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ENCLOSURE 2
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ANALYSIS OF THE REPLACEMENT CRANKSHAFTS FOR EMERGENCY DIESEL GENERATORS SHOREHAM NUCLEAR POWER STATION

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1.0 EXECUTIVE SUMMARY

1.1 Introduction

The crankshaft in one of the three emergency diesel generator sets installed at the Shoreham Nuclear Power Station has been found to be severed, and cracks have been observed in the crank pin fillets of the other two crankshafts. The diesel engines are "Enterprise" Model DSR-48, manufactured by Transamerica Delaval, Inc.

A companion report has been issued on the failure analysis of the crankshafts. That report concluded that the cracking problem was the result of inadequate design, which led to high cycle torsional fatigue of the crank pin fillets. It was also noted that, while the surface machining of the fillets did not contribute to the failure, the fillets had not been shotpeened or rolled, procedures which are commonly performed to improve the fatigue resistance of crankshafts.

Three new crankshafts of modified design have been delivered for the SNPS engines. The principal modification is an increase in the crank pin diameter from 11 inches to 12 inches. In addition, the stress concentration in the pin fillet radius is reduced by increasing the pin fillet's radius from one-half inch to three-quarter inch.

As part of the emergency diesel generator recovery program, Failure Analysis Associates (FaAA) performed a complete evaluation of the integrity of the new crankshafts. For the analysis of the 13 × 11 crankshaft, FaAA had developed analytical procedures for predicting detailed dynamic response of the generator set and stress distributions in the fillet region and had confirmed these procedures by strain gage measurements on an operating engine. Therefore, it was decided to apply these same analytical techniques to the new crankshaft. Also, given Transamerica Delaval's use of new input data, it was possible to compare the manufacturer's design stress levels with industry standards. FaAA also performed an independent analysis to check compliance with industry standards.

1.2 Conclusions

The increase in diameter from 11 inches to 12 inches both stiffens the crankshaft, thus reducing the amplitude of oscillating torque throughout, and reduces the average crank pin stress resulting from the application of the torque. Comparisons between the accepted industry design standards, the manufacturer's design stress level, and FaAA's calculations from the modal superposition analysis are summarized in the following table.

Method of Analysis	Average Torsional Stress (psi) Due to Single 4th Order	Average Torsional Stress (psi) Due to Summation of Orders
TDI Analysis	2990	Not Calculated
FaAA Modal Superposition	3300	5640
DEMA Recommendation	< 5000	< 7000

This comparison clearly shows that the new design meets accepted industry standards. In contrast, the design of the original 13 × 11 crankshaft was found to exhibit stress levels in excess of 5,000 psi for the fourth order and in excess of 8000 psi for the summation of orders.

Detailed stress analysis was conducted to compare the range of maximum principal stress in the most highly-stressed fillet of the new crankshaft with the stress range at the locations of the fatigue cracks in the old shafts. Stresses were greatly reduced from a combination of lower torque, larger cross section, and more generous fillet radii. The maximum stress range is predicted to be 37 ksi, compared with 66 ksi at the location of the fatigue crack. This factor of 1.78 difference in stress is considered adequate to provide indefinite fatigue life for the new crankshafts, even with the same machined surface condition as the old crankshafts. This conclusion is based upon the fact that the pin fillets of the three, cracked crankshafts were subjected to at least 10^7 cycles of full-load torque oscillations, which gives the best possible confirmation of the minimum fatigue properties for crankshafts similar to the original crankshafts.

Following the identical analysis procedures that showed the old crankshaft design to be inadequate, it is predicted that the new shafts with larger crank pins will have an unlimited life without fatigue failure from torsional stress. Additional margin against fatigue cracking has been provided by shot-peening the fillets. The reduced stress levels will be verified by engine testing with an instrumented crankshaft to confirm the conclusion that these crankshafts will have an unlimited life.

1.3 Additional Measures

To provide an additional margin of fatigue resistance, LILCO decided to have all the crankshafts shotpeened in accordance with common industry practice. Shotpeening workhardens the surface and results in a layer of residual compressive stress, both of which retard the initiation of fatigue cracks. Furthermore, the peening process eliminates machining marks.

Finally, LILCO and FaAA will instrument the crank pin fillets of one new shaft with strain gages to verify the reduction in vibratory stress at full load and during transient excursions.

2.0 INTRODUCTION

As a result of fatigue damage in the crankshafts of three emergency diesel generator sets at Shoreham Nuclear Power Station, new crankshafts of modified design have been installed. The principal modification is an increase in pin diameter from 11 inches to 12 inches. As a part of the program to recover from the failure, Failure Analysis Associates was engaged to investigate the adequacy of the new design.

A detailed failure investigation presented in a companion report attributed failure of the original crankshaft to high cycle torsional fatigue resulting from inadequate design. Because of the many common features (e.g., material, throw arrangement, and spacing) of the new and old designs, the same methods employed to predict failure of the original design were applied to analyze the redesigned crankshaft. In particular, the same dynamic analysis approaches, which showed that the original design was overstressed by equipment that met industry standards, were followed to compare the new design with those industry standards. Similarly, the same dynamics analysis approach, which was shown to conform to the measured torsional response of an operating engine with an 11-inch crank pin, was used to calculate the torsional response of the redesigned crankshaft. Results of this dynamic response analysis are in the form of the vibratory torque versus time history experienced by each crankshaft throw.

As in the failure analysis of the original design, a finite element analysis was performed to determine the detailed state of stress throughout the crankshaft throw that experienced the highest torque. Then, employing the same evaluation method that predicted fatigue failure in the crank-pin-to-web fillet radius for the original design, the fatigue behavior of the peak stress location was ascertained. It was determined that the redesigned crankshaft should have an unlimited life without fatigue failure due to torsional stress.

