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PEI CONSULTANTS	Crankshaft Torsional Stress Calculations for 8 L 17x21 Engine-Generator Set	July 16, 1984

CRANKSHAFT TORSIONAL STRESS CALCULATIONS FOR 8 L 17x21 ENGINE-GENERATOR SET

WISCONSIN PROFESSIONAL ENGINEER

BY

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July 19, 1984

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SUBJECT

Independent calculations are presented here concerning the torsional characteristics of the Shoreham TDI inline 8 cylinder, 17 x 21 engine with replacement crankshaft. The rating of the engine is 4889 hp at 450 rpm. The corresponding bmep is 225 psi. The TDI diesel engine is used as the prime mover to drive a 60 hz electric generator, rated at 3500 kw at 450 rpm. The diesel generator system is required to serve as the emergency power system for the Shoreham Nuclear Power Station. The replacement crankshaft has 13" main and 12" crankpin (13 x 12), the failed shaft has 13" main and 11" crankpin (13 x 11).

SCOPE

In October and November of 1983, PEI Consultants were asked by LILCO project management to evaluate the suitability of the TDI 8 L 17 x 21 engine with replacement crankshaft (13 x 12) for service intended at Shoreham Nuclear Power Station (SNPS). A preliminary analysis was made and it indicated the new placement crankshaft was satisfactory and passed the DEMA[1] criteria for stationary applications. The old crankshaft (13" x 11"), using the same analysis method, did not satisfy the DEMA criteria when the tangential effort (Tn) of the gas pressure, as listed by LLOYD[2] references, was used. In February of 1984, LILCO project management requested more detailed calculations for the replacement crankshaft. This report serves to finalize the updated torsional calculation on the replacement crankshaft.

1. "Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines," Sixth Edition, Diesel Engine Manufacturers Association DEMA, 1972.

2. "Lloyd's Register of Shipping:" Guidance Notes on Torsional Vibration Characteristics of Main and Auxiliary Oil Engines, 1976.

EXECUTIVE SUMMARY

1. The replacement crankshaft (13 x 12) is suitable for the intended service. The name plate rating of the engine is 3500 kw continuous at 450 rpm.
2. The replacement crankshaft satisfies the DEMA stationary engine crankshaft standard practice design criteria, which are:
   a. to insure that no harmful torsional virbratory stresses occur within five percent above and below the rated speed.
   b. for crankshaft 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 create by the summation of the major orders of vibration.
3. The general method used in this report is based on:
   a. the conventional Holzer-table forced vibration method.
   b. the advanced harmonic synthesis method[3] (modal superposition).

TORVAP, a common-domain torsional vibration analysis computer program, is employed. Program TORVAP R deals with natural frequency determination forced virbration. TORVAP C deals with the harmonic synthesis methods to determine the vibratory amplitudes and torques of masses and shafts that have been specified by the user. TORVAP program has built-in assumptions on magnifier factor (a function of damping) and on tangential gas pressure effort, based on user's input on horsepower, rpm, and mechanical efficiency.

3. "Torsional Vibration Analysis Program (TORVAP)," Computer Aided Design Centre, Madingley Road, Cambridge, UK CB3 OHB. June, 1975. Available from Comshare Inc., Ann Arbor, Michigan.

SUMMARY AND RECOMMENDATIONS

1. DEMA STATIONARY DIESEL STANDARD

The TDI 13 x 12 replacement shaft meets DEMA crankshaft design objectives and criteria. The maximum single order torsional stress at rated load (3500 kw) is at shaft section 9-10, close to the drive end. The maximum sum of orders torsional per TORVAP C is 5101 psi at shaft section between cylinders 5 and 5. All single order and sum of orders stresses are below DEMA limits.

DEMA Limit Single Order	TORVAP R 4.0 Order	TORVAP C 4.0 Order
5000 psi	3146 psi	3455 psi

DEMA Limit Sum of Orders	TORVAP C SRSS	TORVAP C Sum of (6)
7000 psi	3763 psi	5101 psi

2. ALLOWABLE SPEED RANGE

There are no dangerous torsionals predicted in the speed range between 95% and 105% of the synchronous speed at 450 rpm. The following results show that the highest torsional occurs at 95% rpm, 225 bmep. This is a lug-down condition. In this series of calculations, bmep is kept constant at 225 psi, which is the rating.

Constant 225 Bmep

		Sum of Orders	
Rpm	Single Order	Shaft 6-7	Shaft 9-10
472.5	4010 psi	5490 psi	5673 psi
450.0	3455 psi	5101 psi	4900 psi
427.5	3071 psi	5627 psi	6232 psi

3. OVERLOAD CONSIDERATION

The replacement crankshaft is safe at 3900 kw overload at 450 rpm, the maximum stress of single order is 3740 psi at shaft section 9-10, the TORVAP C sum of order torsional stress is 5401 psi at shaft section 6-7.

Bmep	Single Order Shaft 9-10	Sum of Orders Shaft 6-7	SRSS Shaft 9-10
251	3740 psi	5401 psi	4030 psi

Please see page I-2 for shaft section numbering system.

4. SUITABILITY FOR USE

Based on this detailed analysis, torsiograph data at Shoreham, and guideline, the shaft is adequate for the intended generator service for the SNPS. If this engine is a marine variable speed engine, the engine should have barred speed notice in the 410-430 rpm range for continuous operation. No barred speed notice is necessary for generator application.

5. CHECKING TEST DATA

LILCO has requested FaAA and SWEC to obtain actual dynamic torsiograph data on the replacement shaft. This analysis checks within 14% with those test data. This reasonable amount of the discrepancy could be att[illegible]uted to the following:

a. Experimental error - 10% to 15% is considered acceptable.

b. Tn assumption - PEI uses LLOYD's Table as included in TORVAP software.

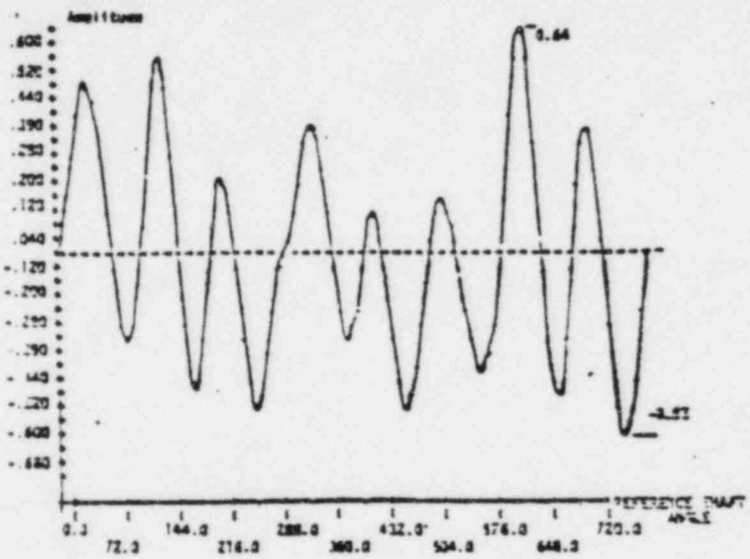
c. Magnifier factor assumption - PEI uses built-in TORVAP software.

6. PREDICTION OF DYNAMIC TORSIONALS

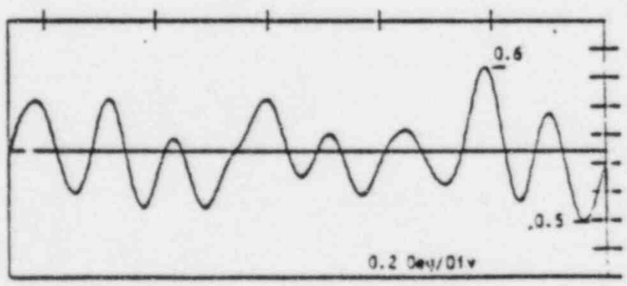
TORVAP C is capable of: a) Prediction of the dynamic torsional stress vector of each mass and shaft section including amplitude and phase angle, b) Summation of the torsionals of all modes.

The good correlation of TORVAP C dynamic results on free end amplitude and the dynamic torsiograph test record, taken by SWEC on January 8, 1984. demonstrated as follows. Twelve (12) order sum simulates even the d; details very well.

TORVAP C Simulation On Free End Amplitude vs. Shaft Angle



Torsiograph Recording[4] Curve B-33



[4]"Evaluation of Emergency Diesel Generator Crankshafts at Shoreham and Grand Gulf Nuclear Power Stations," Failure Analysis Associates, Palo Alto, California. March 30, 1984.

