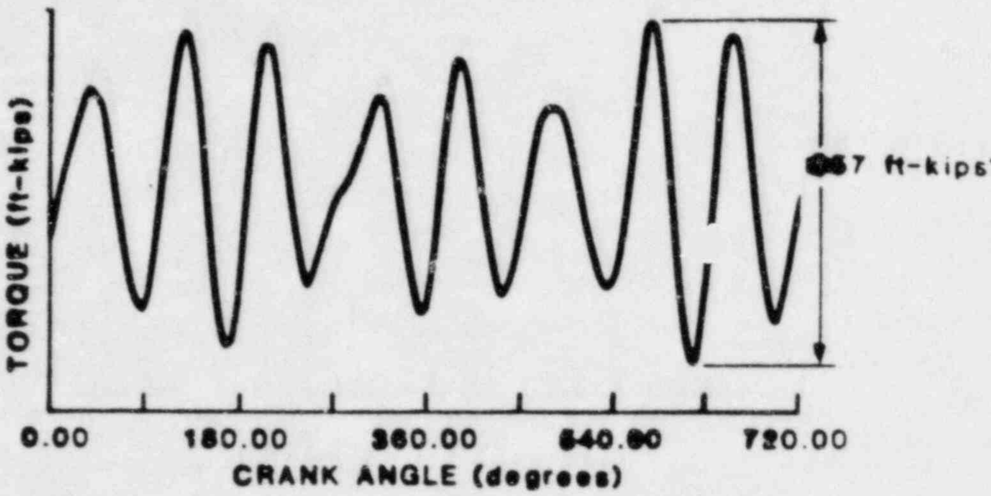
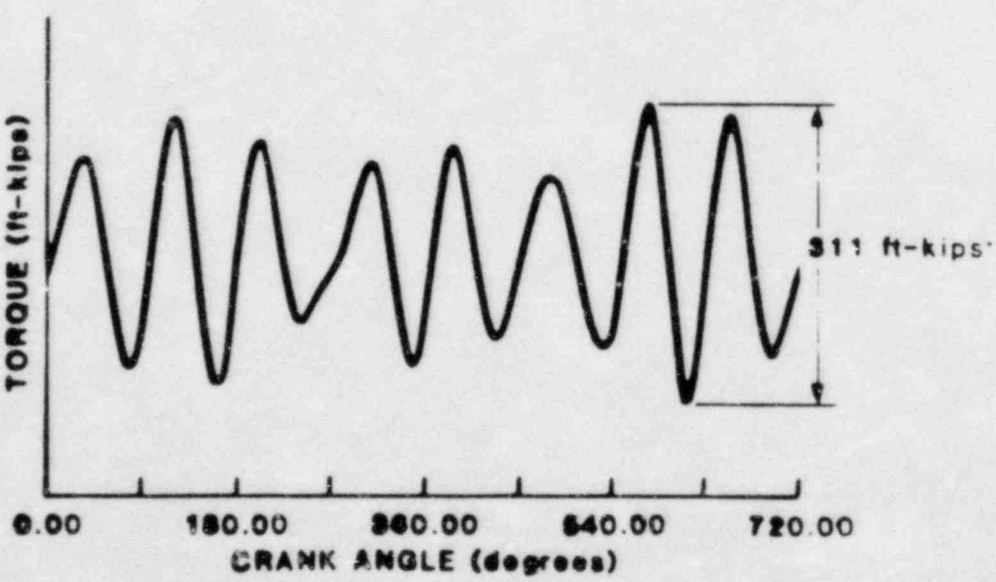


a) Measured.



b) Calculated.

Figure 3-3. Comparison of measured and calculated torque near the flywheel.

*See test for explanation of difference.

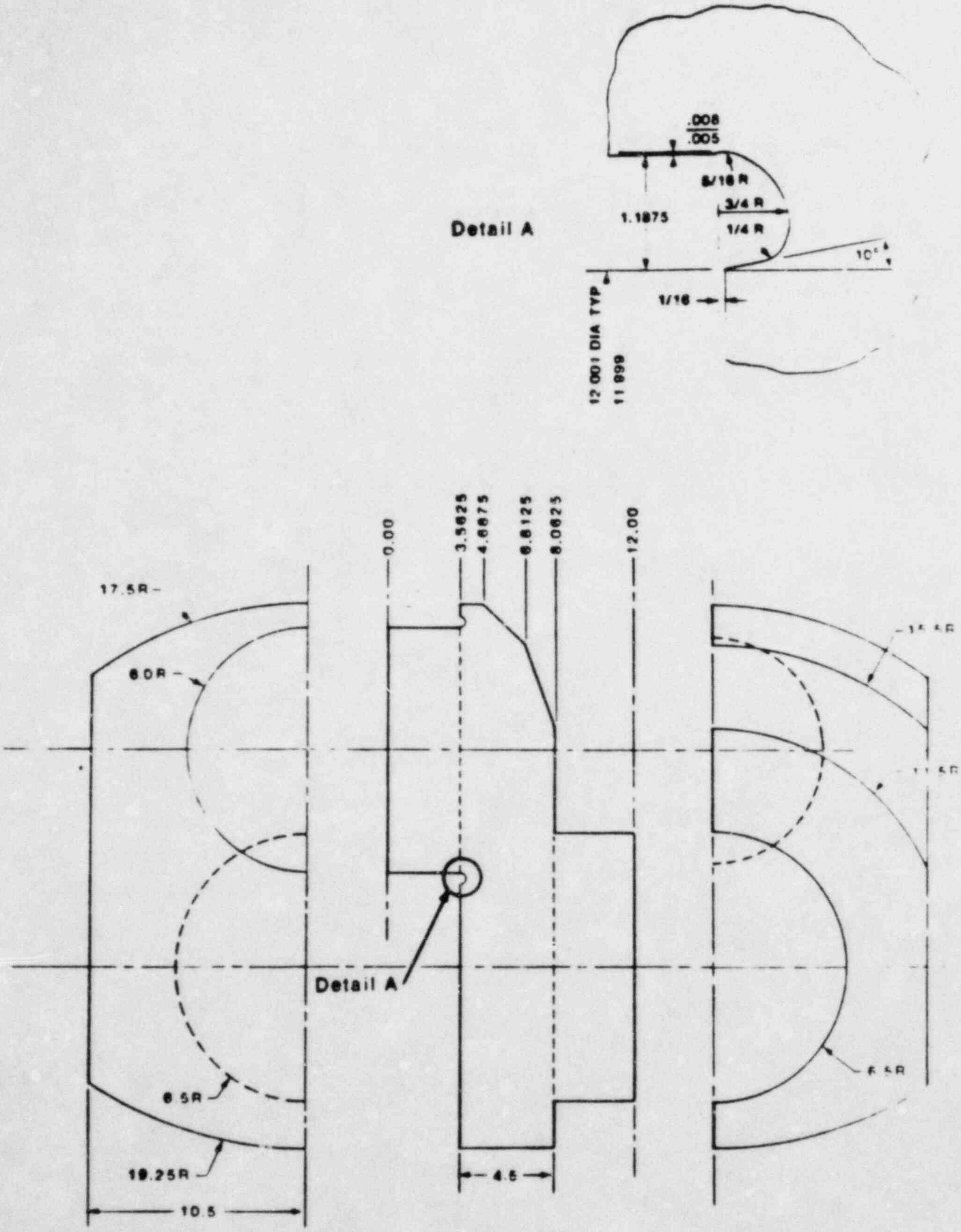


Figure 3-4. Typical structural dimensions of the crankshaft.

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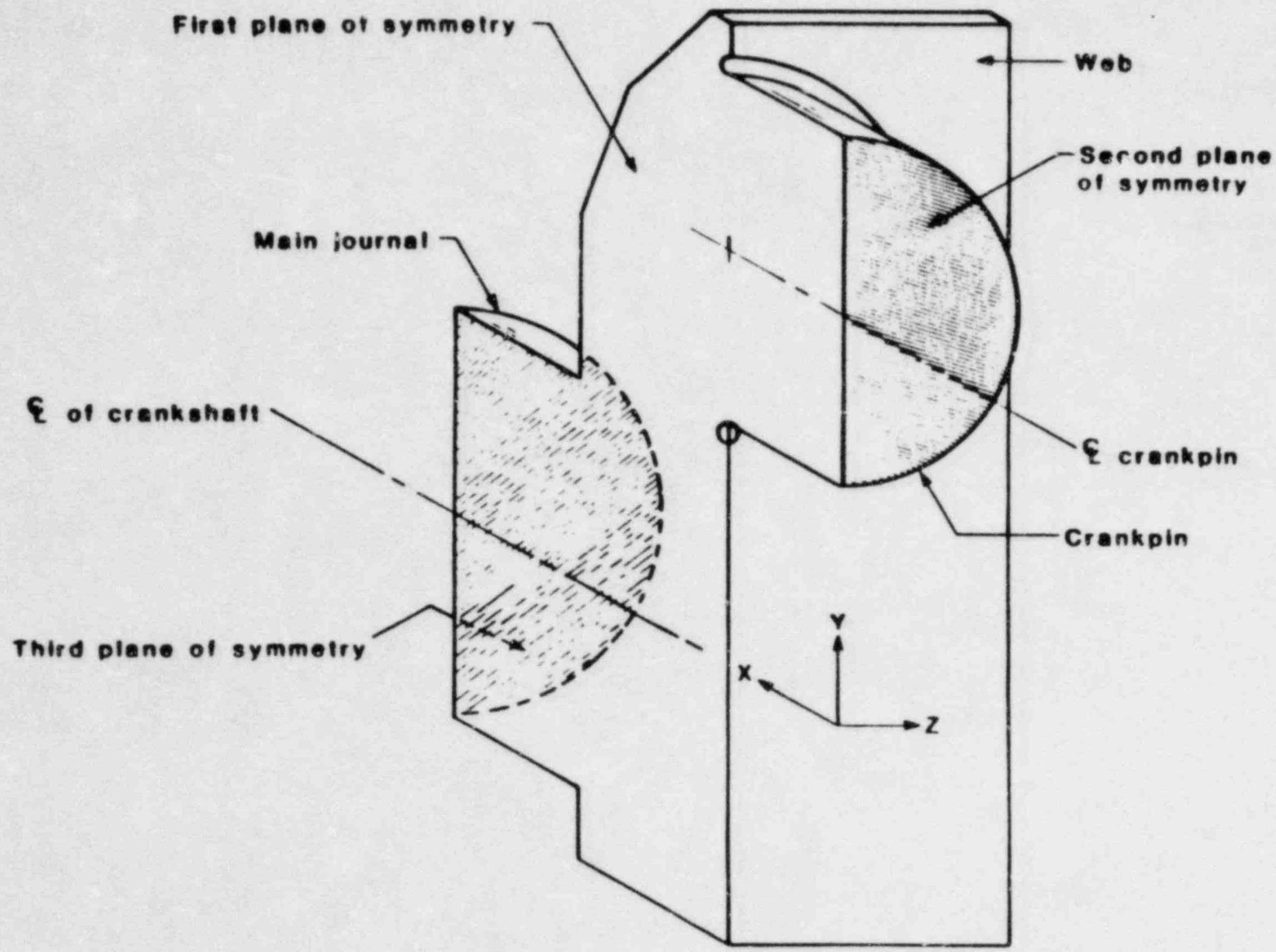

Figure 3-5. Typical section of the crankshaft showing coordinate axes and planes of symmetry.