3.0 ANALYSIS OF DIESEL GENERATOR DYNAMIC RESPONSE

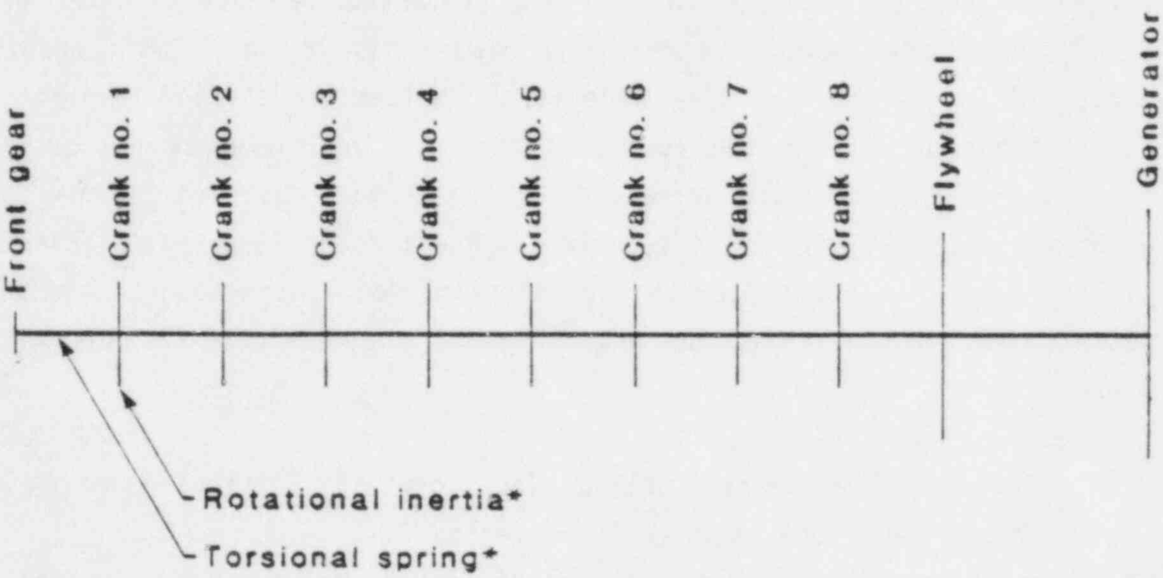
A discussion of diesel generator crankshaft dynamic response analysis and its application to a failed 13 × 11 design is discussed in a companion volume. In this section, results of a torsional dynamic response analysis of the redesigned (13 × 12) crankshaft are presented, without detailed explanations, under the assumption that the reader has in hand the companion report. First, a review will be made of the Holzer and forced vibration design calculations of TDI for the 13 × 12 redesign. The review of the design calculations will be followed by results of FaAA's more detailed dynamic analysis using the modal superposition technique. The torques computed in this section will be used in the next section to calculate local stresses. These analyses demonstrate that stresses on the 13 × 12 crankshaft are below allowable values.

3.1 Review of Transamerica Delaval Inc. Torsional Critical Speed Analysis

The Holzer model used by Transamerica Delaval Inc. (TDI) [1] is shown in Figure 3-1 and the inertia and stiffness values are shown in Table 3-1. The first natural frequency, excluding the rigid body mode, was found to be 38.7 Hz.

In a second analysis step, the response was calculated by TDI for each order of vibration separately. The response is then calculated by one procedure if the harmonic is at resonance and by another if the harmonic is away from resonance.

The purpose of the calculation at resonance was to ensure that the diesel generator could be brought up to operating speed without undergoing excessive stresses as critical speeds were passed. Based on the TDI results and experience with the 13 × 11 design, it is expected that excessive vibration as critical speeds are passed will not be a problem for this engine. Since the engine runs at 450 rpm and the nearest significant critical is



*See Table 3-1 for values of rotational inertias and torsional springs.

Figure 3-1. TDI Holzer model.

TABLE 3-1

Stiffnesses and Inertias for Dynamic Analysis [1]

Inertia Location	Inertia ($lb.ft.sec^2$)	Torsional Stiffness (ft. lb./rad)
Front Gear	6.8	
		58.1×10^6
No. 1	49.2	
		84.7×10^6
No. 2	47.9	
		84.7×10^6
No. 3	47.9	
		84.7×10^6
No. 4	47.9	
		84.7×10^6
No. 5	47.9	
		84.7×10^6
No. 6	47.9	
		84.7×10^6
No. 7	47.9	
		84.7×10^6
No. 8	50.1	
		76.9×10^6
Fly Wheel	1100.1	
		276.8×10^6
Generator	2650.4	

580 rpm, the calculated response at resonance need not be further considered. Away from resonance, at the operating speed of 450 rpm, the 4th order harmonic has an exciting frequency of 30 Hz, and since the first natural frequency is 38.7 Hz, a dynamic amplification of 2.51 is obtained. This value is not significantly reduced by small amounts of damping. The nominal shear stress in the 12 inch pin for each order is then calculated from the dynamic torque, T, using Tr/J where r is the pin radius and J is the polar moment of inertia.

The Diesel Engine Manufacturers Association (DEMA) [2] recommends that the superimposed stress created by a single order of vibration be less than 5,000 psi and that the superimposed stress created by major orders of vibration, which might come into phase periodically, be less than 7,000 psi.

The normalized applied torques for each order, T_n, are shown in the TDI design calculations [1]. Of particular importance are the 4th order values since this order produces the highest stresses in the crankshaft at 450 rpm. The 4th order values used by TDI [1] are compared with those recommended by Lloyd's Register of Shipping (LRS) [3] in Table 3-2 and are in agreement.

Table 3-3 shows that the 13 × 12 crankshaft does satisfy the DEMA recommendation for a single order of vibration [2]. The 7,000 psi combined stress recommendation is also met, as will be shown subsequently.

3.2 Failure Analysis Associates Dynamic Torsional Model

FaAA developed a dynamic torsional model of the crankshaft to overcome limitations in TDI's forced vibration calculations. For instance, the TDI method does not compute the phase relationship between the various orders of response so that it is not possible to compute the summation of all orders. The actual maximum stress is a direct result of this summation.

The dynamic model developed used the same idealized lumped inertia and torsional spring model as the TDI analysis. The inertias and stiffnesses will be verified by a variable speed torsiograph test of the 13 × 12 crankshaft. Figure 3-2 shows the model used which has one additional spring placed between the generator and grid to represent the effect of the grid on dynamic