METHODS OF CALCULATION ON NOMINAL TORSIONAL STRESS

1. The analysis method is based on the vibration theory and experience[5]. TORVAP[6], a Torsional Vibration Analysis Program, was used to check the following:

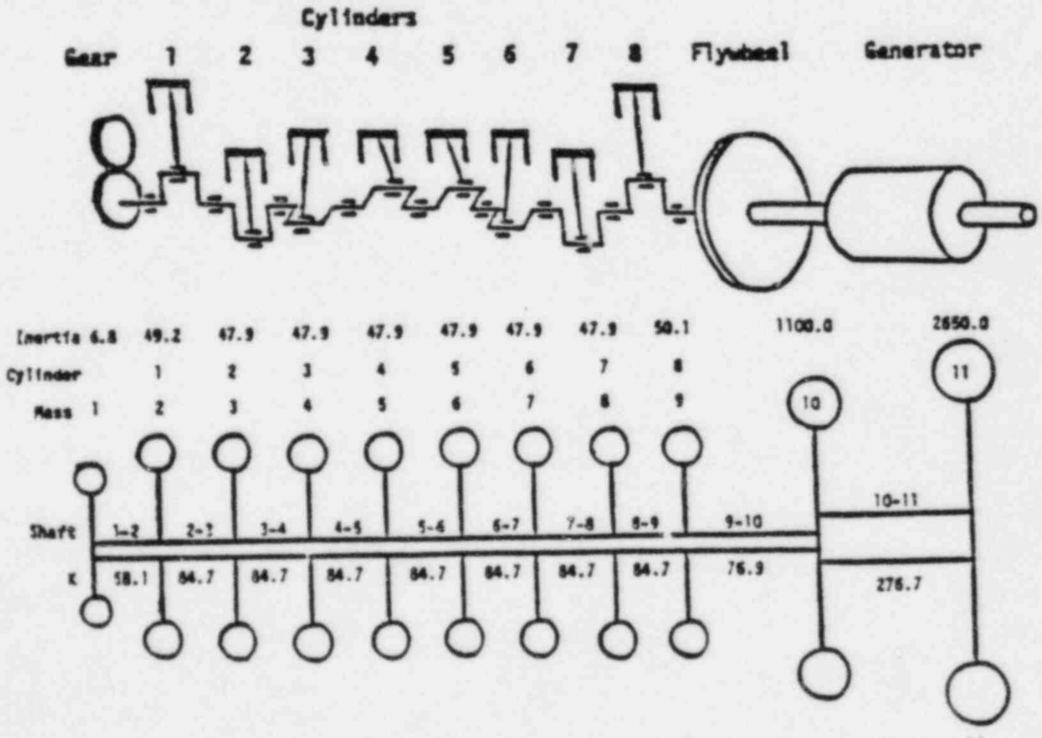
   a. The natural frequency and torsional stress of the engine generator system for the first three modes, per TORVAP R program.

   b. The amplitude sum of six (6) significant orders at the free end of the crankshaft, per TORVAP C program.

   c. The shaft nominal torsional stress for all modes of six (6) significant orders, per TORVAP C program.

   Only results on those crank sections close to the drive end are reported here.

2. The shaft numbering system and mass-elasticity simulation data used are shown in the following.



5. "Vibration Analysis for a Sound Generator-Set Design," Dr. Simon K. Chen, September 19, 1978.

6. "Torsional Vibration Analysis Program (TORVAP)," Computer Aided Design Centre, Madingley Road, Cambridge, UK CB3 OHB. June, 1975.

Power and Energy International Inc. P.O. 1064 555 Lawton Ave. Beloit, WI 53511 608/362-7071

RESULTS AND DISCUSSION

1. NATURAL FREQUENCY AND SYSTEM SIMULATION

The natural frequency of the system is 2323.3 cpm for the first mode, 5575.2 cpm for the second mode, and 7000.4 cpm for the third mode. The results are almost the same as TDI and FaAA's calculations. PEI used the same inputs to simulate the mass-elasticity shaft system of the subject 13 x 12 shaft. The comparison is as follows.

Mode	Natural Frequency, Cpm		
	TDI	PEI	FaAA
First	2323.3	2323.3	2323.8
Second	5575.8	5575.2	5576.4
Third	7000.2	7000.4	7002.0

The resonant rpm of each order for each mode equals the natural frequency, divided by the order number. Only those orders with resonant frequenc[illegible] around or lower than rated speed are investigated. For the first mo[illegible] they are 4.0 order resonance at 581 rpm and 5.5 order resonance at 422 [illegible]

The replacement (13 x 12) shaft has a 9% higher first mode natural frequency than that of the old 13 x 11 shaft, as shown in the following:

Shaft	PEI Calculated	TDI[6] Test
13 x 12	2323.3 cpm	---
13 x 11	2129.8 cpm	2146.0 cpm

[6] "Torsiograph and Shaft Amplitude Tests," Stone and Webster Engineering Corporation for Long Island Lighting Company (LILCO).

2. SINGLE ORDER NOMINAL TORSIONAL STRESS

The major order (4.0) causes torsional stress of 3146 to 3455 psi range, at shaft section, between cylinder 8 and flywheel, at the 225 bmep. The single order stress is increased 15% when the engine is running at 105% rated RPM.

4.0 Order Stress at 225 Bmep

BMEP psi	RPM Speed	TORVAP C Stress	TORVAP R Stress
225	472.5	4010 psi	3698 psi
225	450.0	3455 psi	3146 psi
225	427.5	3071 psi	----

The figures compare favorably with DEMA's allowable torsional st[illegible] of 5000 psi for a single order at rated as well as off speed c[illegible]

Engine manufacturers customarily calculate the single order torsional stress caused by the first mode resonance. This is the TORVAP R program. This first mode, 4.0 order torsional stress is 10% smaller than that calculated per TORVAP C for all modes also shown above.

Another significant order, the 5.5 order, increased dramatically from 1390 psi at rated to 3294 psi when the engine speed decreased to 427.5 rpm (95% of rated). The resonance 5.5 order torsional stress is 4793 psi at 422 rpm (93.7% of rated speed). This resonance torsional stress is still below the DEMA single order limit of 5000 psi.

5.5 Order Stress at 225 Bmep

BMEP	RPM	TORVAP C Stress,
225	472.5	898 psi
225	450.0	1390 psi
225	427.5	3294 psi
225	422.0	4793 psi

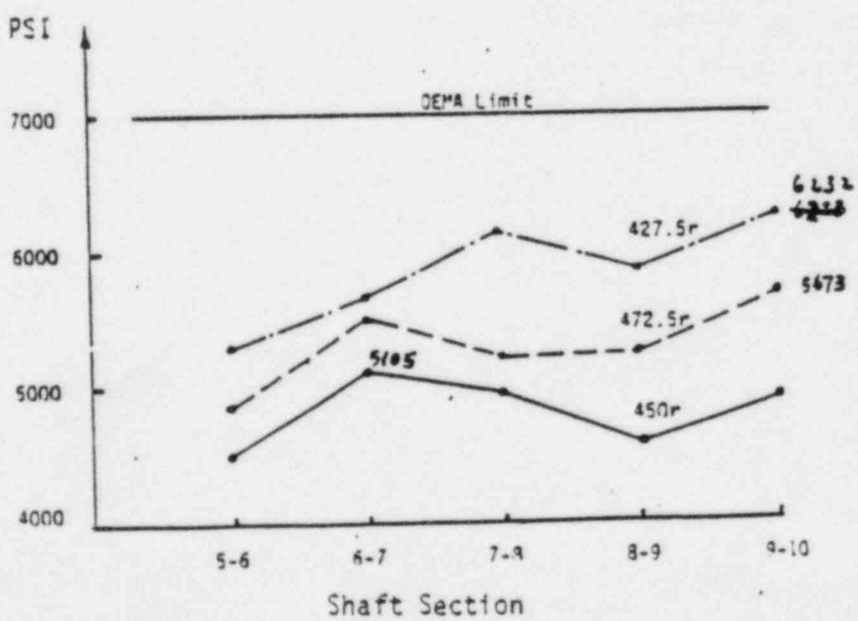
3. SUM OF ORDERS NOMINAL TORSIONAL STRESS

Two methods are used to calculate the sum of order torsional stress. The TORVAP C method which provides a true sum of orders for all modes. The other method is the "square root of sum of square" or SRSS method in conjunction with the TORVAP C calculation. TDI uses the SRSS method. These torsional stress values are compared in the following table.

	Shaft 6-7		Shaft 9-10	
Rating	True Sum	SRSS	True Sum	SRSS
225 bmep, 450 r	5101 psi	3375 psi	4900 psi	3763 psi

These figures compare favorably with the DEMA limit of 7000 psi for the sum of order torsional stress. For off-speed (95-105% rpm) fixed rack (constant 225 bmep) situation, the sum of order (true sum) torsional stress is calculated for those shaft sections from Section 5-6 up to the drive end (Section 9-10). The comparison can be shown in the following graph. It seems that the highest stress occurs at 427.5 rpm, 225 bmep condition.

Torsional Stress Along the Crank Sections at 225 bmep



4. COMPARISON OF TORSIONAL RESULTS

TORVAP C is furnished to us by Comshare and is only capable of calculating the sum of any selected (6) orders. PEI designed a subroutine to combine (2) sets of TORVAP C outputs to get the (12) order sum figures. This is accomplished to further improve the dynamic simulation. As the results show, the true sum of twelve (12) order provides a marginally higher nominal torsional stress level, and it does improve some of the dynamic details when the torsionals are plotted versus the shaft phase angle.

Comparison of Torsional Results

Item \ Shaft No.	Stress in psi					Amplitude
	5-6	6-7	7-8	8-9	9-10	Free E
TORVAP S_6 Sum of (6)	4498	5101	4941	4568	4900	0.56 deg.
TORVAP S_{12} Sum of (12)	4982	6020	5493	5002	5238	0.59 deg.
SRSS of S_6	3187	3375	3512	3664	3763	0.39 deg.

Torsional stress S_6 is the sum of orders for 0.5, 1.5, 2.5, 4.0, 4.5, and 5.5. S_6 is the highest at shaft section 6-7 at rated condition.