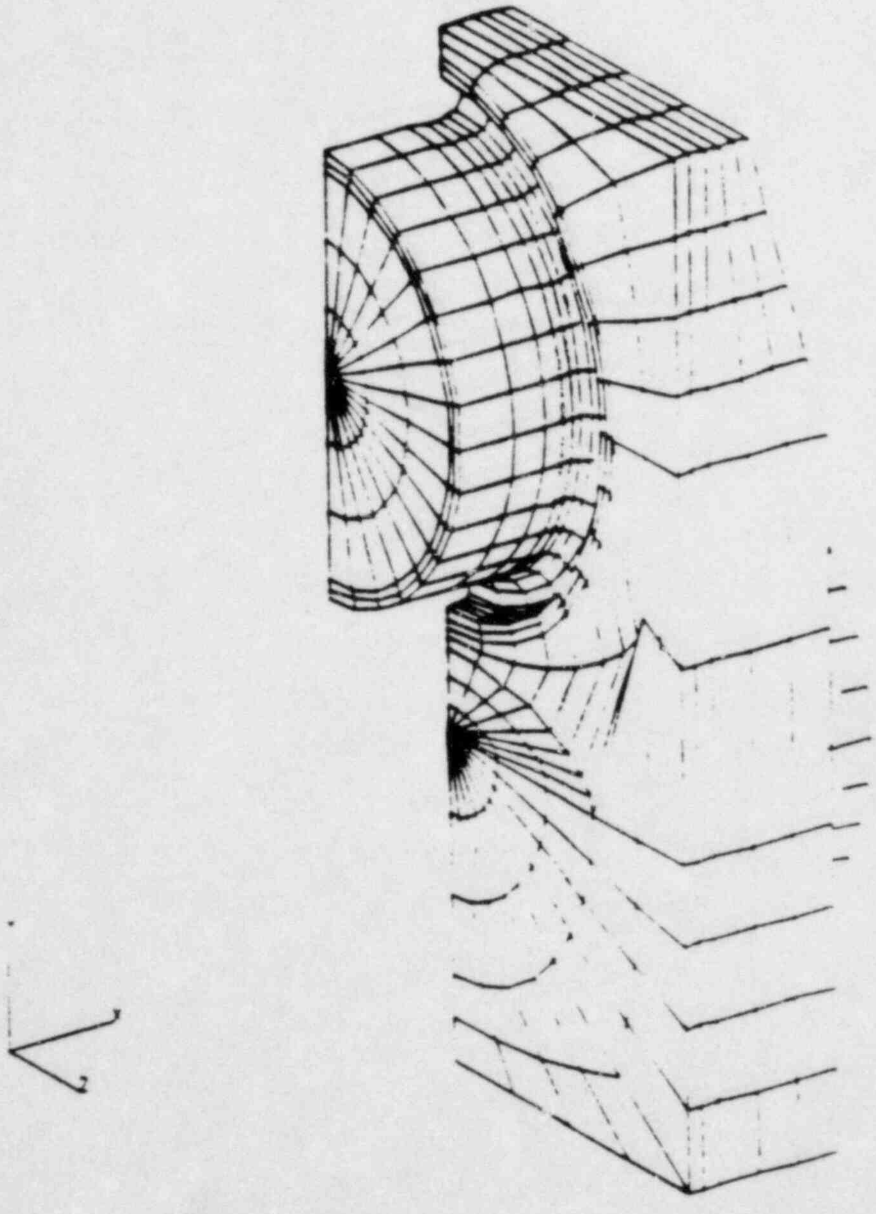

Figure 3-6. Three-dimensional view of the finite element model.

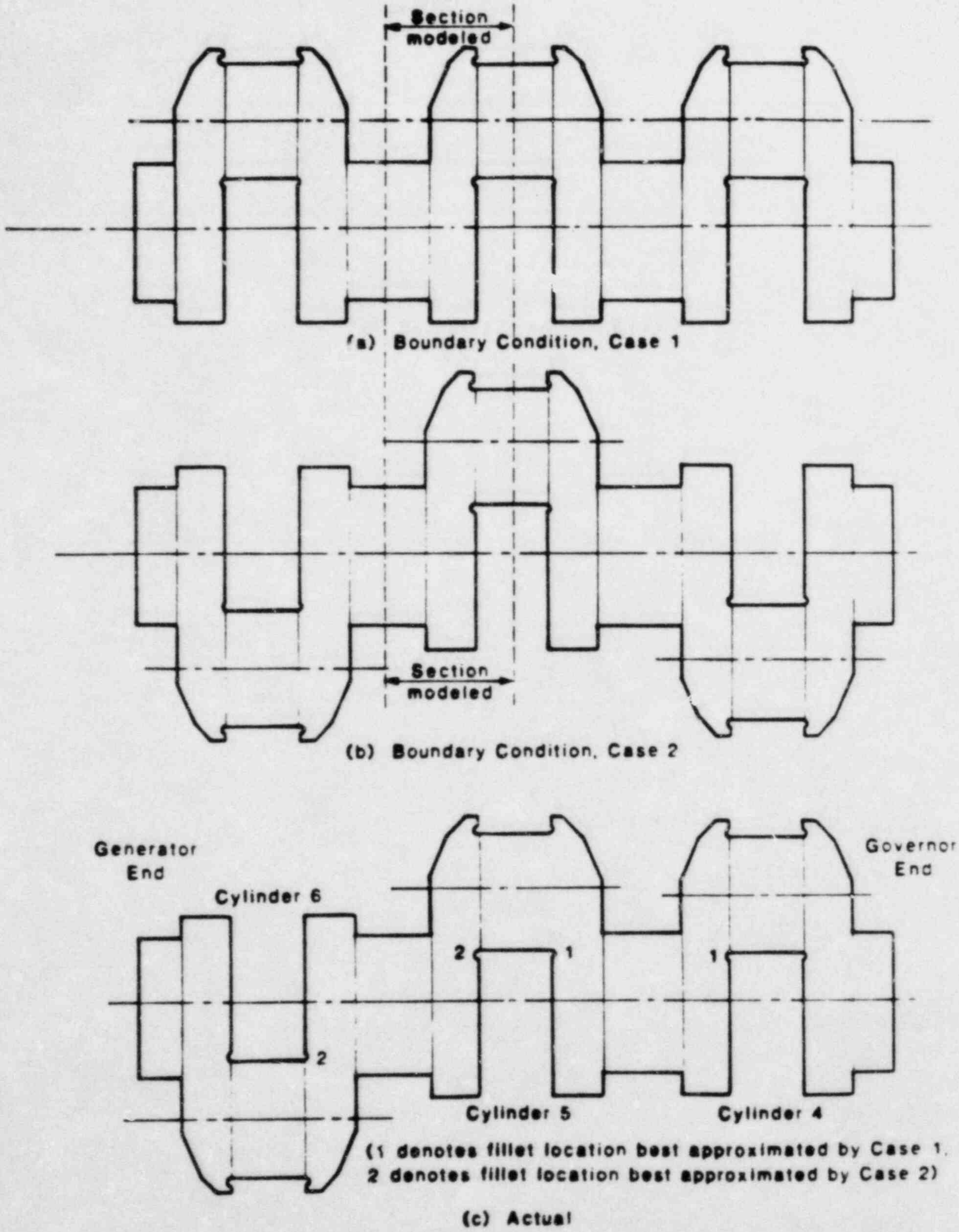

Figure 3-7. Schematic of relative crank throw orientations for (a) F.E. Model, Case 1; (b) F.E. Model, Case 2; and (c) the actual crankshaft.