TABLE 3-2

Comparison of 4th Order T_n Values

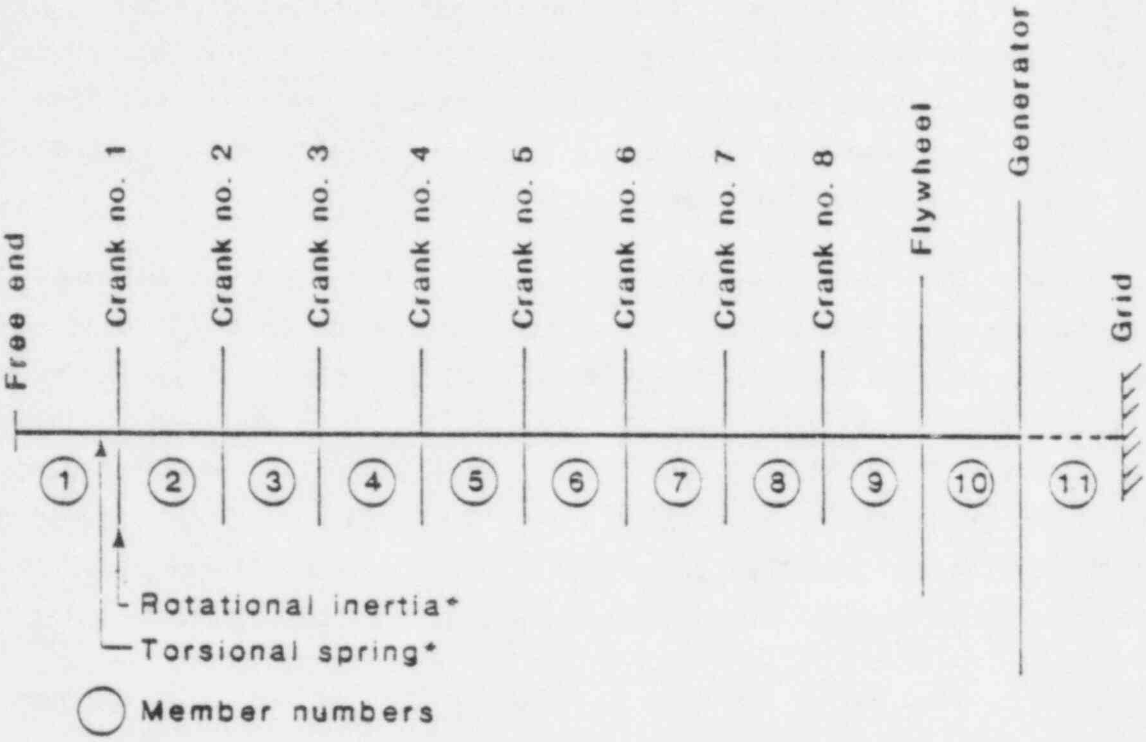
Source	T_n Value (psi)
LRS [3]	30.4*
TDI 12" design [1]	27.7
FaAA	28.7

*Phase for each T_n is not given by LRS [3].

TABLE 3-3

Nominal Shear Stress Due to 4th Order Loading

	Nominal Shear Stress (psi)
TDI 12-inch design [1]	2,990
DEMA allowable [2]	5,000



*See Table 3-1 for values of rotational inertias and torsional springs.

Figure 3-2. Dynamic model.

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 asynchronous operation. The remaining model parameters were the same as those shown in Table 3-1.

The torsional natural frequencies for the 13 × 12 crankshaft are shown in Table 3-4. The natural frequencies for asynchronous operation agree with the values computed by TDI. There is no longer a rigid body rotation mode due to the connection to the grid. The first natural frequency was found to be 2.93 Hz. The remaining natural frequencies shift slightly upward but are essentially unchanged (Table 3-4).

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 [4] was used with harmonic load input. The harmonic loads were assumed to consist of three parts, gas pressure, reciprocating inertia, and friction. Each load was assumed to act on the piston and was then converted to a harmonic torsional load using the geometric relationships between the crank angle, piston, and connecting rod.

The gas pressure loading is obtained from an indicator diagram. TDI did not provide an indicator diagram from which T_n values could be obtained. For this reason, FaAA used a theoretical indicator diagram for this engine. The diagram was based on the brake horse power of the engine and the cylinder peak pressure, and does not depend on the crankshaft design. This indicator diagram (the same as that used in the 13 × 11 analysis) was used to calculate the T_n values for FaAA's analysis. Since the 4th order accounts for considerably more than half the vibratory torque induced in the crankshaft, Table 3-2 compares the 4th order T_n values computed by Lloyd's Register of Shipping [3], FaAA, and TDI [1] designs. Table 3-2 shows that FaAA, LRS, and TDI's 12-inch design values of T_n are in good agreement.

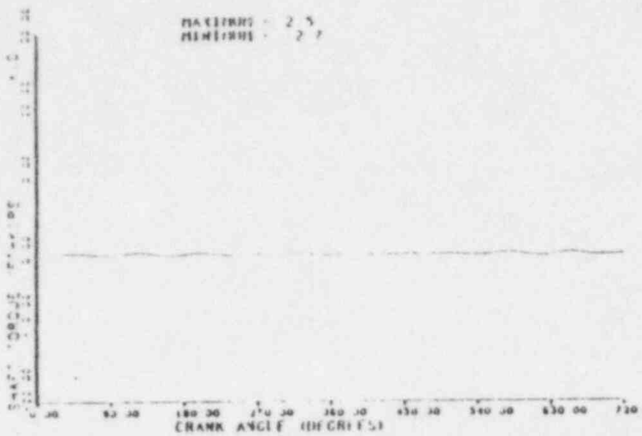
The torsional response for synchronous operation through two engine revolutions at full load is shown for each member in Figure 3-3 (refer to

TABLE 3-4

Natural Frequencies of 13 × 12 Torsional System

Mode	Natural Frequency (Hz) Synchronous	Asynchronous
1	2.93	0.00*
2	38.73	38.72
3	92.94	92.92
4	116.67	116.67
5	184.33	184.33

*Rigid body rotation



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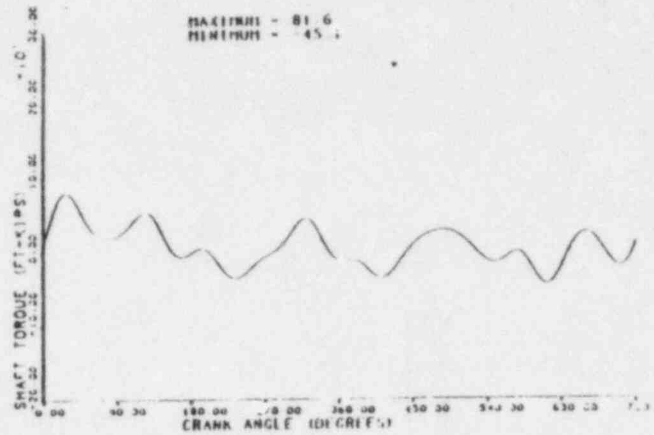
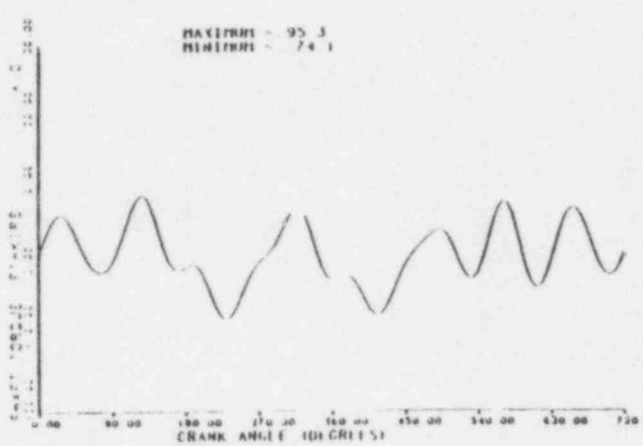


(b) Member 2.