Torsional stress S_{12} is the sum of orders for the above, plus 3.5, 5.0, 6.5, and 8.0. S_{12} is the highest at shaft section 6-.

At rated condition, the twelve (12) order sum gives 0.59 degree, 0.03 degree larger than that of (6) order. SRSS figures, on the other hand, are inherently lower by a substantial amount. They are included here to check with the historical values used by the industry and that furnished by TDI.

PEI believes that the TORVAP C, used together with the PEI subroutine, is the state of art torsional simulation program.

5. COMPARISON OF FREE END AMPLITUDE

There are now several sets of test and calculated data on free end amplitude taken on both the 13 x 11 and 13 x 12 replacement crankshafts at rated conditions. TDI's torsiograph data shows three to five amplitudes of selected orders and the corresponding SRSS value. SWEC test report shows a full array of the free end amplitudes, and its corresponding true sum results (0.69 degree). The experimental spread was 0.55 to 0.69 degree when several recordings were studied.

Original	TDI Test[8]			SWEC Test	FaAA Calc	TORVAP C Calc.	TDI Test		SWEC Test
Date	1/8/84[8]	*	**	1/8/84[8]	3/30/84[4]	5/19/84	9/28/83[4]	12/12/75	9/19/83[9]
Order \ Shaft	13 x 12						13 x 11		
0.5			0.10	0.06	0.07	0.07			
1.5		0.17	0.21	0.17	0.18	0.14		0.21	
2.5		0.12	0.12	0.13	0.14	0.11		0.15	[illegible]
3.5				0.06	0.06	0.05			0.07
4.0		0.36	0.35	0.33	0.34	0.31		0.43	0.46
4.5				0.06	0.07	0.07		0.14	0.12
5.0			0.12	0.03	0.03	0.03			0.04
5.5				0.13	0.12	0.12			0.04
6.0				0.01	0.01	0.01			0.07
6.5				0.01	0.01	0.01			[illegible]
8.0						0.02			
SRSS	0.43	0.45	0.42	0.42		0.40	0.50	0.50	
True Sum				0.55 - 0.69	0.66	0.59			

* Shop Test for Gulf States, SN-74038, overall per meter
** Shop Test for EDOK-ETER, SN-74038, overall per meter

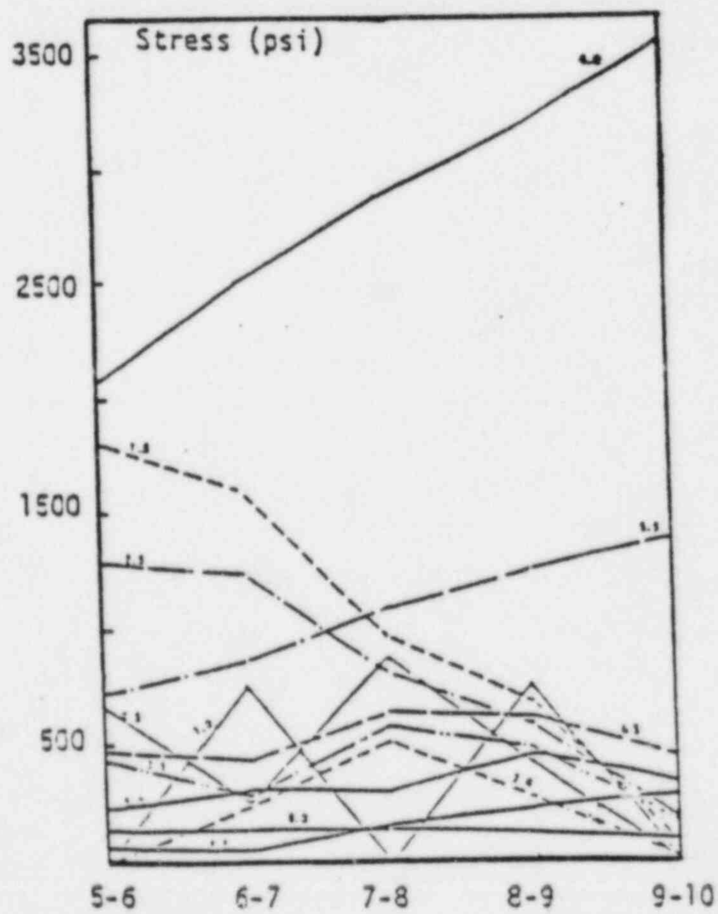
For the 13 x 11 shaft, the SWEC as well as TDI test data shows around 30 - 40% higher 4.0 order amplitude than that of 13 x 12 shaft. This indicates that the 13 x 12 shaft stress level is indeed significantly reduced. For the 13 x 12 shaft TORVAP C true sum amplitude checks well within the experimental spread of the SWEC test. In addition, TORVAP SRSS result checks within a few percent of the test data furnished by either TDI or SWEC.

4. "Evaluation of Emergency Diesel Generator Crankshafts at Shoreham and Grand Gulf Nuclear Power Stations," Failure Analysis Associates, Palo Alto, California. March 30, 1984.
8. "Field Test of Emergency Diesel Generator 103," Bercel and Hall, Stone and Webster Engineering Corporation. Figure 8-33. February, 1984.
9. "Emergency Diesel Generator Crankshaft Failure Investigation Shoreham Nuclear Power Station," Failure Analysis Associates, Palo Alto, California, October 31, 1983.

6. SINGLE ORDER TORSIONAL STRESS ALONG THE CRANK SYSTEM

TORVAP C is capable of calculating torsional stress for every shaft section, and for each significant single order input. For the major order (4.0) of an eight cylinder inline engine, the torsional stress increases as it approaches the first mode nodal point, which is located close to the flywheel end. For the subject shaft, both the 4.0 and 5.5 order torsionals are highest for shaft section 9-10. The 1.5 and 2.5 order torsionals are higher at shaft section 5-6, and decrease drastically along the shaft section toward the flywheel end. Some integral orders, such as 1.0 and 2.0, on the other hand, provide additional "nodal" points along the shaft, as shown below. This figure substantiates the importance of using the modal superposition principle to predict shaft stress for all shaft sections. TORVAP R only deals with orders having a first mode resonant frequency close to the rated engine speed considered.

Stress Distribution For Individual Orders At Rated Conditions



7. COMPARISON OF GAS PRESSURE TANGENTIAL EFFORT Tn

The TORVAP program has a built-in gas pressure tangential effort (Tn) data base software for each order, based on Lloyd's table. For this analysis, only horsepower, displacement, pm and mechanical efficiency at rating has to be input. Several sets of Tn magnitudes used by TDI and FaAA are compared with Lloyd figures as follows:

Source / Order	TDI 1973-74[9]	Lloyd 1976[2]	TDI 1983[6]	FaAA 1984[10]
0.5	11.0	88.5	155.9	
1.0	-	95.4	106.9	
1.5	19.0	90.1	129.5	112.3
2.0	-	74.5	122.7	
2.5	20.2	62.0	71.7	77.0
3.0	-	51.2	51.4	
3.5	16.7	41.5	42.8	48.0
4.0	13.3	32.7	27.7	35.6
4.5	9.9	25.3	23.8	26.2
5.0	7.3	19.2	17.4	
5.5	5.7	14.7	12.8	15.5
6.0	4.2	11.3	5.7	
6.5	3.3	9.0	4.5	
7.0	2.7	7.3	3.7	
7.5	2.2	5.9	3.1	
8.0	1.9	4.7	2.5	

10. "Evolution Of Emergency Diesel Generator Crankshafts At Shoreham And Grand Gulf Nuclear Power Stations." FaAA, April 19, 1984.

This comparison shows the following:

- The set of Tn figures used by TDI in 1973-74 correspond to a very low imep (20 psi) level, as compared with those listed in Lloyd.
- Lloyd Tn figure for the 4.0 order is 10% higher than that used by TDI in 1984, and is 8% lower than used by FaAA.
- Lloyd Tn figures for lower orders, especially those 0.5, 1.0, 1.5 and 2.5 orders are lower than those figures used by either TDI or FaAA. These lower Tn inputs could explain why TORVAP's sum of orders torsional results are somewhat lower than those of FaAA.
- Lloyd Tn figures for the 6.0, 6.5 and 8.0 orders are substantially higher than those used by TDI. These higher Tn inputs explain why some of PEI's resonant critical stresses below rated rpm are significantly higher than those calculated by TDI.

Lloyd Tn Table is based on many years of industrial experience on commercial engines running at a range of imep levels. The table is based on a representative commercial engine, and therefore, should be most acceptable for the purpose of predicting torsional stress levels for satisfying codes and design standards. The dynamic torsional amplitude predicted by using this set of representative Tn figures might not exactly reproduce the dynamic test data of a particular engine in question. To reproduce the test data more exactly, a carefully measured indicator diagram is required.

Lloyd's Table does not take into account the difference of engine design and operating parameters, such as:

- degrees of intercooling
- operating fuel-air ratio and turbo match
- injection timing
- machanical frictions, etc.

8. MAGNIFIER AND DAMPING FACTORS

The magnifier assumption is built in the TORVAP software. It is a function of the equilibrium amplitude θ_0. The magnifier factor is a reverse function to the damping factor. Most engine manufacturer's prefer to use their own test-proven damping relationship, in order to achieve a close check between the calculation and test results. Without this design and development background, PEI uses the TORVAP magnifier assumption as is.

The comparison of TORVAP built-in magnifier factor and that used by TDI is shown in the following table.