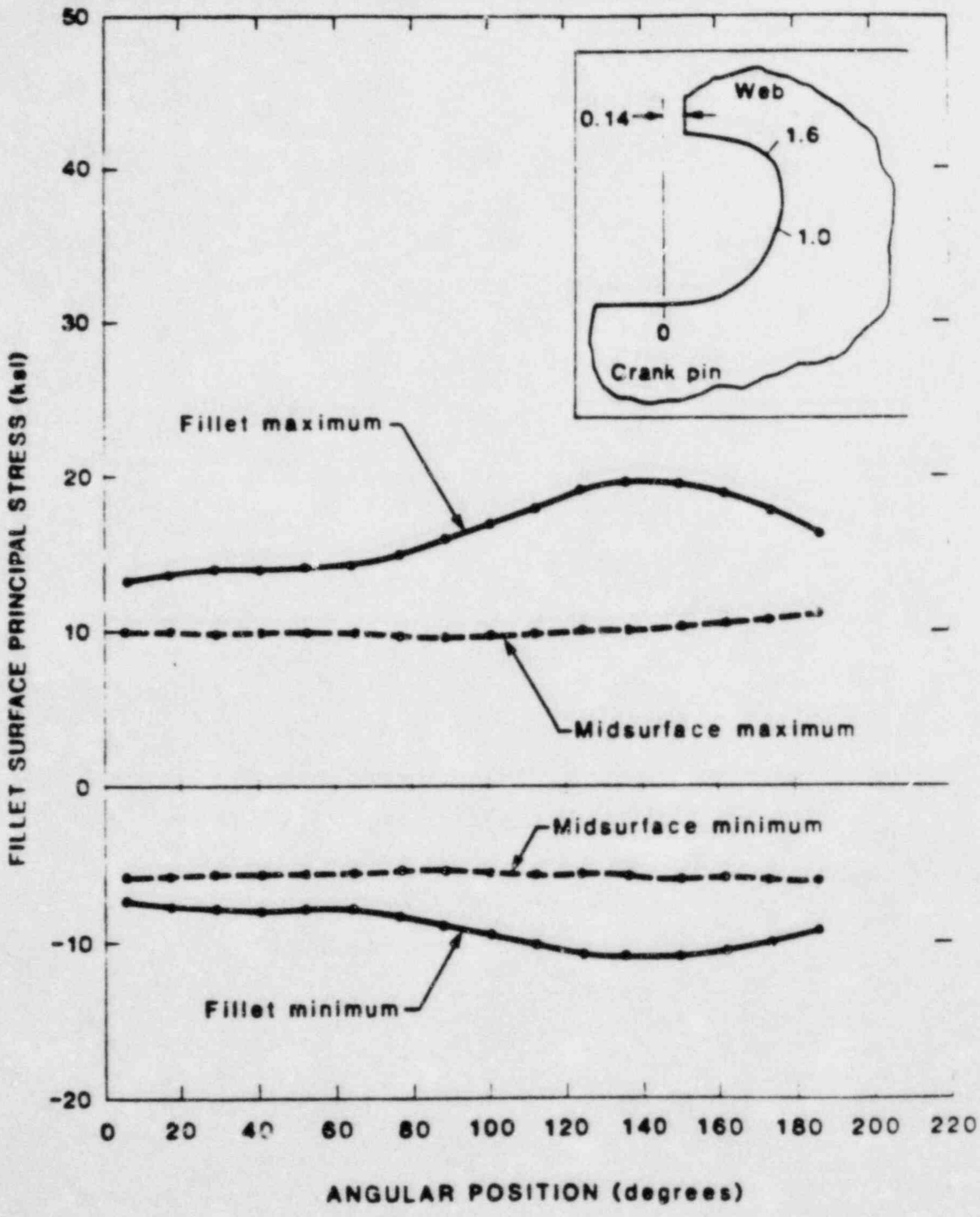

Figure 3-8. Circumferential variation of maximum principal for torsion Case 1 boundary conditions.

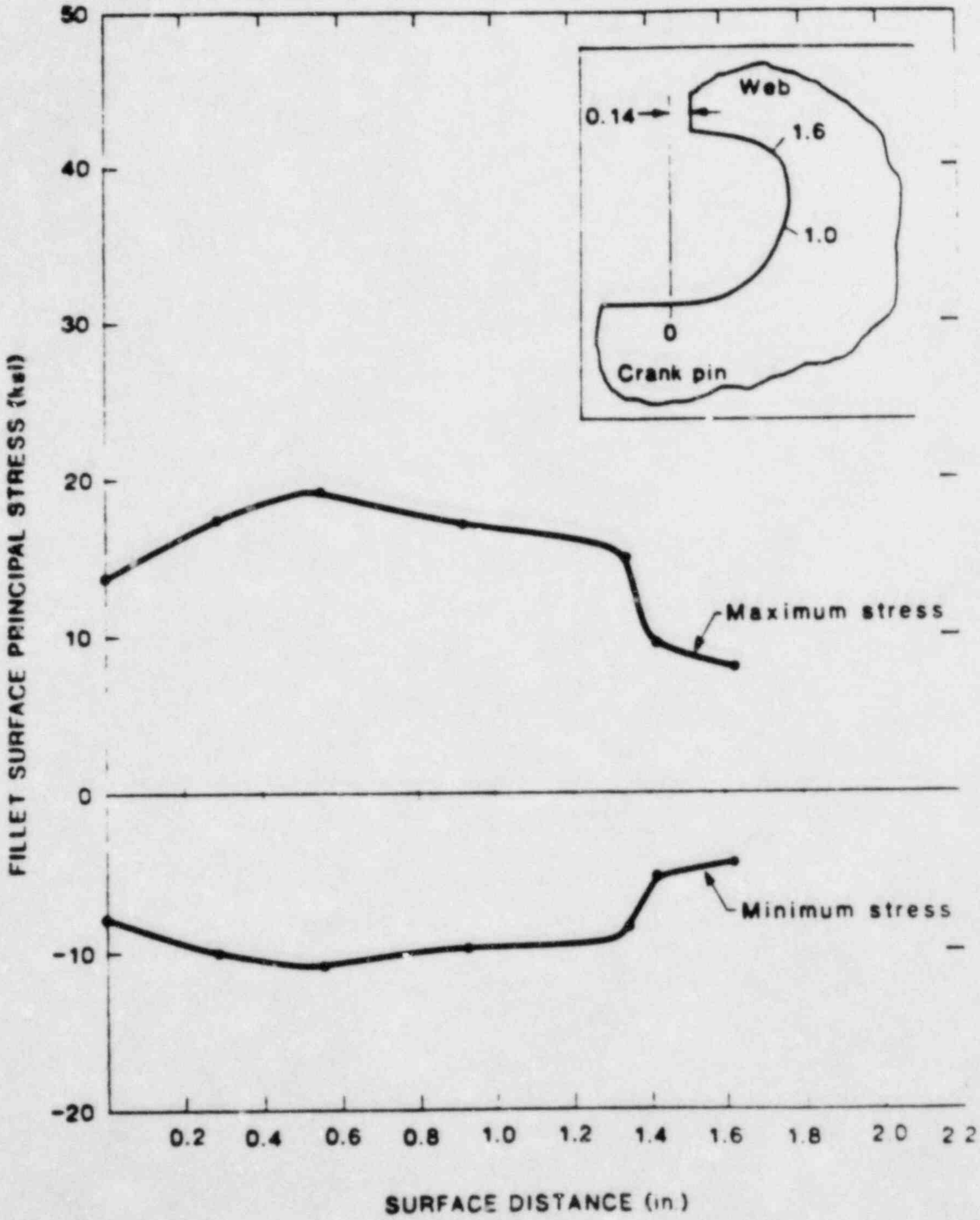

Figure 3-9. Axial variation of maximum principal stress for torsion Case 1 boundary conditions.

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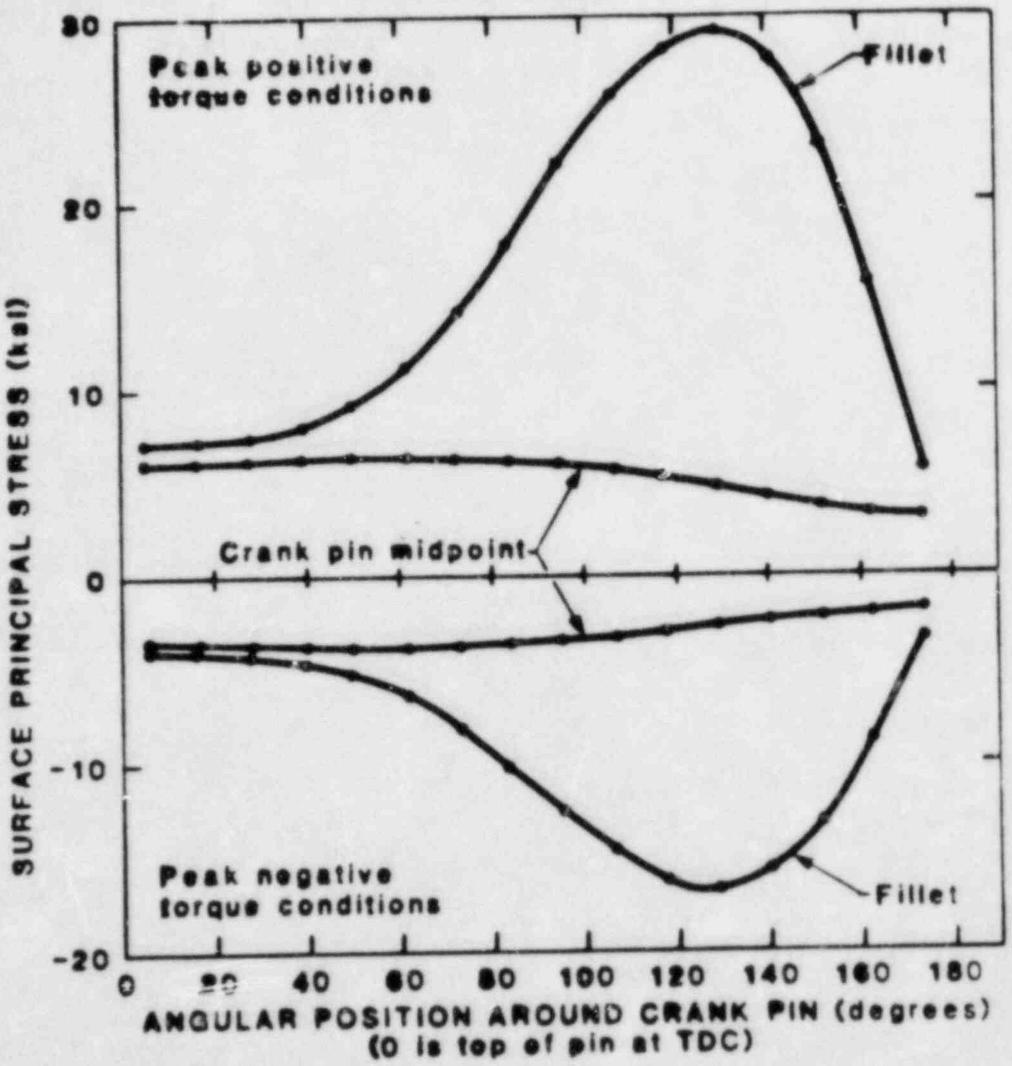

Figure 3-10. Circumferential variation of maximum principal stress for torsion Case 2 boundary conditions.