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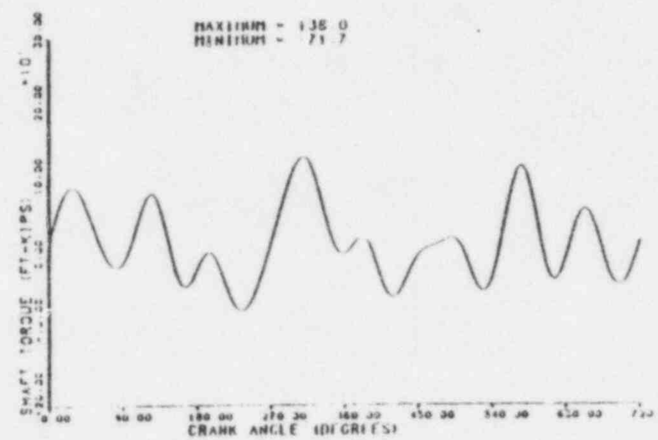
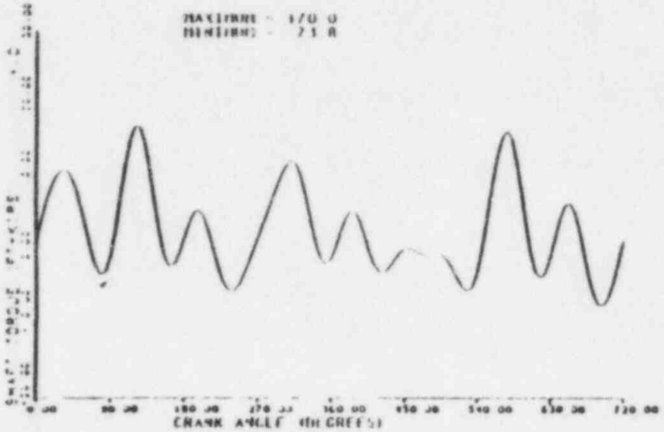
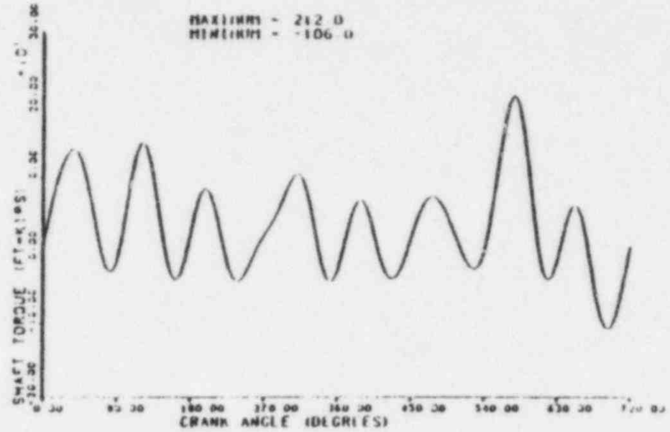


(d) Member 4.

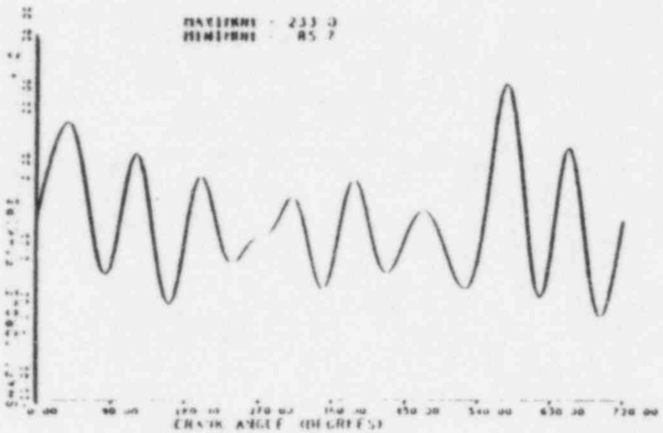
Figure 3-3. Dynamic model response of 13 x 12 diesel generator at full load.



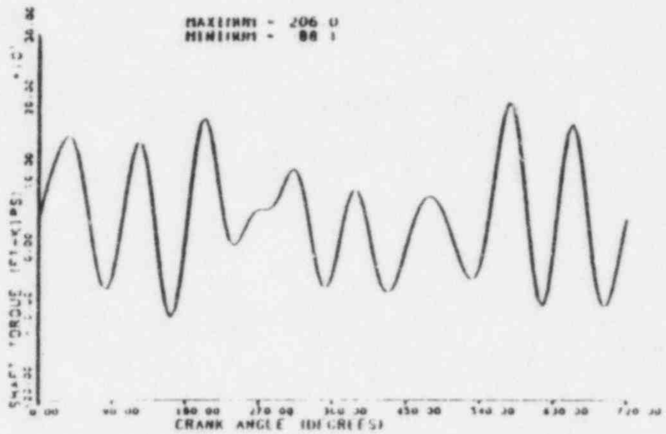
(e) Member 5.



(f) Member 6.

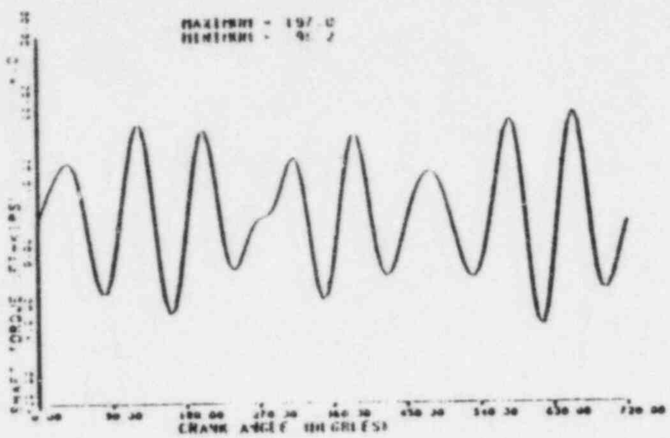


(g) Member 7.