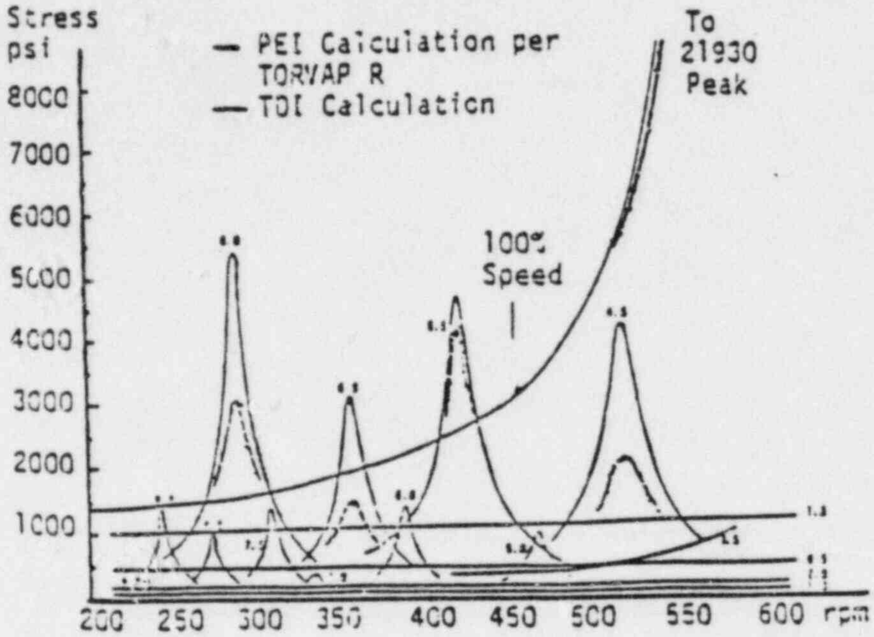
COMPARISON OF MAGNIFIER FACTOR USED

Order	TDI	TORVAP	%
4.0	16.6	17.3	8%
4.5	15.2	29.8	96%
5.5	28.4	28.8	2%
6.5	28.3	32.0	14%
8.0	28.5	27.0	-6%

The comparison is reasonable for 4.0, 5.0, and 8.0 orders. The comparison is not good for the 4.5 order. This magnitude of difference will not alter the rated rpm torsional stress level, but it could alter substantially the offspeed torsional characteristics at those resonance speed ranges.

To ascertain more realistic damping characteristics of a specific engine, one must first run a variable speed run,perhaps at low bmep, and measure torsional amplitudes at free end throughout the speed range studied. Then, one would use a series of rational damping functions with damping coefficients varied until the simulation checks with the test curve. After this test procedure, the damping coefficient can then be used in future simulation of the same engine design. TORVAP's magnifier factors range from 17.3 to 32.0, for those significant orders considered. These figures are quite in line with an average factor of 25, oftenly used as a quick guess.

The effect of using (2) different sets of magnifier assumptions can be further demonstrated by (2) sets of calculated torsional amplitudes plotted versus rpm. The solid curve is per TORVAP R software. The dotted curve is taken from a TDI calculation using their own magnifier assumption.



The 4.0 order response is reasonably close despite the 15% difference in Tn and 8% difference in magnifier factor. This is the major order.

The large 4.5 order difference can be entirely attributed to the fact that TDI's magnifier factor is half that of TORVAP.

The 5.5 order difference can be completely attributed to the 14% difference in Tn

The large 6.5 order difference can be entirely attributed to the Tn figures. LLOYD's Tn is twice as much as that used by TDI.

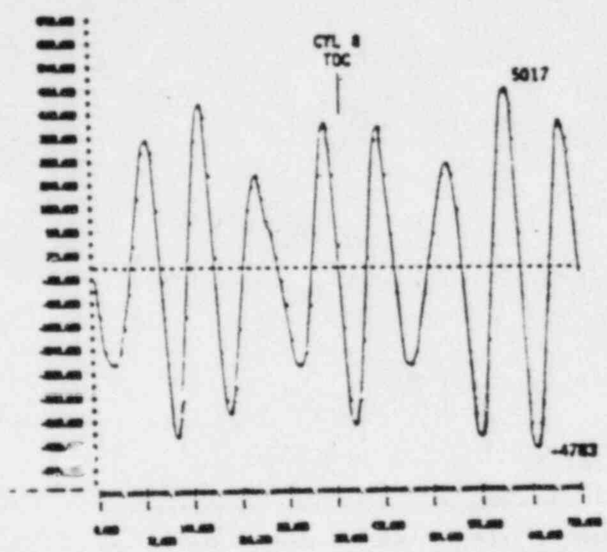
The large 8.0 order difference can again be attributed to the fact that LLOYD's Tn is 90% higher than that used by TDI.

This discussion and analysis show the importance of having accurate Tn figures and magnifier factors when the entire dynamic response of the shaft is to be simulated, which is required for a variable speed application. In the case of SNPS application, only, 450 rpm and its ± 5% variation are needed for torsional scrutiny.

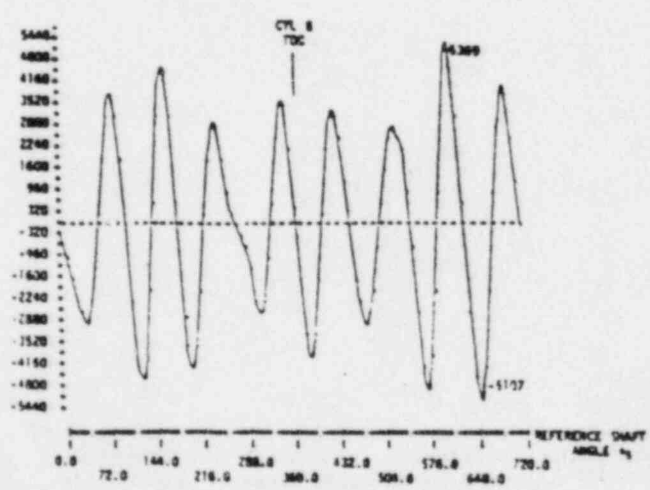
9. EFFECT OF NUMBER OF ORDERS USED IN THE CALCULATION

Two (2) sets of torsional stress at rated condition were obtained for comparison. The (12) order sum dynamic results provided some 6-7% higher peak-to-peak amplitudes quite consistently when several traces were studied. The details of dynamic amplitude at a specific phase angle, however, could vary up or down as shown below. PEI believes that (6) order true sum is quite adequate for most use. Twelve (12) order is only worthwhile when more exacting simulation of the dynamic response is desired.

(6) Order Sum Torsional Stress, Shaft 9-10, At Rated Condition



(12) Order Sum Torsional Stress, Shaft 9-10, At Rated Condition



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APPENDIX I

TORVAP Program and Modification

MODIFICATION OF TORVAP C PROGRAM

The Torsional Vibration Analysis Program (TORVAP) is used throughout this report. The TORVAP software is available from Comshare Inc., Ann Arbor, Michigan.

The TORVAP program is made up of two parts, TORVAP R and TORVAP C. TORVAP R is used to give a general conception of the frequency response characteristics of the entire system, to search the natural frequency and to determine the significant order and the stress caused by it. Most engine build- in the 1973-75 period use TORVAP R type of torsional analysis. As before, TORVAP R program is virtually the computerized version of the Holzer table - Dan Hartog - Ker Wilson classical torsional calculations.

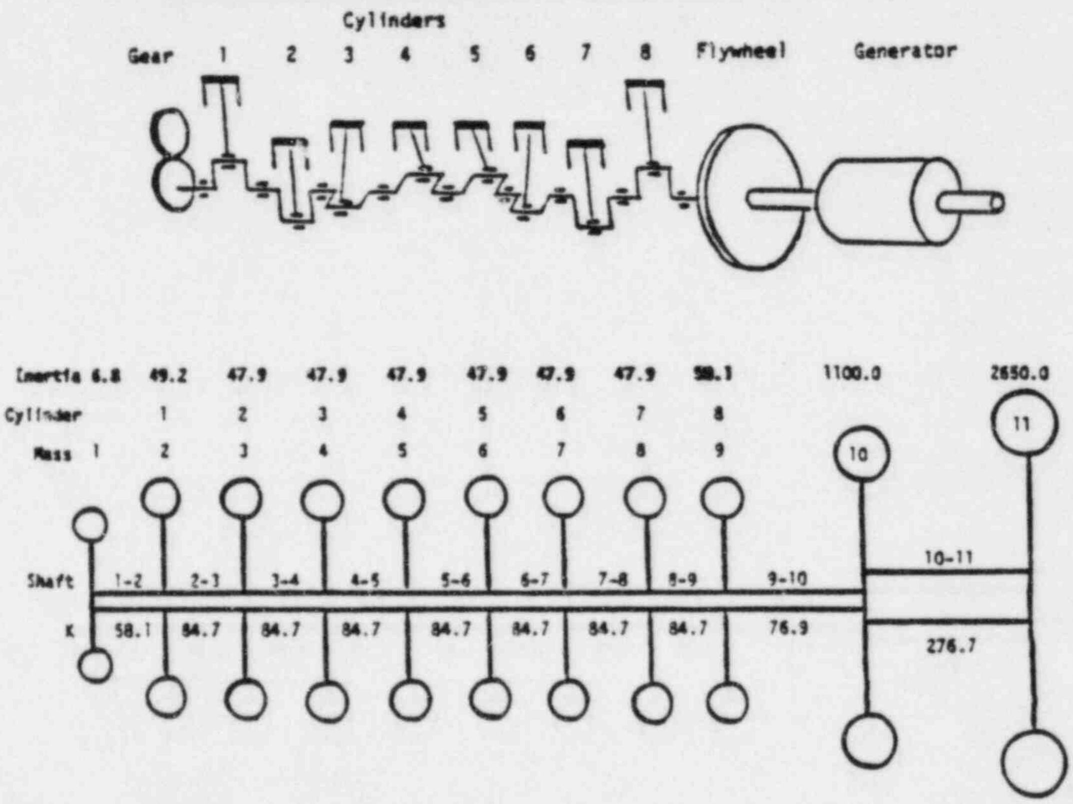
TORVAP C is used to solve torsional differential equations of the entire system for specified excitation torque, and to determine the torsional amplitudes and stresses at any section of the shaft. TORVAP C has optional built-in inputs and assumptions, including a Tn table at different imep, based on Lloyd reference. TORVAP C calculates the sum of order results up to six (6) orders, for all modes. In order to enhance prediction of the dynamic behavior, PEC Consultants designed a sub-routine to add vectorally the results of two (2) separate six (6) order programs. The dynamic free amplitude and torsional stress graphs are plotted, based on the sum of twelve (12) orders. Twelve (12) order results seem to provide slightly improved dynamic details than those based on six (6) order programs.