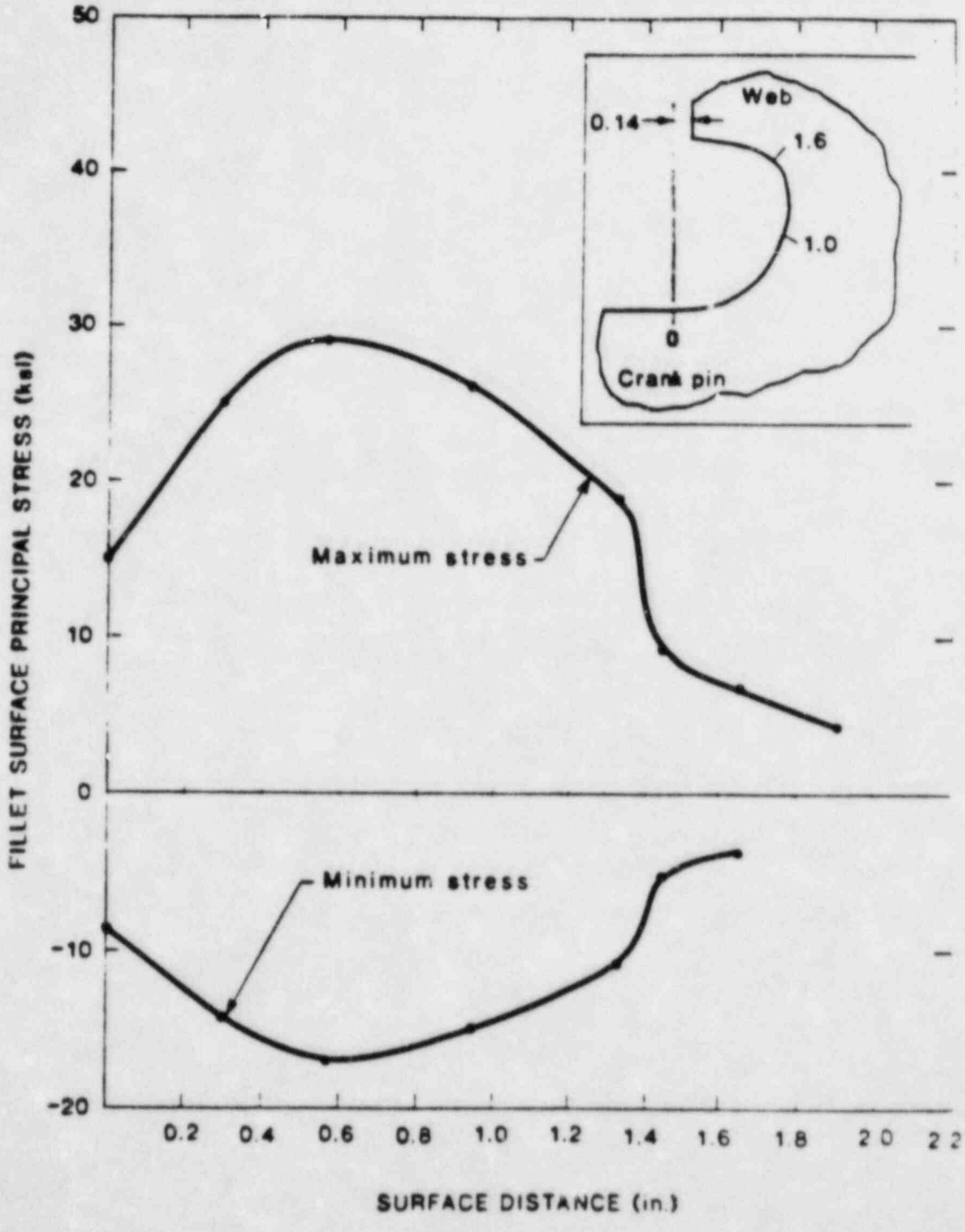

Figure 3-11. Axial variation of maximum principal stress for torsion Case 2 boundary conditions.

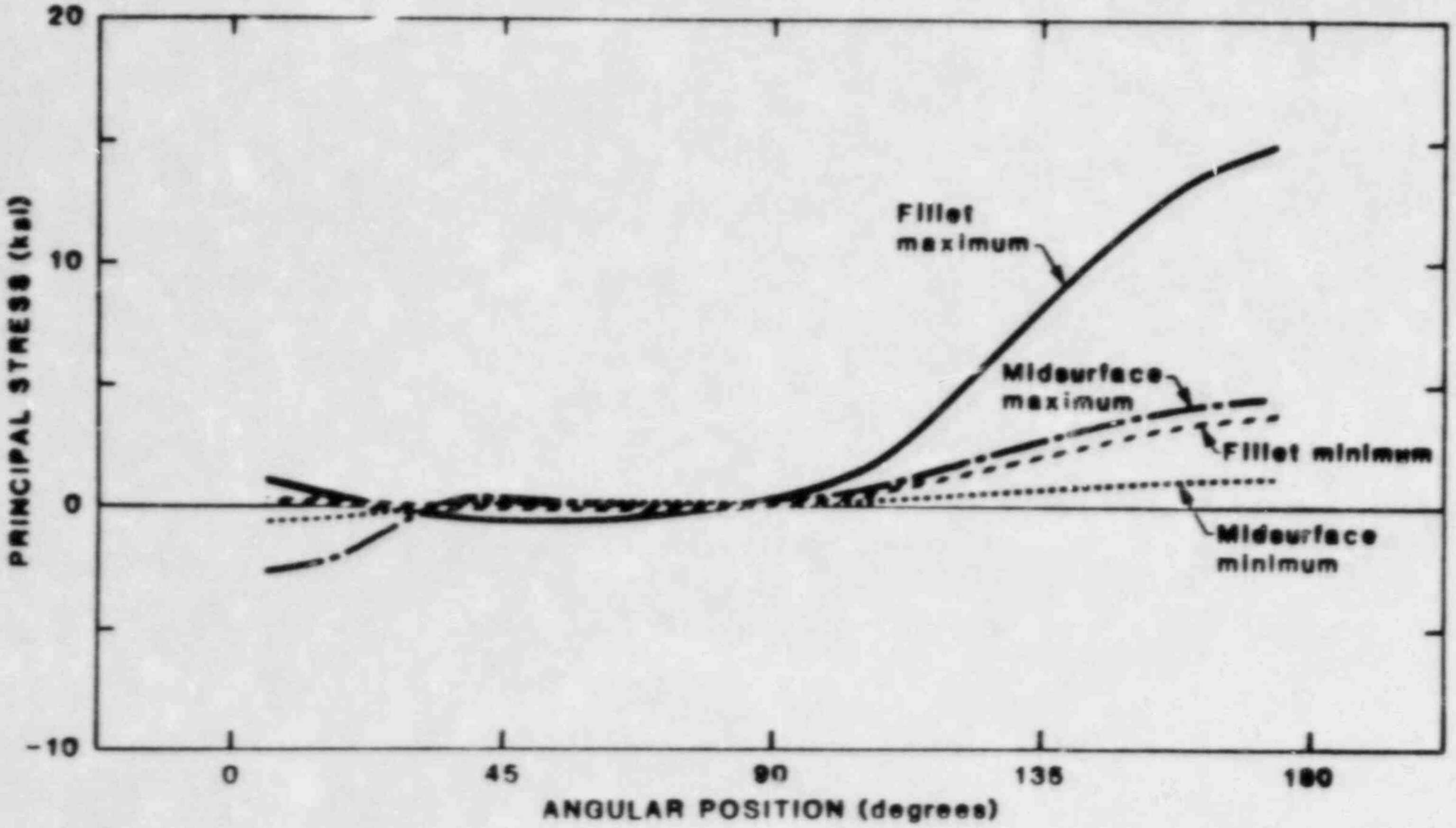

Figure 3-12. Circumferential variation of maximum principal stress for gas pressure loading.

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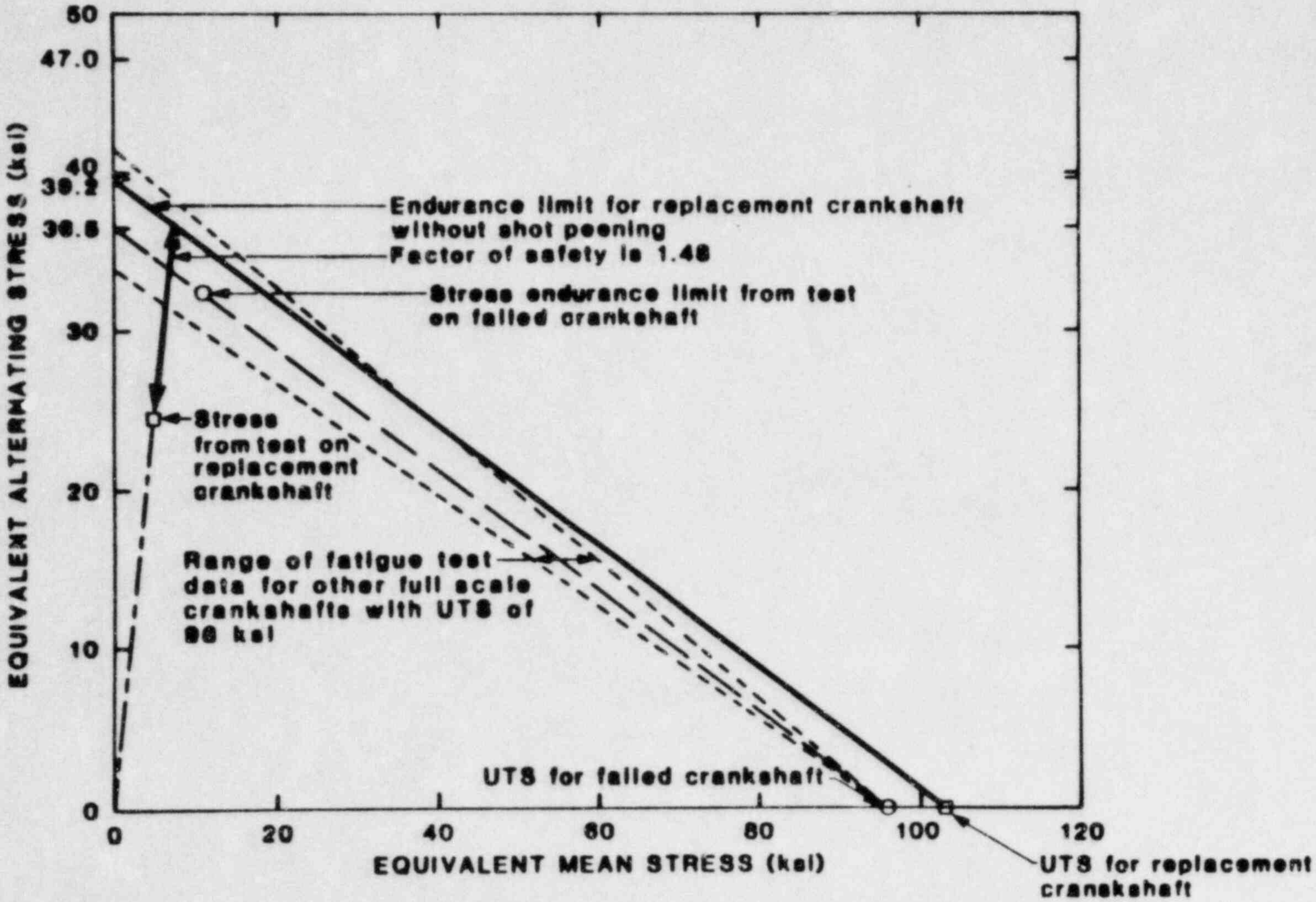

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Figure 3-13. Goodman diagram for replacement crankshafts.