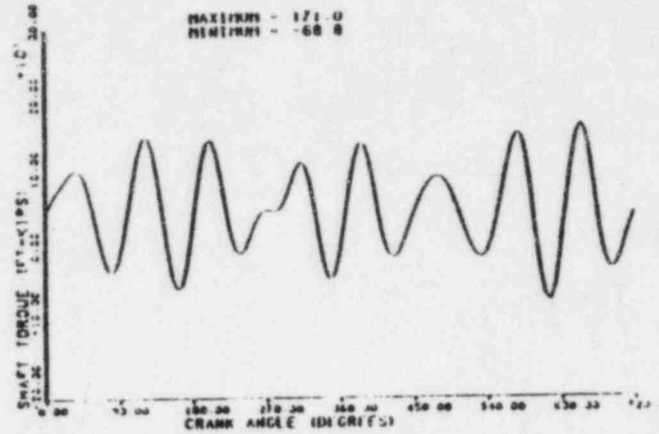


(h) Member 8.

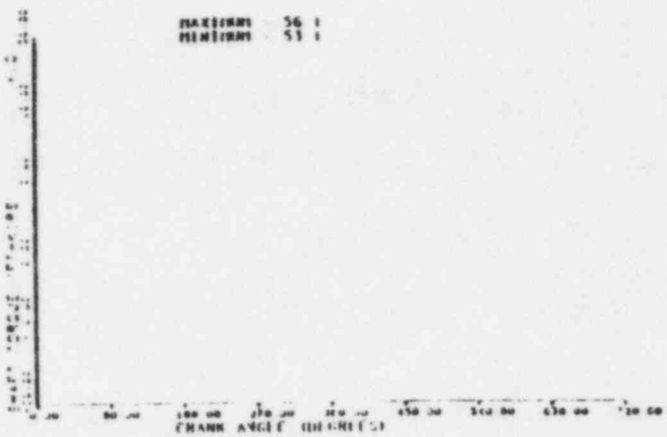
Figure 3-3 (continued). Dynamic model response of 13 x 12 diesel generator at full load.



(i) Member 9.



(j) Member 10.



(k) Member 11.

Figure 3-3 (continued). Dynamic model response of 13 x 12 diesel generator at full load.

Figure 3-2 for "Member" designations). The zero degree crank angle is the location where crank No. 1 is at top dead center during its firing cycle. It can be seen that the shaft torque is low near the free end of the diesel generator. It becomes largest in the area from piston No. 5 to the generator. Peak torques as high as 233 ft-kips and torque ranges of 319 ft-kips are experienced. The steady output torque of the diesel generator is 55 ft-kips.

The effect of differences between synchronous and asynchronous operation of the diesel generator on torsional vibratory torque was found to be insignificant in the 13 × 11 analysis, and the same is true for the redesigned crankshaft.

Table 3-5 shows the single amplitude of alternating nominal shear stress (calculated by Tr/J) at the generator end of the engine (member No. 9) and at the maximum location (member No.7). The contribution of the 4th order is also shown in this table. This contribution increases as one moves toward the generator as predicted by TDI's analysis. The maximum value of 3,300 psi is in agreement with TDI's calculation of 2,990 psi. The difference between these values is due to the use of only one mode by TDI and the small difference between TDI's and FaAA's T_n values for the 4th order. This table clearly shows that the nominal design stresses in the original 13 × 12 crankshaft satisfy the 5,000 psi recommendation for a single order and the 7,000 psi recommendation for combined orders specified by DEMA [2].

The free end amplitude for the 4th order and total response is shown in Table 3-6. These values may be compared to a torsiograph test when the DG-101 engine is field tested with a new 13 × 12 crankshaft installed. The amplitudes are shown in Table 3-6 to be considerably reduced from those which occurred in the 13 × 11-inch crankshaft.

TABLE 3-5

Amplitude of Nominal Shear Stress in 12-inch Crank

Member Between	Amplitude of Nominal Shear Stress (psi)	
	4th order	Total
Piston 6 and Piston 7	2,750	5,640
Piston 8 and Flywheel	3,300	5,180
DEMA allowable	5,000	7,000

TABLE 3-6

Free End Amplitudes

	4th Order (degrees)	Total (degrees)
Dynamic Model of 13 × 12	0.30	0.53
Dynamic Model of 13 × 11	0.49	0.84

Section 3 References

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

3. Lloyd's Register of Shipping, Guidance Notes on Torsional Vibraion Characteristics of Main and Auxillary Oil Engines.

4. Timoshenko, S., D.H. Young, and W. Weaver, Jr., Vibration Problems in Engineering. 4th ed., Wiley, 1974.

4.0 CRANKSHAFT STRESS ANALYSIS

Stress analyses were performed on the 13 × 12 crankshaft (redesign) configuration in a manner similar to that reported for the 13 × 11 crankshaft (original design) in Reference 1. Results of these analyses are reported in this section, under the assumption that the reader has in hand the 13 × 11 crankshaft failure report.

4.1 Finite Element Model

Typical structural dimensions of one throw of the redesign crankshaft are shown in Figure 4-1. The most significant changes are the pin diameter change from 11 inches to 12 inches and modifications of the fillet radius detail where the crank pin meets the web. Material properties are the same as for the original design.

The 13 × 12 design is subjected to loads similar to those for the original design, except that the magnitude is considerably less, as discussed in Section 3. Since torque was shown to be the dominant load in that design, only the results of the applied torque load case are reported here. These torque loads are taken from the dynamic response analysis reported in Section 3.

The stress analysis for the redesign was performed using the computer program MARC, K.1-1 Version. All planes of symmetry used were identical to those for the analysis of the original design.

The same finite element and basic mesh were used for the redesign model as for the original design model. The crank-pin-to-web fillet model was changed to reflect the change in design details as shown in Figure 4-1, Detail A. Figure 4-2 shows an overall view of the model and the coordinate system used.

Since it was shown in the analysis of the original design that the Case 2 boundary conditions were conservative and close to the actual behavior, only the Case 2 results are reported for the 13 × 12 analysis. The torque load on the main journal was applied in an identical manner to that in the 13 × 11 analysis.

The same 1.08 factor, an adjustment to account for mesh refinement, was used to scale the finite element results. For the same reasons as discussed in the 13 × 11 analysis, it was not considered necessary to combine stresses from piston forces on the crank pin with those from the torque load.