Nominal Torsional Stress Calculations

1. SYSTEM SIMULATION

The simulation of the mass-elasticity system is the same as that used by FaAA and TDI. PEI Consultants agree with the mass-elasticity figures, after examining several calculations and their close checks with the test results on natural frequency of the shaft system. The shaft system is represented below.

Shaft Numbering System and Mass-Elasticity Data



Inertia (I) is the rotating mass inertia of the shaft in lb·ft·sec^2. The values are furnished by TDI/FaAA

Shaft stiffness data (K) in 10^6.ft.lb/rad are furnished by TDI/FaAA.

PEI spot-checked some of the I & K figures. The TDI figures seemed reasonable.

2. NATURAL FREQUENCY

Using the Holzer method to search the natural frequency of the system at a speed range of 1000 to 8000 cpm, the natural frequency is obtained as follows:

First Mode	Second Mode	Third Mode
2323.3 cpm	5575.2 cpm	7000.4 cpm

The first mode natural frequency, 2323.3 cpm, is 5.2 times that of the rated speed. The major order, 4.0, has it's resonance at 581 rpm. The 13 x 12 shaft is affected by the 4.0, 4.5, 5.5, and 8.0 orders. The second and third mode natural frequencies are 12.4 and 15.6 t[illegible] that of the rated speed, respectively. These modes a[illegible] significant.

The natural frequencies of this replacement crankshaft are higher than the old 13 x 11 shaft, due to higher stiffness (K). The comparison is as follows.

	13 x 12 Shaft	13 x 11 Shaft
First mode	2323.3 cpm	2129.8 cpm
Second mode	5575.2 cpm	5455.4 cpm
Third mode	7000.4 cpm	6495.2 cpm
Shaft Stiffness (K)	$84.7 \cdot 10^6 \cdot$ ft·lb/rad	$71.2 \cdot 10^6 \cdot$ ft·lb/rad
Crankpin Diameter	12 inch	11 inch

The shaft stiffness (K) of the replacement shaft is nearly 20% stronger than that of the old shaft, due to a 9% increase in the crankpin diameter, with the main diameter remaining at 13 inches.

This higher K value contributes to two important improvements [illegible] replacement shaft:

- Increase the modulus or the strength of the shaft.
- Increase the natural frequency of the shaft by 9 % and thereby reduce the equilibrium amplitude by as much as 31%.

3. FIRST MODE HOLZER TABLE

m	Shaft	I	θ	Tm	ΣTm	K	Δθ
	Free				0.00		
1	1-2	6.8	1.00	0.01	0.01	1.01	0.01
2	2-3	49.2	0.99	0.05	0.06	1.48	0.04
3	3-4	47.9	0.95	0.05	0.11	1.48	0.07
4	4-5	47.9	0.88	0.04	0.15	1.48	0.10
5	5-6	47.9	0.78	0.04	0.19	1.48	0.12
6	6-7	47.9	0.66	0.03	0.22	1.48	0.15
7	7-8	47.9	0.51	0.03	0.25	1.48	0.17
8	8-9	47.9	0.34	0.02	0.26	1.48	0.18
9	9-10	50.1	0.16	0.01	0.27	1.34	0.20
10	10-11	1100.0	-0.04	-0.04	0.23	4.83	0.05
11	Free	2650.0	-0.09	-0.23	0.00		

m = mass number

Shaft = shaft number

I = mass moment of inertia, $lb \cdot ft \cdot sec^2$

K = shaft stiffness, $10^6 \cdot ft \cdot lb/deg$ = K in rad divided by 57.3

θ = relative vibration amplitude, assume 1 deg at mass 1

$Tm = I \cdot \omega^2 \cdot \theta$ = inertia torque, $10^6 \cdot ft \cdot lb$, calculated for $\theta_1 = 1$ degree at m 1

ΣTm = summation of inertia torque, $10^6 \cdot ft \cdot lb$

$\Delta\theta = \frac{1}{K} \Sigma Tm$ = relative angular vibration amplitude between masses, in degree

Results

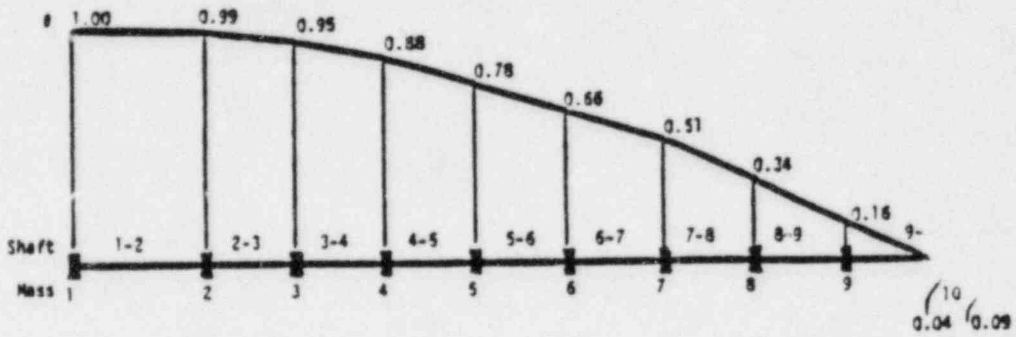
max $\Delta\theta$ = 0.20 degree at shaft 9-10, when θ_1 = 1 degree

max ΣTm = 270,000 ft·lb at shaft 9-10, when θ_1 = 1 degree

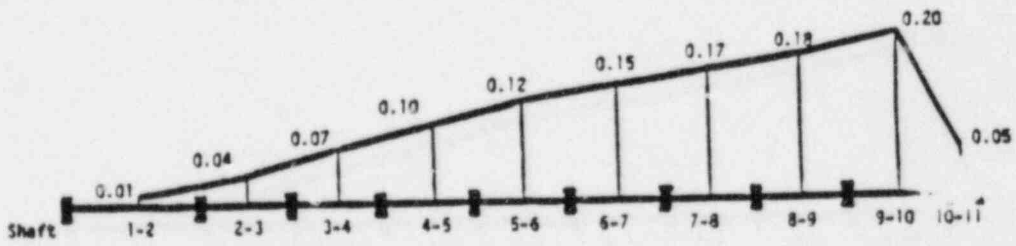
4. **RELATIVE AMPLITUDE FROM FIRST MODE HOLZER TABLE**

The relative torsional amplitudes θ of each mass at first mode resonance frequency and the relative displacements $\Delta\theta$ between masses are shown as follows.

Relative Amplitude θ at Resonance, First Mode



Relative Displacement Between Masses $\Delta\theta$ at Resonance, First Mode



The relative displacement ($\Delta\theta$) curve shows that the maximum value of 0.20 is situated at shaft 9-10 between cylinder 8 (mass no. 9) and the flywheel. This maximum $\Delta\theta$ location should correspond to the maximum relative stress caused by the major order of the first mode torsional vibration.

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5. RELATIVE STRESS FROM FIRST MODE HOLZER TABLE

The first mode natural frequency as previously stated is 2323.3 cpm. The sum of relative inertia torque, Tm, and relative torsional stress, S, are tabulated below for the resonance condition.

m	Shaft	β rad or deg	ΣTm $\frac{nm}{rad}\cdot 10^6$	ΣTm $\frac{ft\cdot lb}{deg}$	S $\frac{n/mm^2}{rad}$	S $\frac{psi}{deg}$
1	1-2	1.00	0.55	700	100	250
2	2-3	0.99	4.47	57500	800	2030
3	3-4	0.95	8.13	104700	1460	3700
4	4-5	0.88	11.53	148400	2070	5250
5	5-6	0.78	14.54	187100	2620	6620
6	6-7	0.66	17.06	219600	3070	7770
7	7-8	0.51	19.01	244800	3420	8660
8	8-9	0.34	20.33	261700	3660	9260
9	9-10	0.16	20.99	270200	3780	9560
10	10-11	-0.04	17.79	229100	1350	3420
11		-0.09				

$\Sigma Tm = \Sigma(I\cdot\omega^2\cdot\beta)$ in nm when $\beta_1 = 1$ rad, and in ft·lb when $\beta_1 = 1$ deg

S = relative torsional stress = $\Sigma Tm/Z$

Z = crankpin section modulus = $\pi d^3/16$, where d = 12", except when d = 16" for shaft 10-11

Results

$\Sigma I\beta^2$ = effective inertia of system = 225.6 ft·lb·sec^2 or 305.8 kg·m·sec^2

G = equilibrium amplitude factor = θ_0/Tn = 0.00085 deg/psi (1 deg per 1174 psi)

Maximum relative torsional stress at shaft 8-9 and 9-10 with relative stresses of 9260 and 9560 psi/degree, respectively, for the first mode.