4.0 DISCUSSION AND CONCLUSIONS

DSR-48 engines with 13-inch by 12-inch crankshafts are in diesel generator set service at seven other locations as shown in Table 4.1. This data shows that there has been extended service (long enough to produce more than 10^7 stress cycles) on several engines with 80% to 94% load, and limited service at 100% load.

The fillet regions of Crankpin Nos. 5 through 8 of the Shoreham replacement crankshafts were eddy current tested after 102 to 114 hours of operation at 100% or greater load (see Table 4.2). No relevant indications were found. Thus, there are no cracks in the high stresses fillet regions.

The drawing of the replacement crankshaft has been certified by the American Bureau of Shipping for compliance with their rules [4-1] for sizing of the pins, journals, and webs.

The following conclusions are made:

1. The design calculations on the 13-inch by 12-inch crankshafts performed by TDI are appropriate and show that the crankshaft stresses are below DEMA recommendations for a single order. Combined stress is not calculated by this method, but may be determined by torsiograph testing.

2. The SWEC torsiograph test results show that the 13-inch by 12-inch crankshaft stresses are below the DEMA recommended levels for both single order and combined orders for both 3500 kW (100% rated load) and 3800 kW. A linear extrapolation to 3900 kW also shows compliance.

3. The factor of safety against fatigue failure was found to be 1.48 if the effect of shot peening the fillet regions is ignored and is even greater if the shot peening of the Shoreham crankshafts is considered.

4. The replacement crankshafts are suitable for unlimited operation in the emergency diesel generators at SNPS.

Section 4 References

4-1 American Bureau of Shipping, Rules for Building and Classing Steel Vessels, New York, 1984.

TABLE 4.1

AVAILABLE LOGGED HOURS OF OPERATION OF DSR-48, RATED 3500 KW @ 450 RPM

Serial Number	Location	Kilowatt Rating @ 450 rpm	Total Hours Logged	Date Logged	Average Load Reported	Other Loads and Hours Reported
74010	SNPS	3500	368	3-21-84	--	>3500 kW for 114 hrs.
74011			430	2-13-84		>3500 kW for 116 hrs.
74012			345	3-14-84		>3500 kW for 110 hrs.
75005	KOUSHENG,	3600	246	3-15-84	Mostly	--
75006	TAIWAN		221	3-15-84	100%	
75007			368	3-15-84		
75008			299	3-15-84		
76010	DHUBA,	3500	19800	3-17-84	--	--
76011	SAUDI		23300	3-17-84		
76012	ARABIA		23800	3-17-84		
76013			19700	3-17-84		
76014			23500	3-17-84		
76026	ONEIZA,	3515	16204	3-17-84	--	3000/3200 kW for 9000 hrs.
76027	SAUDI		12428	3-17-84		
76028	ARABIA		14978	3-17-84		
78029	U.of	3500	8180	3-15-84	1100 kW	--
78030	TEXAS		5385	3-01-84	1100 kW	
78044	WADI	3515	10882	3-17-84	2200/3000 kW	--
78045	DAWASIR,		10832	3-17-84	2200/3000 kW	
78046	S.ARABIA		11212	3-17-84	2200/3000 kW	
79002	RAFHA,	3515	12667	3-16-84	--	3300 kW for 6200 hrs.
79003	SAUDI		11655	3-16-84	--	3200 kW for 8250 hrs.
79004	ARABIA		13186	3-16-84	--	3200 kW for 5500 hrs.
80001	RABIGH,	3515	10196	3-16-84	2700 kW	--
80002	SAUDI		10245	3-16-84	2800 kW	
80003	ARABIA		11602	3-16-84	2800 kW	

TABLE 4.2

HOURS OF OPERATION OF SHOREHAM REPLACEMENT CRANKSHAFTS AT TIME OF EDDY CURRENT TESTING

DIESEL GENERATOR	HOURS OF OPERATION AT ALL LOADS	HOURS OF OPERATION AT LOADS > 100% RATED LOAD
101	368	114
102	281	102
103	345	110

PART B:

REVIEW OF DSRV-16-4
13-INCH BY 13-INCH CRANKSHAFT

5.0 INTRODUCTION TO REVIEW OF DSRV-16-4 13-INCH BY 13-INCH CRANKSHAFT

This report presents Failure Analysis Associates' findings on the adequacy of the crankshafts in the emergency diesel engines at Grand Gulf Nuclear Power Station. The crankshaft is required to meet the recommendations of the Diesel Engine Manufacturers Association (DEMA). In Section 6.0, the design calculations and torsiograph test results of Transamerica Delaval Inc. (TDI) [5-1, 5-2] are reviewed for compliance with the DEMA stress allowables. In Section 7.0, a torsional dynamic analysis is used to compute nominal torsional stresses at each crank throw.

5.1 Industry Experience

The following information has not been verified by FaAA and consequently is not subject to FaAA's usual quality assurance procedures. Based upon information supplied by TDI [5-3, 5-4] three DSRV-16-4 13-inch by 13-inch crankshafts have failed since 1976. These failures were all attributed to torsional fatigue cracks initiating in the oil holes of Main Journal Nos. 6 or 8 (i.e., between Cylinder Nos. 5 and 6 and between Cylinder Nos. 7 and 8).

TDI subsequently modified the location of the oil hole and increased the radius at the intersection of the hole and the main journal. While increasing the radius should make the journal less subject to machining irregularities, it does not reduce the concentrated stress. Furthermore, the torsional stress is independent of angular location around the journal. Thus this area is still considered to be critical.

The failures are as follows: [5-4]

Date	Serial No.	Site	Location of Failure
Feb. 1976	73048	Mora, Minnesota	Main Journal No. 8
June 1976	73038	Anamax	Main Journal No. 8
March 1979	73034	Anamax	Main Journal No. 6

It was found that these engines had a 4th order critical speed at 446 rpm, which is close to the operating speed of 450 rpm. The engines were then fitted with four small counter-weights which moved this critical speed down to about 430 rpm. The installation at Grand Gulf has a 4th order critical speed of about 430 rpm.

Section 5 References

5-1 Yang, Roland, "Torsional and Lateral Critical Speed, Engine Numbers 74033/36 Delaval-Enterprise Engine Model DSRV-16-4 7000 KW/9770 BHP at 450 RPM". Transamerica Delaval Inc., Engine and Compressor Division, Oakland, California, October 22, 1975.

5-2 Yang, Roland, "Torsiograph of Middle South Energy Engine No. 74033 Generator Set DSRV-16-4, 9750 BHP, 7000 KW @ 450 RPM, 225 BMEP in Oakland Plant". Transamerica Delaval Inc., Engine and Compressor Division, Oakland, California, (not dated).

5-3 Transamerica Delaval Memo on cracked Anamax crankshaft, from Harold V. Schilling to E. G. Deane, Dec. 11, 1979.

5-4 Telephone conversation with Roland Yang, Transamerica Delaval, April 1984.

6.0 COMPLIANCE OF CRANKSHAFT WITH DIESEL ENGINE MANUFACTURERS ASSOCIATION RECOMMENDATIONS

The purchase specifications for the diesel generator sets require that the recommendations of the Diesel Engine Manufacturers Association, DEMA [6-1], 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 October, 1975, Transamerica Delaval Inc. (TDI) performed a torsional critical speed analysis of the crankshafts [6-2]. In Section 6.1, this analysis is reviewed for compliance with the above allowable stresses. Also, TDI conducted a torsiograph test on a 13-inch by 13-inch crankshaft for one of the Grand Gulf engines in TDI's Oakland plant [6-3]. In Section 6.2, the test results are compared with the above allowable stresses.

6.1 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 the rated load, 7000 kW.

6.1.1 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 6-1. The inertia and stiffness values are shown in Table 6.1.

It has long been standard practice in the diesel engine industry to solve this eigenvalue problem by the Holzer method [6-4]. This method has been used for at least 40 years [6-5], and thus is well established.

TDI used the Holzer method to calculate the system's first three natural frequencies, which are shown in Table 6.2. The first natural frequency was found to be 28.8 Hz, which produces 4th order resonance at 432 rpm.

6.1.2 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 signifcant orders used by TDI are shown in Table 6.3. These values may be compared to those recommended by Lloyd's Register of Shipping, LRS [6-6]. While certain of TDI's values are higher than LRS's values for low orders and lower for high orders, the largest single order was measured to be within 4% of those computed using TDI's values of T_n. 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 can be brought up to operating speed without undergoing excessive stresses as critical speeds are passed. Since the engine runs at 450 rpm and the 4th order critical speed is 432 rpm, the resonant response is important for the 4th order. In a V16 engine with articulated rods, the 4th order loading from one bank almost cancels that from the other bank, significantly reducing the excitation. However, this excitation is sensitive to the balance between the two banks.

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 [6-7] for the nth order harmonic and given mode shape. The nominal shear stress, τ, in the 13-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 6.4. It is seen that the largest single order stress of 1956 psi for the 3 $^1/_2$ 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 computation of the combined response is reported in Section 7.0.

6.2 Review of Transamerica Delaval Inc. Torsiograph Test

Torsiograph tests are commonly used to confirm torsional vibration calculations. The 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. The second stage is performed at rated speed of

450 rpm with variable load, and is used to confirm the forced vibration calculations.

6.2.1 Natural Frequencies

The variable speed torsiograph test found the first natural frequency to be 28.7 Hz by locating the 4th order resonance. This value is in excellent agreement with TDI's computed value of 28.8 Hz.

6.2.2 Nominal Stresses

The torsiograph provides the angular displacement response of the free end of the crankshaft. This displacement is usually decomposed into components corresponding to each order. The peak-to-peak response is also obtained.

The nominal shear stress, τ, in Crankpin No. 8 may be established from the amplitude of free-end vibration by assuming the shaft is vibrating in the first mode. The nominal shear stress is then found to be 8540 psi per degree of free-end vibration from the TDI analysis [2-2].

TDI tabulated the single order and total response for both 7000 kW (100% of rated load) and for 7700 kW (110% load). These values have been factored to obtain nominal shear stresses and are shown in Table 6.5. The results at 7000 kW show that the largest single order has a stress of 2028 psi which is well below the DEMA allowable of 5000 psi.