4.2 Results

Stresses, obtained from applying the unit torsional rotation, were scaled to represent positive and negative torques of 233,000 and - 85,700 ft-lbs, as determined in Section 3, then scaled in the fillet region by 1.08 to account for mesh refinements. From the element integration points, stresses were extrapolated to the surface. Figure 4-3 shows the circumferential variation of the largest principal stress component for maximum and minimum torque conditions. Figure 4-4 shows the axial variation of maximum principal stress in the fillet radius at the peak-stress, circumferential location. Stresses on the negative z side of the crankshaft (see Figure 4-2) are equal to those presented, but of opposite sign.

4.3 Conclusions

In the analysis of the original design, it was shown that the fatigue cracks that have been observed are predictable by comparing stress analysis model results, laboratory (full scale specimen) fatigue data, and strain gage data from an engine operating at full power. More important, the fatigue cracks observed in the field provide the most accurate means possible for ascertaining the fatigue limit of crankshafts in this application.

The range of cyclic maximum principal stress, predicted in the fillet radii of the redesign crank pin, is 37 ksi compared to 66 ksi for the original design crank pin; 66 ksi is taken as the field-proven fatigue limit. This ratio (66/37 = 1.78) provides an ample factor of 1.78 below the fatigue limit for the 13 × 12 crankshaft design. The redesigned crankshaft should be suitable for unlimited life without fatigue failure due to torsional stress.

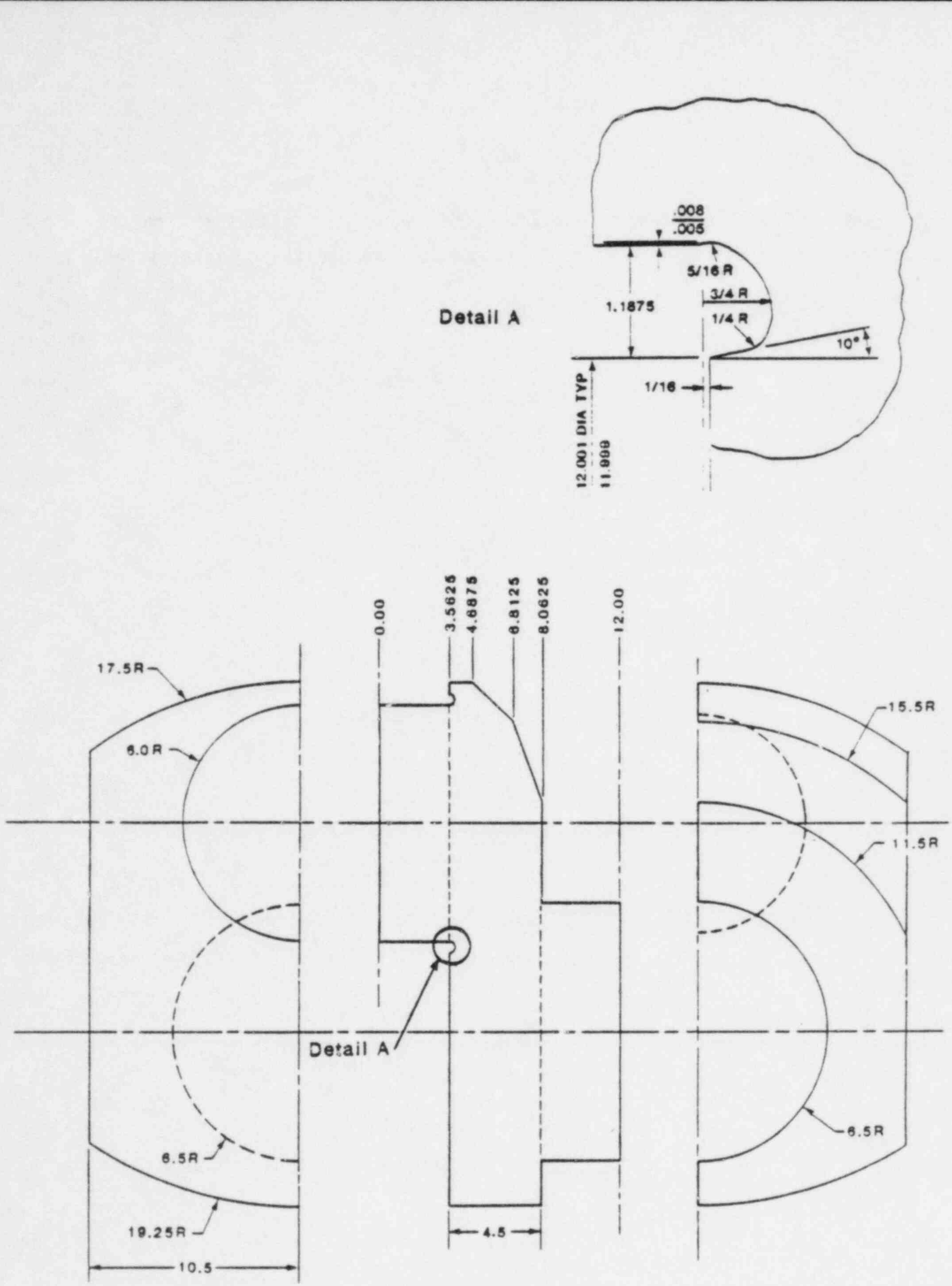


Figure 4-1. Typical structural dimensions of the crankshaft.