6. <u>NOMINAL TORSIONAL STRESS AT RESONANCE AND SERVICE SPEED FOR FIRST MODE, PER TORVAP R PROGRAM</u>

The previous table shows that the maximum relative torsional stress occurs at shaft 9-10. These stress levels at the resonance speed (NC) and rated load speed (450 rpm) are calculated for all orders, 0.5 to 10.0, for the first mode, and are shown below.

ORDER	NC	NC/NS	$\Sigma\delta$	Tn	θ_0	AT NC SPEED		AT 450 RPM	
						Mr	Sr	Mf	Sf
.5	4647.	10.38	0.70	88.5	0.05	21.8	11010.	1.01	510.
1.0	2323.	5.16	0.15	95.4	0.05	21.7	11200.	1.04	133.
1.5	1549.	3.44	1.39	90.1	0.11	18.3	18690.	1.09	1117.
2.0	1162.	2.58	0.38	74.5	0.03	15.5	30760.	1.18	170.
2.5	929.	2.07	1.39	62.0	0.07	20.1	14140.	1.31	920.
3.0	774.	1.72	0.15	51.2	0.01	34.3	2830.	1.51	[illegible]
3.5	664.	1.48	0.70	41.5	0.03	26.4	6240.	1.[illegible]	[illegible]
4.0	581.	1.29	5.28	32.7	0.13	17.3	21930.	2.[illegible]	[illegible]
4.5	516.	1.15	0.70	25.3	0.02	29.8	4310.	4.13	[illegible]
5.0	465.	1.03	0.15	19.2	0.00	47.3	1080.	15.30	350.
5.5	422.	.94	1.39	14.7	0.02	28.8	4790.	7.15	1191.
6.0	387.	.86	0.38	11.3	0.00	42.7	1470.	2.84	98.
6.5	357.	.79	1.39	9.0	0.01	32.5	3330.	1.71	175.
7.0	332.	.74	0.15	7.3	0.00	50.0	430.	1.19	10.
7.5	310.	.69	0.70	5.9	0.00	42.9	1450.	0.90	30.
8.0	290.	.65	5.28	4.7	0.02	27.4	5540.	0.71	144.
8.5	273.	.61	0.70	3.9	0.00	47.5	1064.	0.58	13.
9.0	258.	.57	0.15	3.4	0.00	50.0	200.	0.49	2[illegible]
9.5	245.	.54	1.39	3.1	0.00	42.6	1480.	0.[illegible]	[illegible]
10.0	232.	.52	0.38	2.7	0.00	50.0	420.	0.36	[illegible]

Where:

NC = critical speed of an order = 2323/order

NS = rated speed = 450 rpm

$\Sigma\delta$ = relative amplitude vector sum of all cylinders, in degree

Tn = tangential effort of cylinder gas pressure from Lloyd, extrapolated to 250 imep, based on 225 bmep and mech. eff. = 0.9

θ_0 = equilibrium amplitude of shaft 9-10 without magnification

= $\theta_f \cdot Tn \cdot \Sigma\delta$, deg

θ_f = equilibrium amplitude factor from Holzer Table and engine data, in degree

Mr = $3.8 \cdot \theta_0^{-1/2}$ magnifier of resonance, only engine damping is considered

Mf = flank magnifier at service speed = $\left(\left[1 - \left(\frac{NS}{NC}\right)^2\right]^2 + \left(\frac{NS}{NC}\right)^2 \cdot \frac{1}{Mr^2}\right)^{-1/2}$

Sr = $\theta_0 \cdot Mr \cdot S$, torsional stress at resonant speed, psi

Sf = $\theta_0 \cdot Mf \cdot S$, torsional stress at service speed, psi

7. NOMINAL TORSIONAL STRESS AT OVERSPEED CONDITION

The previous calculations were made at rated. An overspeed condition of 225 bmep and a 105% rated speed of 472.5 rpm was evaluated to consider the potential, but rare, over-shoot condition. Shaft section 9-10 is evaluated as follows:

ORDER	NC	NC/NS	Σβ	Tn	θ_0	AT NC SPEED		AT 472.5 RPM	
						Mr	Sr	Mf	Sf
0.5	4647.	9.83	0.70	88.5	0.05	22.0	10660.	1.01	510.
1.0	2323.	4.92	0.15	95.4	0.05	21.7	11140.	1.04	135.
1.5	1549.	3.28	1.39	90.1	0.10	18.4	18190.	1.10	1127.
2.0	1162.	2.46	0.38	74.5	0.21	15.5	30850.	1.20	159.
2.5	929.	1.97	1.39	62.0	0.07	20.2	13780.	1.35	94[illegible]
3.0	774.	1.64	0.15	51.2	0.01	34.1	2880.	1.59	[illegible]
3.5	664.	1.41	0.70	41.5	0.02	26.6	6090.	2.02	[illegible]
4.0	581.	1.23	5.28	32.7	0.13	17.5	21360.	2.93	3698.
4.5	516.	1.09	0.70	25.3	0.02	30.1	4210.	6.05	874.
5.0	465.	0.98	0.15	19.2	0.00	47.6	1060.	24.88	570.
5.5	422.	0.89	1.39	14.7	0.02	29.0	4690.	3.93	655.
6.0	387.	0.82	0.38	11.3	0.00	42.9	1440.	2.04	71.
6.5	357.	0.76	1.39	9.0	0.01	32.7	3260.	1.34	[illegible]
7.0	332.	0.70	0.15	7.3	0.00	50.0	420.	0.97	8.
7.5	310.	0.66	0.70	5.9	0.00	43.2	1410.	0.75	26.
8.0	290.	0.62	5.28	4.7	0.02	27.6	5420.	0.61	122.
8.5	273.	0.58	0.70	3.9	0.00	47.9	1040.	0.50	11.
9.0	258.	0.55	0.15	3.4	0.00	50.0	200.	0.43	2.
9.5	245.	0.52	1.39	3.1	0.00	42.9	1440.	0.37	12.
10.0	232.	0.49	0.38	2.7	0.00	50.0	400.	0.32	3.

Comparing these data with those for the rated condition, the fourth order torsional stress is 17% higher than that at rated condition because NS of 472.5 rpm is closer to the resonance speed NC.

The second and third mode natural frequencies cause negligible shaft stress on shaft 9-10. The value of the 4.0 order is around 11 psi for the second mode, and 174 psi for the third mode at rated condition.

8. TORSIONAL STRESS DISTRIBUTION (4.0 ORDER) ALONG THE SHAFT SYSTEM

From the TORVAP C program, the nominal torsional stress at each shaft number can be calculated for all modes. The fourth order results are tabulated as follows.

m	Shaft	Mass No.	Mass Vibration θ, deg	ϕ_m, deg	Shaft Twist $\Delta\theta$, deg	ϕ_s, deg	ΣTm ft·lb	S psi
GEAR	1-2	1	0.31	28.42	-0.00	28.42	1310	46
CYL 1	2-3	2	0.31	28.42	-0.01	28.45	16671	590
CYL 2	3-4	3	0.30	28.42	-0.02	28.45	31446	1112
CYL 3	4-5	4	0.28	28.42	-0.03	28.44	45590	1612
CYL 4	5-6	5	0.25	28.42	-0.04	28.44	58818	2080
CYL 5	6-7	6	0.21	28.41	-0.05	28.43	70864	2506
CYL 6	7-8	7	0.16	28.40	-0.06	28.42	81487	2882
CYL 7	8-9	8	0.10	28.39	-0.06	28.40	90471	3200
CYL 8	9-10	9	0.04	28.38	-0.07	28.38	97696	3455
F.W.	10-11	10	-0.03	28.38	-0.02	28.38	76694	1144
GEN		11	-0.05	28.38				

The vibrations and twists in the table are represented by a cosine wave form and assume a positive rotation direction.

$$\theta\ (t) = \theta \cos [n\ (\omega t - \phi_m)]$$

$$\Delta\theta\ (t) = \Delta\theta \cos [n\ (\omega t - \phi_s)]$$

n = the order of harmonics

ω = the angular rotation speed of the engine

ϕ_m and ϕ_s = phase angle for mass and shaft, respectively, with respect to firing TDC of the first cylinder.