The measured response (Table 6.5) was in good agreement with that calculated by TDI and shown in Table 6.4, although somewhat higher in some cases than the calculated values. TDI determined the stress to be 2366 psi at 7700 kW (110% load), also well below the DEMA allowable. In determining the combined total response, TDI utilized the Bell & Howell C.E.C. instrumentation. Subsequent examination of the data analysis characteristics of this instrumentation revealed that the B & H unit measured twice the square root of the sum of the squares, SRSS, of the amplitudes of individual orders rather than the true peak-to-peak. The peak-to-peak range divided by two is larger than the SRSS. The peak-to-peak range of nominal stress was calculated from a torsiograph test performed by FaAA on the Duke Power Co. Catawba No. 1A diesel generator set, which is nominally identical in specification to the engines at GGNS [5-4].

6.3 Nominal Stresses for Underspeed and Overspeed Conditions

The crankshaft stresses are within the DEMA allowables for a speed range of 440 rpm to 450 rpm +5%, provided adequate engine balance is maintained. The engines should not be allowed to operate below 440 rpm except during startup and shutdown.

The balance specifications provided by TDI may be adequate. However, on the basis of one engine test it is not possible to conclude that all engines will respond identically to this balance. Until the Owners' Group has established a data base for the expected variations of speed and balance, it is suggested that additional torsiograph tests be conducted.

Section 6 References

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

6-2 Yang, Roland, "Torsional and Lateral Critical Speed, Engine Numbers 74033/36 Delaval-Enterprise Engine Model DSRV-16-4 7000 KW/9770 BHP at 450 RPM". Transamerica Delaval Inc., Engine and Compressor Division, Oakland, California, October 22, 1975.

6-3 Yang, Roland, "Torsiograph of Middle South Energy Engine No. 74033 Generator Set DSRV-16-4, 9750 BHP, 7000 KW @ 450 RPM, 225 BMEP in Oakland Plant". Transamerica Delaval Inc., Engine and Compressor Division, Oakland, California, (not dated).

6-4 Thomson, William T., Theory of Vibration with Applications. Second edition, Prentice-Hall, 1981.

6-5 Den Hartog, J., Mechanical Vibrations. Third edition, McGraw-Hill, 1947.

6-6 Lloyd's Register of Shipping, Guidance Notes on Torsional Vibration Characteristics of Main and Auxillary Oil Engines.

6-7 Craig, Roy R., Jr., Structural Dynamics: An Introduction to Computer Methods. Wiley, 1981.

TABLE 6.1

STIFFNESS AND INERTIAS FOR TORSIONAL DYNAMIC ANALYSIS OF DSRV-16-4 13-INCH BY 13-INCH CRANKSHAFT

Inertia Location	Inertia (lb. ft. sec^2)	Stiffness (ft. lb./rad)
Front Gear	14.8	57.3×10^6
Cylinder No. 1	78.3	101.9×10^6
Cylinder No. 2	76.8	101.9×10^6
Cylinder No. 3	102.6	101.9×10^6
Cylinder No. 4	102.6	101.9×10^6
Cylinder No. 5	102.6	101.9×10^6
Cylinder No. 6	102.6	101.9×10^6
Cylinder No. 7	76.8	101.9×10^6
Cylinder No. 8	79.8	72.7×10^6
Flywheel	532.4	216.3×10^6
Generator	9069.4	

TABLE 6.2

TORSIONAL NATURAL FREQUENCIES FROM TDI ANALYSIS

Mode	Natural Frequency (Hz)
1	28.8
2	83.0
3	113.0

TABLE 6.3

TORSIONAL LOADINGS FOR TDI ANALYSIS

Order	Torsional Loading, T_n (psi)
1.5	129.5
2.5	71.7
3.5	42.8
4.0	27.7
4.5	23.8
5.5	12.8

TABLE 6.4

SINGLE-ORDER NOMINAL SHEAR STRESSES FROM TDI ANALYSIS

Order	Amplitude of Nominal Shear Stress (psi)
1.5	1361
2.5	1645
3.5	1956
4.0	853
4.5	265
5.5	86
DEMA Allowable for Single Order	5000

TABLE 6.5

NOMINAL SHEAR STRESSES CALCULATED FROM TDI TORSIOGRAPH TEST

Order	Amplitude of free-end rotation (degrees)		Amplitude of Nominal Shear Stress (psi)*	
	100% load	110% load	100% load	110% load
1.5	0.16	0.21	1352	1775
2.5	0.22	0.27	1859	2282
3.5	0.24	0.28	2028	2366
4.0	0.10	0.12	845	1014
4.5	0.05	0.08	423	676
5.5	0.03	0.03	254	254
DEMA Allowable for a Single Order			5000	5000
SRSS**	0.42	0.45	3507	3803
DEMA Allowable ½ peak to peak			7000	7000

* Amplitude of nominal shear stress is calculated to be 8452 psi per degree of free-end rotational amplitude.

**Measurement corresponds to the square root of the sum of the squares, SRSS, of individual orders rather than the peak-to-peak divided by two.

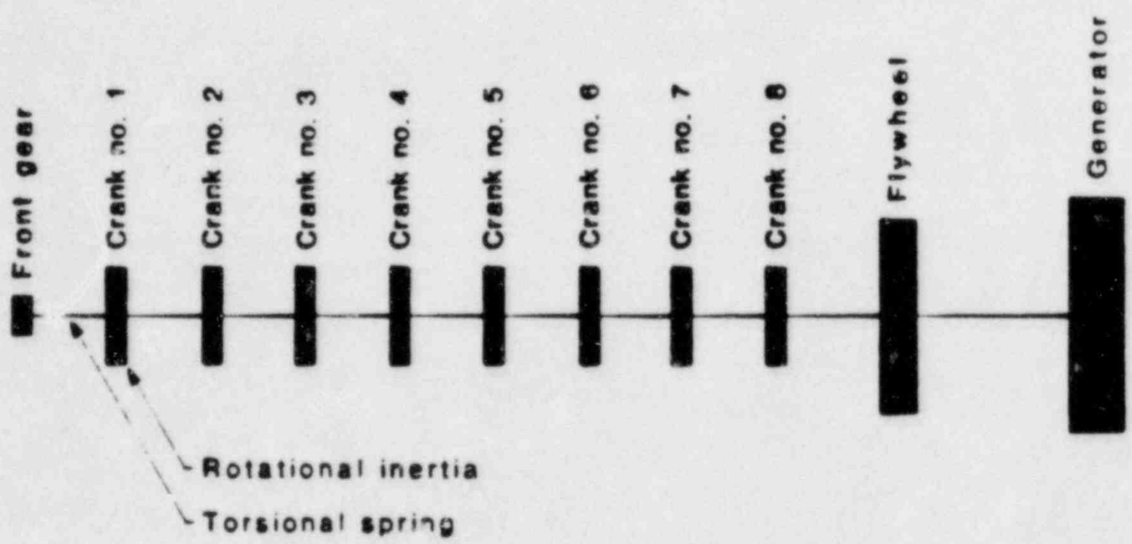

Figure 6-1. TDI dynamic model.

FaAA-84-3-16

7.0 CRANKSHAFT DYNAMIC TORSIONAL ANALYSIS

In Section 6.0 it was found that the crankshafts satisfy the DEMA single order nominal stress recommendation of 5000 psi for both 7000 kW (100% of rated load) and 110% load. However, the stresses for combined orders were neither calculated nor measured by TDI, and thus could not be compared to the DEMA recommended limit of 7000 psi. A dynamic torsional analysis of the crankshaft is performed to determine the true range of torque at each crank throw. The FaAA dynamic model allows all orders and modes to be summed using appropriate phase angles. In this section this model is compared with TDI torsiograph test data for the amplitudes of free-end vibration.

7.1 Torsional Model

FaAA developed a dynamic torsional model of the crankshaft to supplement TDI's conventional forced vibration calculations. The original TDI procedure did not compute the phase relationship between the various orders or modes, so it was not possible to compute the true summation. The actual maximum stress is a direct result of this summation. Furthermore, the original TDI method always predicted maximum stress in Crankpin No. 8, which is generally true for a single order but not true for the combined response. The dynamic model developed used the same idealized lumped inertia and torsional spring model as the TDI analysis (Figure 6-1 and Table 6.1).

The first five torsional natural frequencies for the replacement crankshaft are shown in Table 7.1. The natural frequencies are in agreement with those computed and measured by TDI.

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 model superposition analysis [7-1] was used with harmonic load input. The calculation of the harmonic loads will be discussed in the next section.

7.2 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 may be obtained from pressure versus crank angle data. This pressure was measured in a Stone & Webster Engineering Corporation (SWEC) test on a similar TDI series R-4 cylinder at Shoreham Nuclear Power Station [7-2]. 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 100% load was reduced by FaAA to obtain the pressure curve shown in Figure 7-1. The pressure variation with crank angle is assumed to be the same on the articulated side as on the master side except for a one degree difference in timing between the two banks*.

The reciprocating mass of the connecting rod and piston was found to be 820 lbs for each connecting rod. 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 resolved 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 7.2.

7.3 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.

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*The name plate on the Catawba engine lists the timing as 22° and 21° BTDC for the left and right bank, respectively.

The dynamics of a V16 engine result in the 4th order load components from the left and right banks almost canceling. In practice, this does not completely happen due to differences in timing, pressure/volume diagram, and articulated rod versus master rod geometry. To simulate these effects, a one degree delay was applied to the right bank of cylinders. This produces a small 4th order loading which is then amplified by the proximity of the first natural frequency of the shaft (28.8 Hz) to this loading frequency (30 Hz). It should be recognized that for this order the analysis is dependent on the torsiograph test results.

The calculated amplitude of free-end displacement is compared to the amplitude measured by TDI, in Table 7.3. It is seen that the agreement is close for all significant orders. The vector summation from the FaAA analysis is half the peak-to-peak displacement. The largest response occurs for the 3.5 order.

The model also calculates the range of torque at each crank throw, which is shown along with the corresponding nominal shear stress ($\tau = Tr/J$) in Table 7.4. The stresses in Table 7.4 show that the stress level is highest between cylinders 4 and 5, 5 and 6, and 7 and 8. The highest nominal shear stress amplitude is found to be 5367 psi. This value is lower than the 7000 psi DEMA allowable for combined orders, even though the calculated value was computed using all modes. The computed torque as a function of crank angle for each crank throw is shown in Figure 7-2. The nominal shear stress amplitudes for 110% load may be calculated by extrapolation and are found to be below the DEMA recommended allowables.