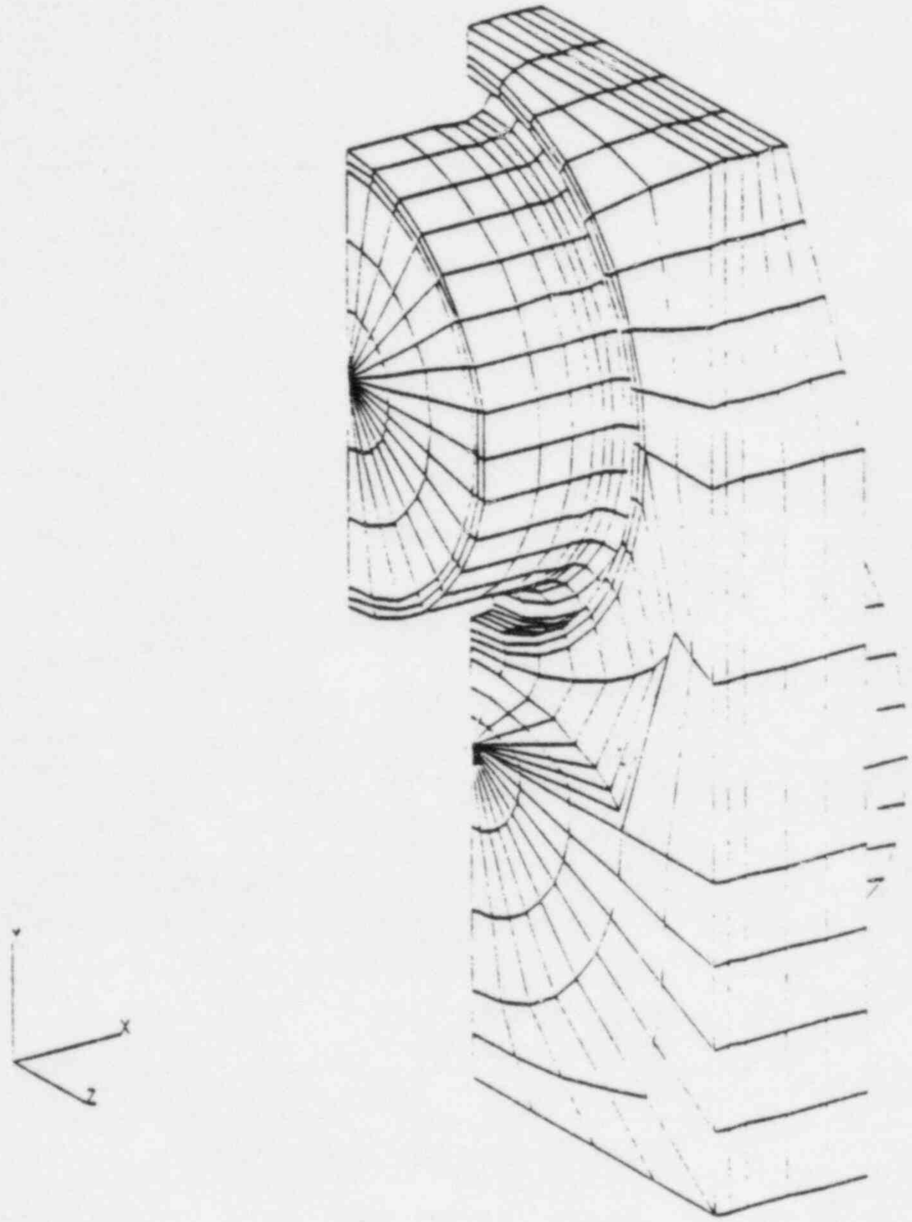


Figure 4-2. Three dimensional view of the finite element model.

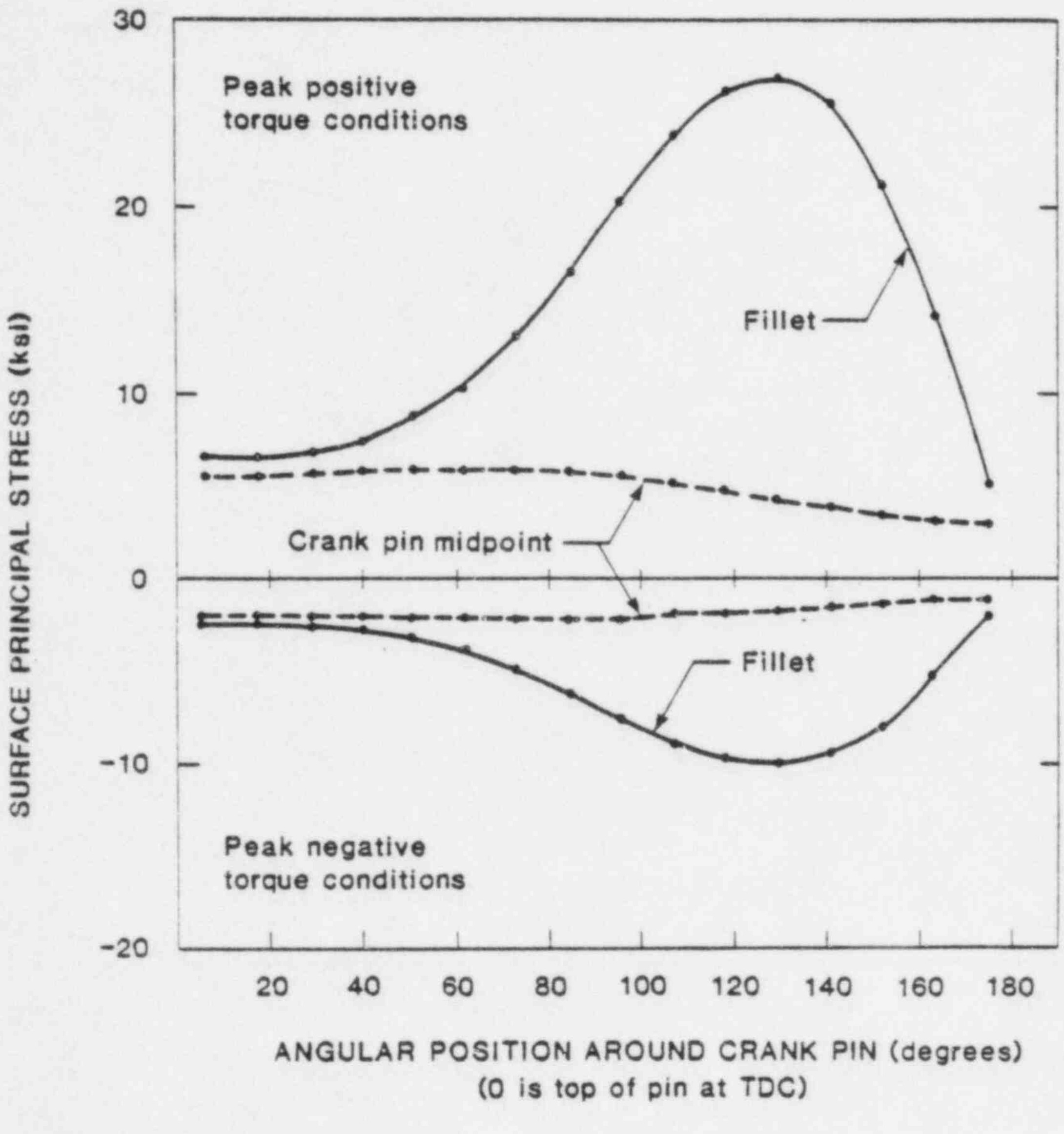


Figure 4-3. Circumferential variation of maximum principal stress for case 2 boundary conditions.