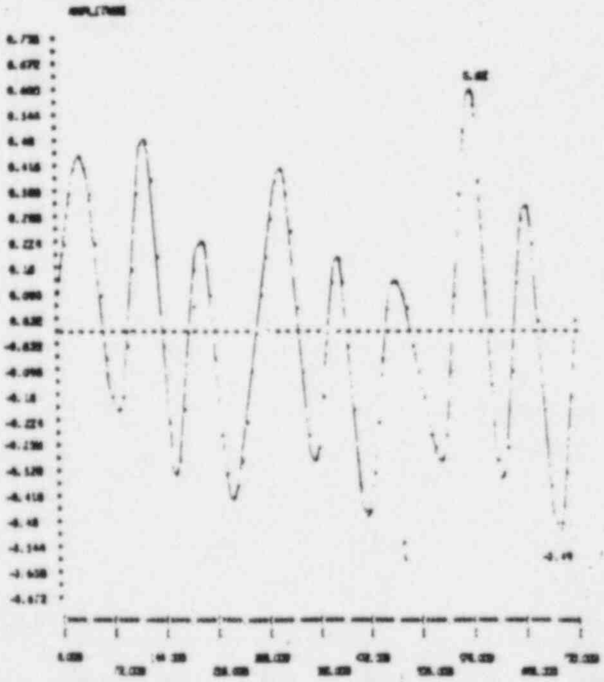
The maximum torsional stress caused by the 4.0 order Tn is 3455 psi for the sum of all modes. This is approximately 9% higher than that caused by the first mode alone, (per TORVAP R).

9. FREE END TORSIONAL AMPLITUDE AT RATED CONDITION

TORVAP C gives torsional amplitudes for each order and for the sum of orders. Six (6) significant orders were selected for calculation. The square root of the sum of square (SRSS) amplitude was also calculated to compare with TDI data, as shown in the following table.

Order	Amplitude θ, degree	Phase Angle ϕ_m, degree
0.5	0.07	37.4
1.5	0.14	83.5
2.5	0.11	22.0
4.0	0.31	28.4
4.5	0.07	21.1
5.5	0.12	60.9
SRSS	0.39	

The dynamic free end torsional amplitude (true sum) for the six (6) selected orders is shown as follows.



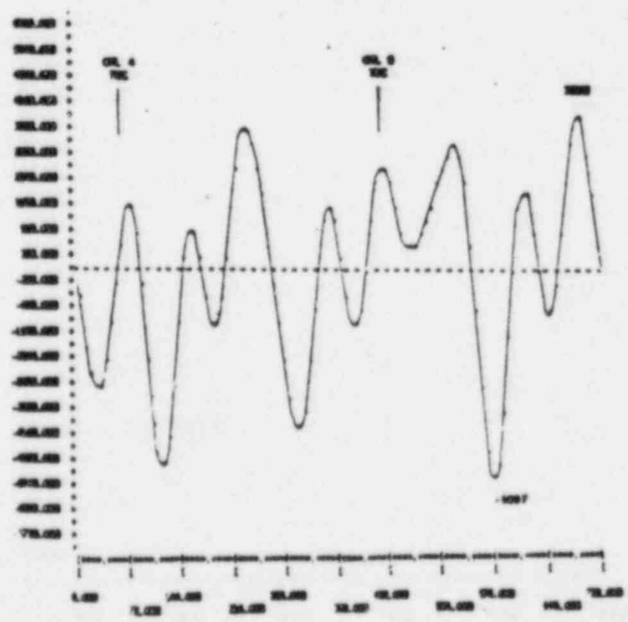
The mean amplitude is 0.56 degree, based on a maximum 0.62 degree and a minimum -0.49 degree.

10. NOMINAL TORSIONAL STRESS FOR SHAFT 5-6 AT RATED CONDITION

The shaft stress at rated condition is shown in the following table. Six (6) selected orders were used in this series of calculations. The SRSS torsional stress is also listed to compare with TDI data.

Order	Stress, psi	Phase Angle $\emptyset_s$, degree
0.5	674	484.6
1.5	1718	219.2
2.5	1296	85.3
4.0	2080	73.4
4.5	485	64.7
5.5	714	26.2
SRSS	3187	

The dynamic stress amplitude (true sum) for the six (6) selected orders is shown as follows.



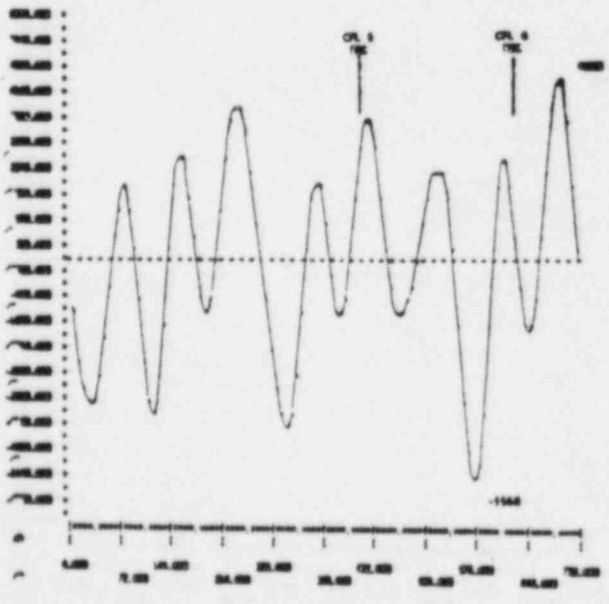
The mean stress is 4498 psi, based on a maximum 3898 psi and a minimum -5536 psi.

11. <u>NOMINAL TORSIONAL STRESS FOR SHAFT 6-7 AT RATED CONDITION</u>

The shaft stress at rated condition is shown in the following table. Six (6) selected orders were used in this series of calculations.

Order	Stress, psi	Phase Angle $\emptyset_s$, degree
0.5	261	350.7
1.5	1604	204.8
2.5	1240	93.2
4.0	2506	73.4
4.5	439	60.4
5.5	858	27.9
SRSS	3375	

The dynamic stress amplitude (true sum) for the six (6) selected orders is shown as follows.



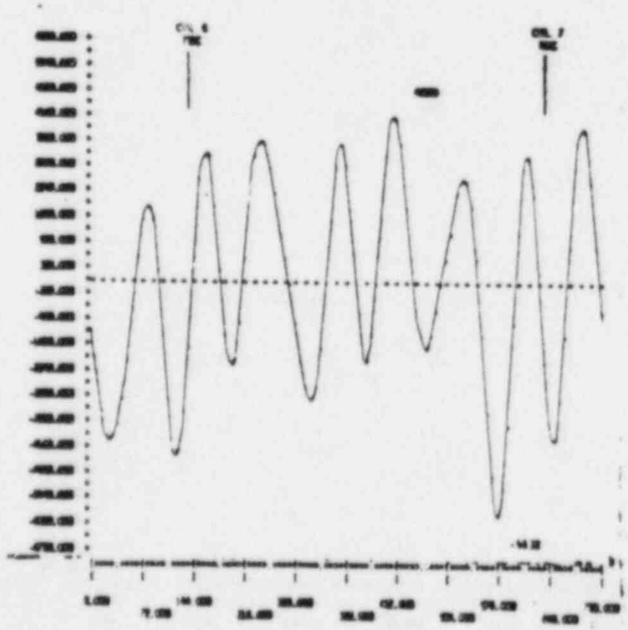
The mean stress is 5101 psi, based on a maximum 4634 psi and a minimum -5568 psi.

12. NOMINAL TORSIONAL STRESS FOR SHAFT 7-8 AT RATED CONDITION

The shaft stress at rated condition is shown in the following table. Six (6) selected orders were used in this series of calculations.

Order	Stress, psi	Phase Angle $\emptyset_s$, degree
0.5	882	350.2
1.5	978	204.8
2.5	814	93.4
4.0	2883	73.4
4.5	646	59.0
5.5	1098	28.2
SRSS	3512	

The dynamic stress amplitude (true sum) for the six (6) selected orders is shown as follows.



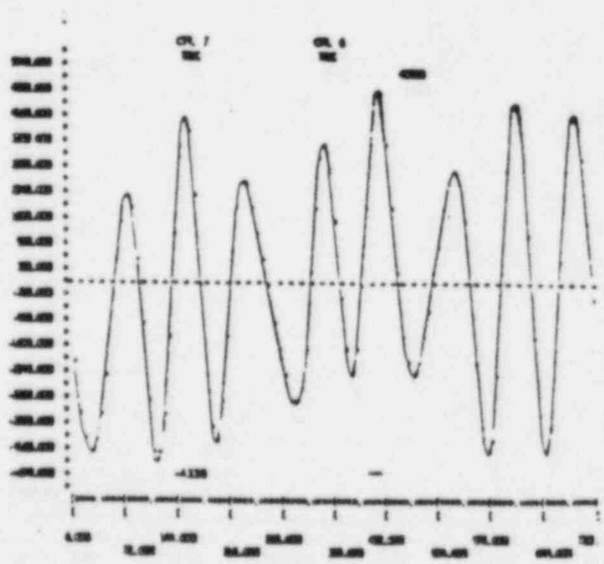
The mean stress is 4941 psi, based on a maximum 4250 psi and a minimum -5632 psi.

13. NOMINAL TORSIONAL STRESS FOR SHAFT 8-9 AT RATED CONDITION

The shaft stress at rated condition is s. wn in the following table. Six (6) selected orders were used in this series of calculations.

Order	Stress, psi	Phase Angle $\emptyset_s$, degree
0.5	625	439.7
1.5	695	178.1
2.5	600	106.0
4.0	3200	73.4
4.5	609	62.5
5.5	1258	28.9
SRSS	3664	

The dynamic stress amplitude (true sum) for the six (6) selected orders is shown as follows.