Section 7 References

7-1 Timoshenko, S., D.H. Young, and W. Weaver, Jr., Vibration Problems in Engineering. Fourth edition, Wiley, 1974.

7-2 Bercel, E., and Hall, J.R., "Field Test of Emergency Diesel Generator 103," Stone & Webster Engineering Corporation, April 1984.

TABLE 7.1

**NATURAL FREQUENCIES FOR DSRV-16-4
13-INCH BY 13-INCH CRANKSHAFT**

Mode	Natural Frequency (Hz)
1	28.84
2	83.01
3	113.03
4	149.32
5	196.64

TABLE 7.2

TORSIONAL LOADING FOR FaAA ANALYSIS

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
5.5	15.5

TABLE 7.3

FREE-END VIBRATION AT 100% LOAD FOR
DSRV-16-4 13-INCH BY 13-INCH CRANKSHAFT

Order	Amplitude of Vibration (degrees)	
	FaAA Analysis	TDI Test
1.5	0.179	0.16
2.5	0.252	0.22
3.5	0.269	0.24
4.0	0.103	0.10
4.5	0.054	0.05
5.5	0.012	0.03
Vector Summation	0.586	--

TABLE 7.4

TORQUE RANGE AT 100% LOAD FOR
DSRV-16-4 13-INCH BY 13-INCH CRANKSHAFT

Location	Torque Range (ft. lbs.)	Amplitude of Nominal Shear Stress (psi)
Between Cylinder No. 1 and Cylinder No. 2	199.6×10^3	2776
Between Cylinder No. 2 and Cylinder No. 3	229.6×10^3	3193
Between Cylinder No. 3 and Cylinder No. 4	383.5×10^3	5334
Between Cylinder No. 4 and Cylinder No. 5	347.3×10^3	4831
Between Cylinder No. 5 and Cylinder No. 6	385.9×10^3	5367
Between Cylinder No. 6 and Cylinder No. 7	338.3×10^3	4705
Between Cylinder No. 7 and Cylinder No. 8	379.3×10^3	5276
Between Cylinder No. 8 and Flywheel	286.3×10^3	3982

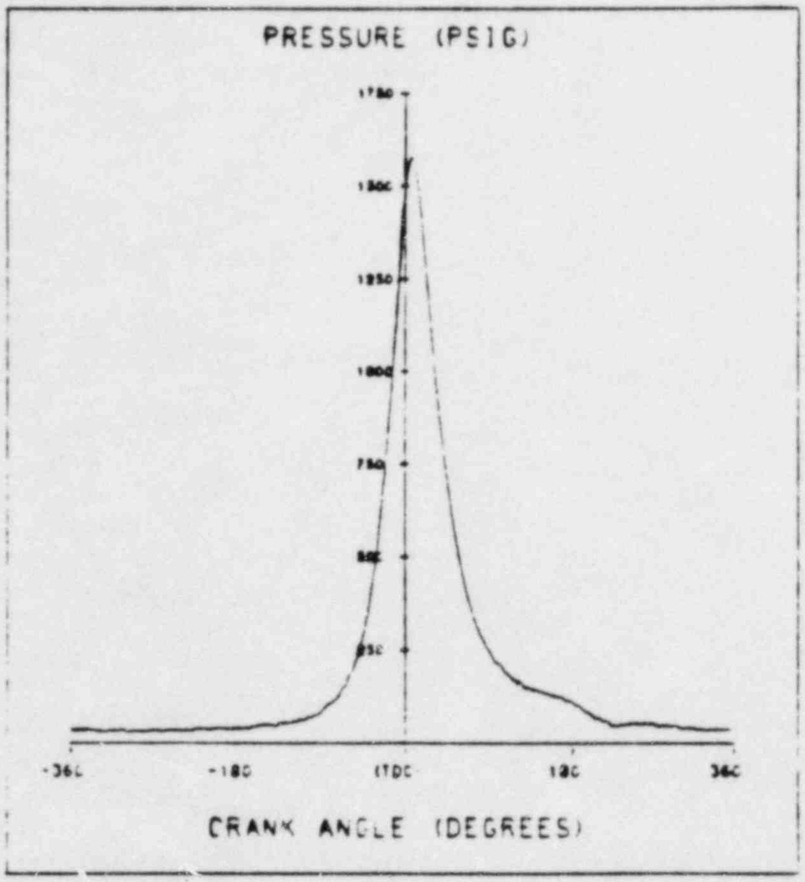
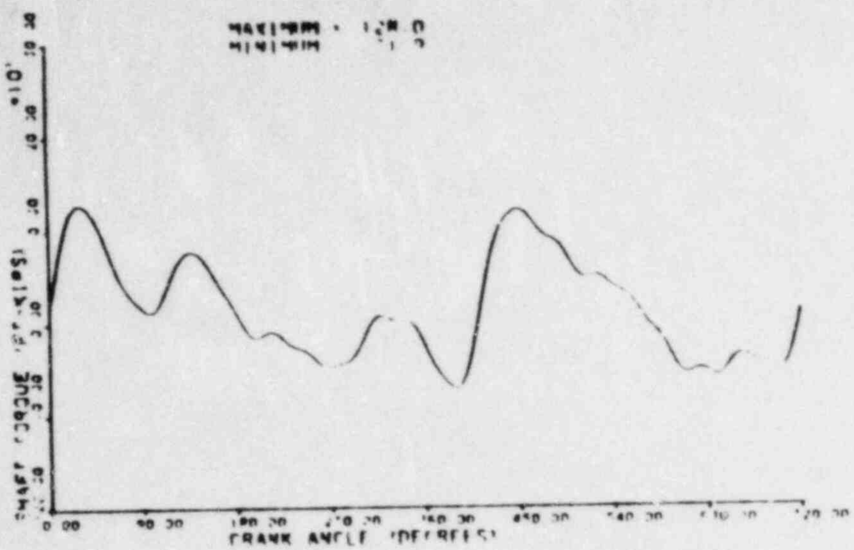
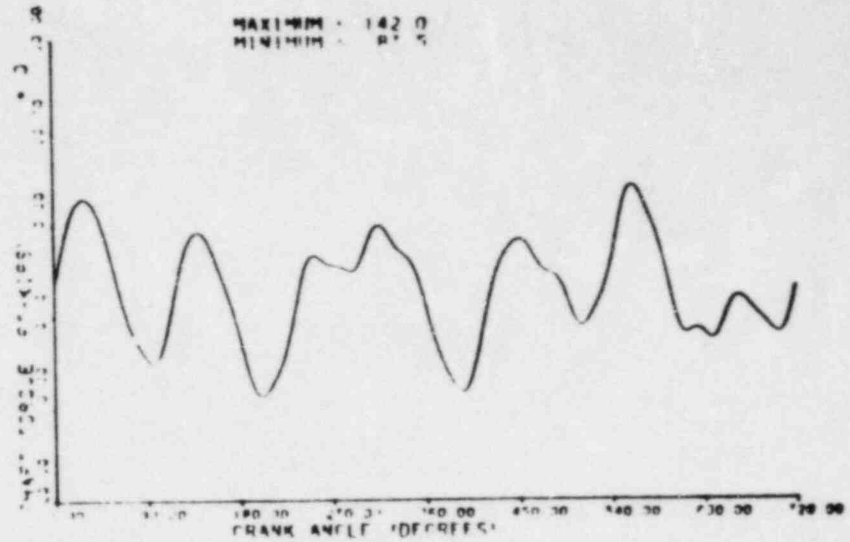


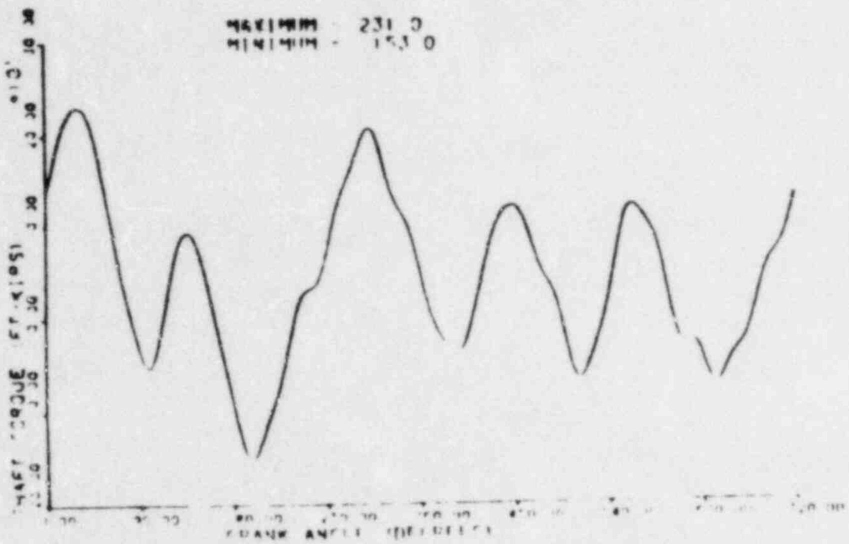
Figure 7-1. Measured pressure versus crank angle at 100% load.



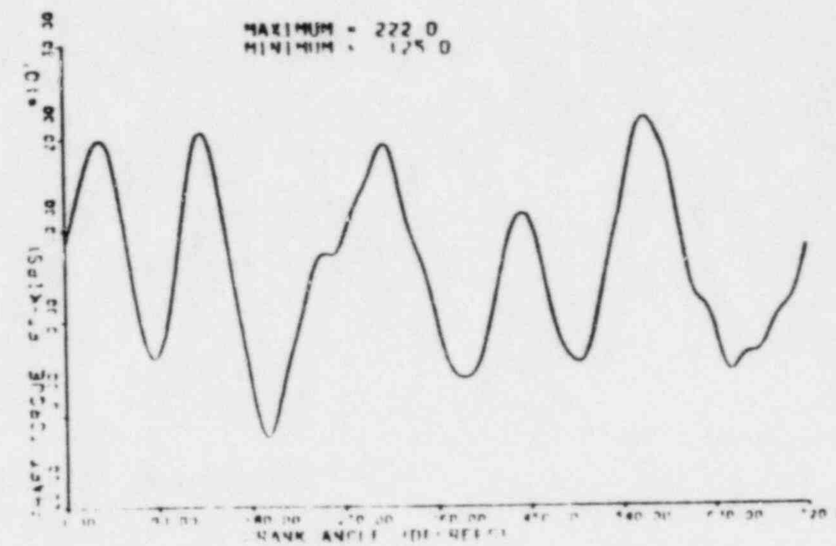
a) Cylinder 1 to 2.



b) Cylinder 2 to 3.