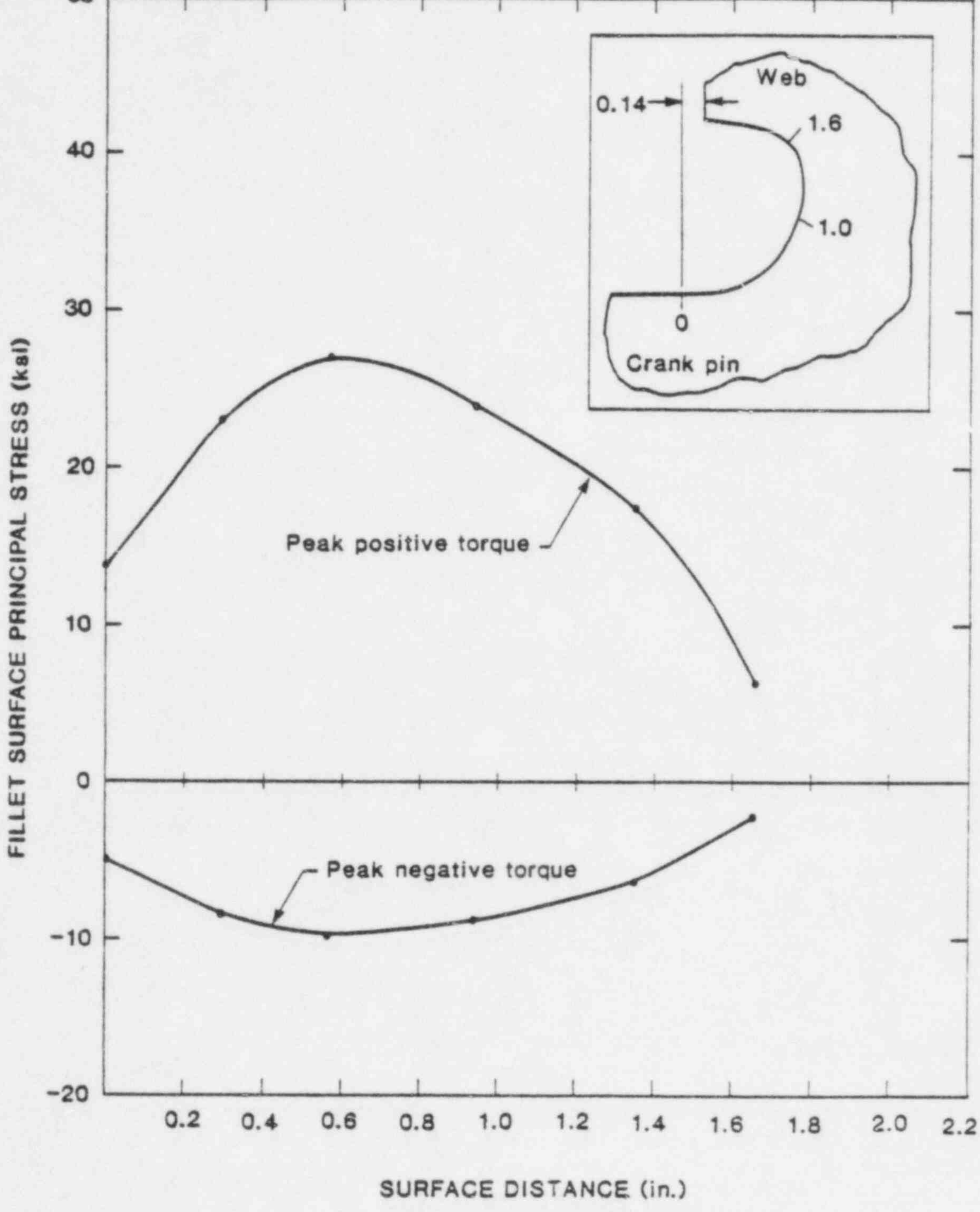


Figure 4-4. Axial variation of maximum principal stress, Case 2.

5.0 DISCUSSION

During investigation of fatigue cracking of the original 13 × 11 crankshaft design, analytical models, which were consistent and in close conformity to measured crankshaft behavior, were developed. A torsional vibration model was in close agreement with measured output, oscillatory torque and with torsiograph (angular displacement) measurements at the free end of the crankshaft. This close agreement was achieved in amplitude as well as in signature. The signature of strain gages mounted in a fillet radius of the No. 5 crank pin was also closely matched by the torsional vibration model. A detailed finite element model of one crankshaft throw correctly predicted the magnitude and orientation of the maximum principal stress as measured by these strain gages. Finally, the range of maximum principal stress, as predicted by the model and measured by the strain gages, fell within the range of fatigue limits determined by others on full-scale crankshafts.

These several comparisons, which yield close agreement between the analytical methods used by FaAA and with measured behavior of the 13 × 11 crankshaft, give high confidence that the same methods used in this evaluation correctly predict the behavior of the redesigned 13 × 12 crankshaft. This confidence is further enhanced by the observation that the predicted dynamic behavior and the predicted detailed distribution of stresses for the 13 × 12 shaft have the same overall features as for the 13 × 11 crankshaft design. Thus, extrapolation of the models outside the regime of proven performance was not required.

Application of the proven models to the redesigned crankshaft showed that the torsional response was considerably reduced. The maximum range of oscillatory torque on any throw was reduced from 388 ft-kips to 319 ft-kips. This reduction, in connection with additional stress reduction from the larger cross-section and more generous fillet radius detail, reduced the range of maximum principal stess from 66 ksi to 37 ksi.

The range of maximum principal stress predicted for the crank pin fillets represents a reduction by a factor of 1.78 from the levels for the original crankshaft design. Since the cracked crankshafts had been subjected to around 10^7 cycles of the largest stress oscillations, the stress amplitudes in the fillet radii were only slightly in excess of the fatigue limit for the material and the machined surface. Therefore, the reduced stress in the new crankshafts would, by itself, be expected to insure an unlimited life without fatigue failure from torsional stresses.

The variability of surface roughness and residual stress in machined surfaces, along with material properties, govern the fatigue limit. This variability probably accounts for the fact that no cracking occurred at certain locations of maximum stress in the original crankshafts. Such variability is to be expected for large, machined components. Consequently, it was decided to shotpeen the fillets in order to produce a consistent, high level of compressive residual stress in the surface and to eliminate machining marks. In addition, shotpeening work hardens the surface and raises the fatigue limit. To provide additional assurance of structural integrity, the fillets will be inspected by a high-resolution, eddy-current method after the break-in run.

Finally, the proven torsional vibration model showed that the redesigned crankshaft meets industry standards with respect to maximum, nominal stress amplitude. This provides additional assurance that fatigue performance will be satisfactory.