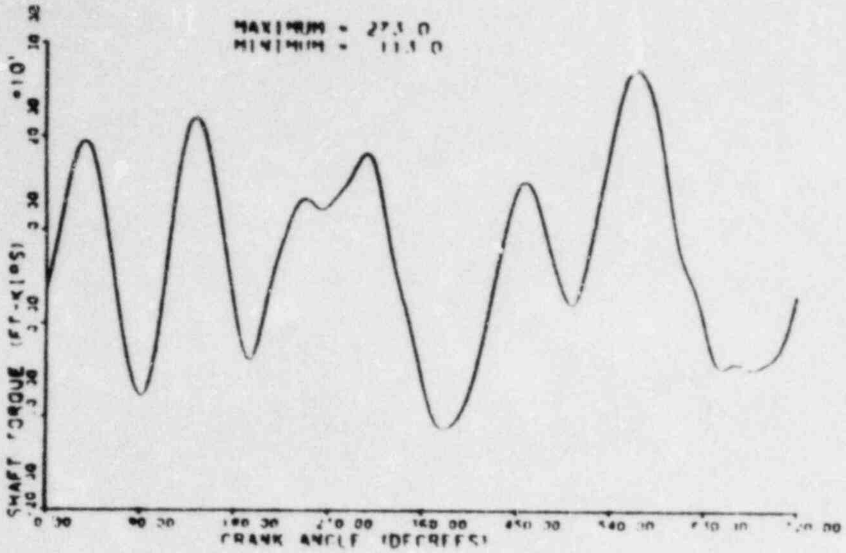


c) Cylinder 3 to 4.

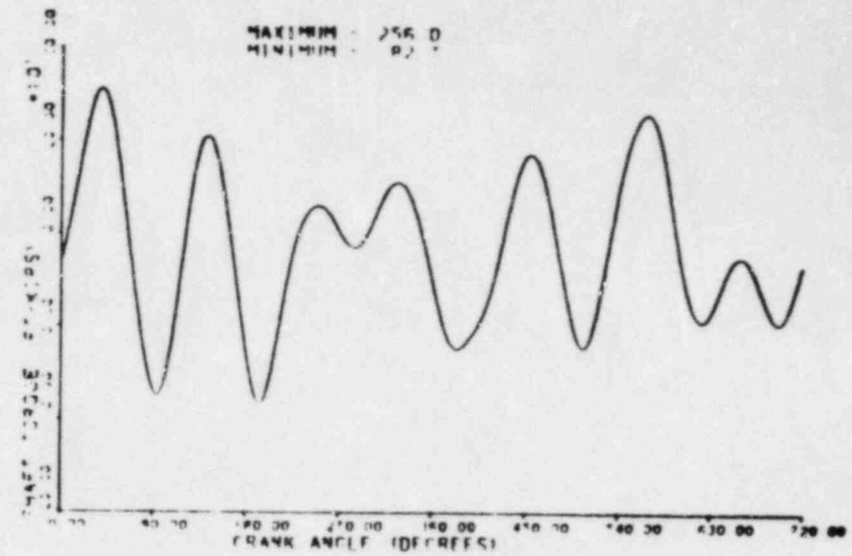


d) Cylinder 4 to 5.

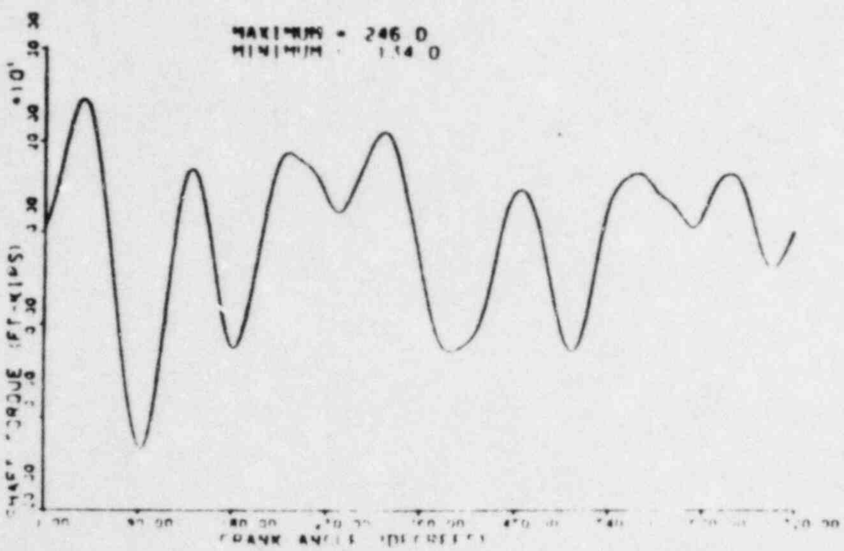
Figure 7-2. Dynamic model torsional response at 100% load for Grand Gulf 13 inch by 13 inch crankshaft.



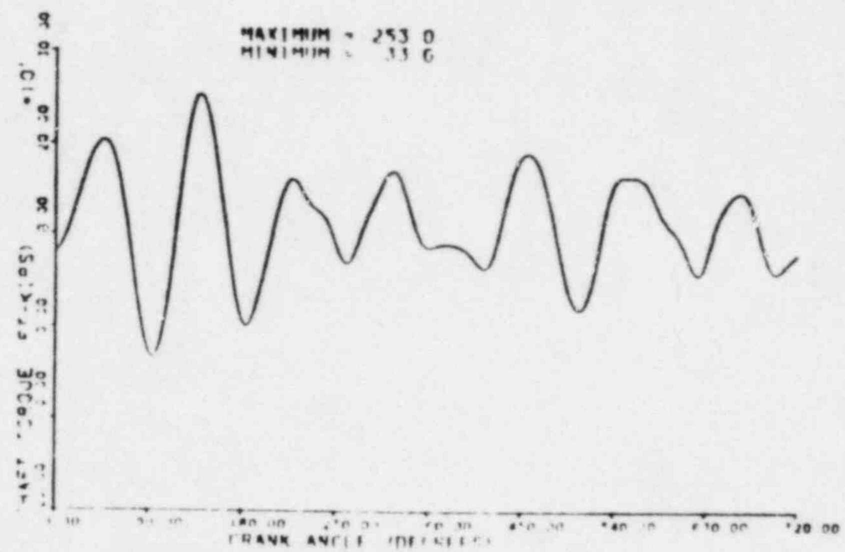
e) Cylinder 5 to 6.



f) Cylinder 6 to 7.



g) Cylinder 7 to 8.



h) Cylinder 8 to flywheel.

Figure 7-2 (continued).

8.0 RECOMMENDATIONS AND CONCLUSIONS

The crankshafts are adequate for their intended service provided the recommendations below are followed.

The following recommendations are made:

1. The oil holes in the main journals numbers 4, 6, and 8 represent a more critical stress concentration in torsion than the crankpin fillets and should be inspected for fatigue cracks and machining discontinuities.

2. The engines should not be allowed to operate below 440 rpm except during startup and shutdown.

3. The adequacy of the TDI specification for balancing of cylinders, in combination with the speed tolerances allowed by the governor should be determined by torsiograph testing. Once the expected ranges of speed and balance and their effects on stress have been established, it may be possible to eliminate inspection of oil holes. If an engine is operated in a severely unbalanced condition, it may be necessary to reinspect the oil holes for fatigue cracks.

The following conclusions are made:

1. The design calculations originally performed by TDI are appropriate and show that the crankshaft stresses are below DEMA recommendations for a single order. The stress resulting from combined orders is not calculated by this method.

2. The TDI torsiograph test results on the GGNS engine show that the crankshaft stresses are below the DEMA recommended levels for a single order for both 7000 kW (100% rated load) and 7700 kW (110% load). The peak-to-peak response was not measured.

3. The torsiograph test recently performed on a DSRV-16-4 diesel generator set at the Duke Power Catawba Nuclear Generating Station shows the peak-to-peak crankshaft stresses to be within the DEMA recommendations.

APPENDIX A

Component Design Review
Task Description

COMPONENT DESIGN REVIEW

CRANKSHAFT
PART NO. 03-310A

Classification A
Completion 3/5/84

PRIMARY FUNCTION:

The crankshaft converts reciprocating motion, component inertial forces and gas pressure piston forces to rotary motion and torque at the output flange.

FUNCTIONAL ATTRIBUTES:

1. Structural stiffness of the crankshaft must be sufficient to maintain acceptable states of stress in the crank pin web and main journal areas and to maintain system natural frequencies which are sufficiently removed from engine operating speeds. The crankshaft design should also be sufficient to withstand normal main bearing misalignments inherent in service.

2. The journal area of the main and connecting rod (crank pin) bearing must be sufficiently large for proper bearing oil film pressure but the journal length must be sufficiently short to prevent end wear of the bearing sleeves.

3. The material of the crankshaft and the surface finish should be sufficient to resist fatigue crack initiation.

SPECIFIED STANDARDS:

1. IEEE

2. ASTM

3. DEMA

EVALUATION:

1. Review TDI calculations and tests

2. Conduct engine test of 13 x 12 shaft

3. Conduct modal superposition and Holzer torsional analyses of:
   a. SNPS (R-48)
   b. GGNS (RV-16)
   c. Midland (RV-12)
   d. San Onofre (RV-20)

4. Conduct finite element analysis of R-48 12-inch crankpin fillets

5. Compare measured and calculated stresses R-48 13 x 12 shaft

6. Compare measured and calculated output torque and free end torsiograph traces for R-48.

7. Compare stress levels with endurance limit for R-48

8a. Compare nominal stresses of R-48 and RV-16 with those recommended by other standards.

b. Compare nominal stresses of RV-12 and RV-20 with those recommended by various organizations.

9. Complete final report on SNPS and GGNS crankshaft integrity.

10. Complete final report on Midland RV-12 and San Onofre RV-20

REVIEW TDI ANALYSES:

1. Experimental stress analysis (static) of DSR-46 crankshaft

2. Torsiograph tests

3. Holzer Table calculations

INFORMATION REQUIRED:

1. TDI drawings for DSR-48 and RV engines

2. Test reports for DSR-48 and RV engines

3. Original Holzer calculations and revisions for R-48 and RV-16, RV-12 and RV-20 engines

4a. Experimental pressure vs. time curve for R-48 and RV-16 engines.

b. Experimental pressure vs. time curve for RV-12 and RV-20 engines.