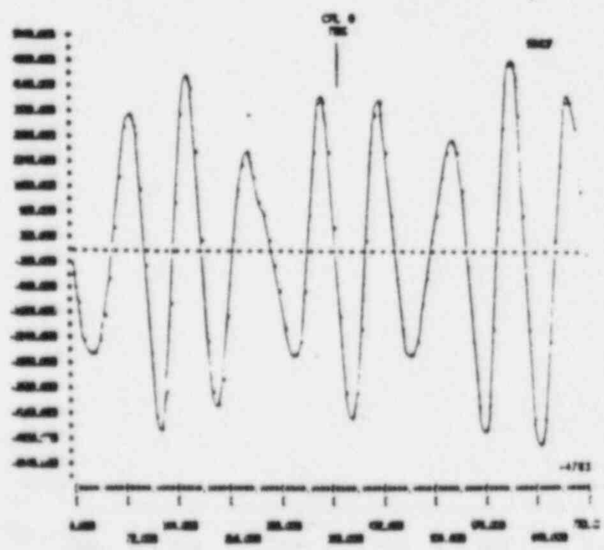
The mean stress is 4568 psi, based on a maximum 4798 psi and a minimum -4338 psi.

14. NOMINAL TORSIONAL STRESS FOR SHAFT 9-10 AT RATED CONDITION

The shaft stress at rated condition is shown in the following table. Six (6) selected orders were used in this series of calculations.

Order	Stress, psi	Phase Angle Ø-, degree
0.5	4	399.5
1.5	85	204.2
2.5	199	94.4
4.0	3455	73.4
4.5	497	61.4
5.5	1390	28.4
SRSS	3763	

The dynamic stress amplitude (true sum) for the six (6) selected orders is shown as follows.



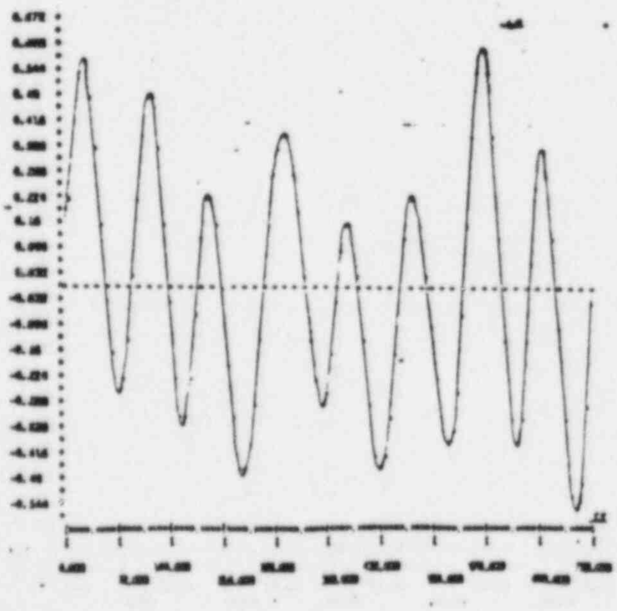
The mean stress is 4900 psi, based on a maximum 5017 psi and a minimum -4783 psi.

15. FREE END TORSIONAL AMPLITUDE AT 225 bmep, 472.5 rpm

TORVAP C gives torsional amplitudes for each order and the amplitude for the sum of orders. Six (6) significant orders were selected for calculation. The square root of the sum of square (SRSS) amplitude was also calculated to compare with TDI data, as shown in the following table.

Order	Amplitude θ, degree	Phase Angle ϕ_m, degree
0.5	0.07	37.4
1.5	0.14	83.5
2.5	0.12	21.4
4.0	0.37	28.6
4.5	0.10	21.5
5.5	0.06	64.5
SRSS	0.42	

The dynamic free end torsional amplitude (true sum) for the six (6) selected orders is shown as follows.



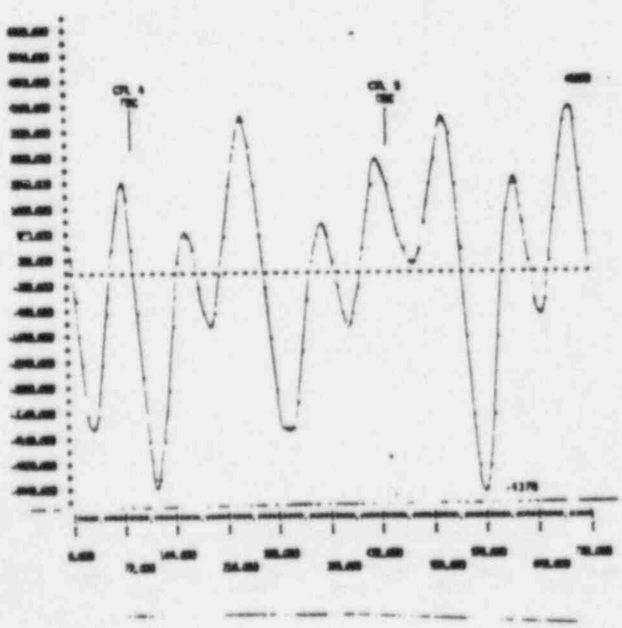
The mean amplitude is 0.59 degree, based on a maximum 0.62 degree and a minimum -0.55 degree.

16. NOMINAL TORSIONAL STRESS FOR SHAFT 5-6 AT 225 bmep, 472.5 rpm

The shaft stress at 225 bmep, 472.5 rpm is shown in the following table. Six (6) selected orders were used in this series of calculations.

Order	Stress, psi	Phase Angle $\emptyset_s$, degree
0.5	676	484.6
1.5	1729	219.2
2.5	1320	84.5
4.0	2466	73.6
4.5	684	64.0
5.5	333	24.8
SRSS	3442	

The dynamic stress amplitude (true sum) for the six (6) selected orders is shown as follows.



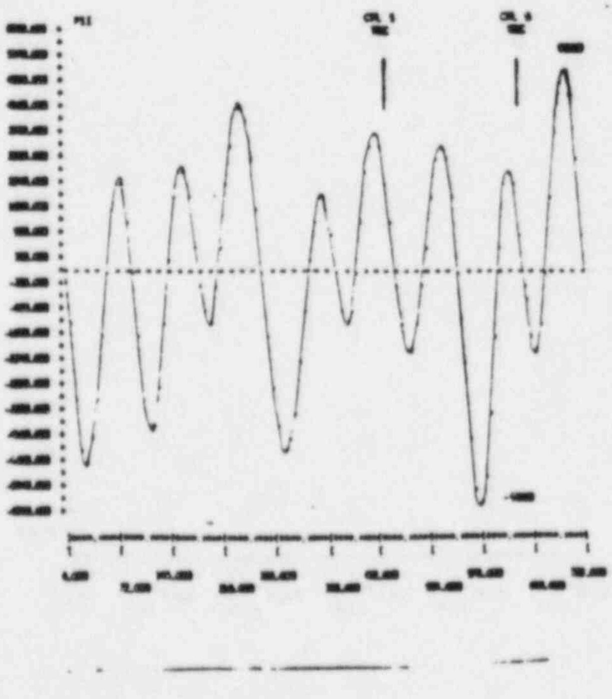
The mean stress is 4816 psi, based on a maximum 4263 psi and a minimum -5370 psi.

17. NOMINAL TORSIONAL STRESS FOR SHAFT 6-7 AT 225 bmep, 472.5 rpm

The shaft stress at 225 bmep, 472.5 rpm is shown in the following table. Six (6) selected orders were used in this series of calculations.

Order	Stress, psi	Phase Angle $\emptyset_s$, degree
0.5	261	350.8
1.5	1617	204.8
2.5	1269	93.3
4.0	2960	73.6
4.5	672	61.1
5.5	405	28.3
SRSS	3698	

The dynamic stress amplitude (true sum) for the six (6) selected orders is shown as follows.



The mean stress is 5490 psi, based on a maximum 5100 psi and a minimum -5880 psi.

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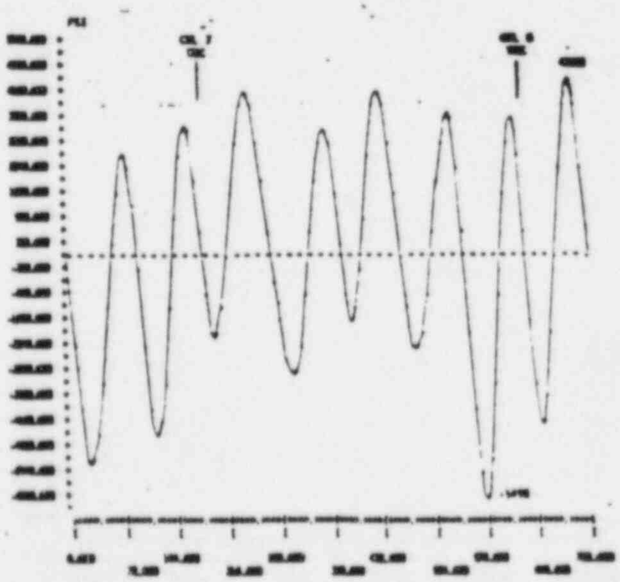
18. NOMINAL TORSIONAL STRESS FOR SHAFT 7-8 AT 225 bmep, 472.5 rpm

The shaft stress at 225 bmep, 472.5 rpm is shown in the following table. Six (6) selected orders were used in this series of calculations.

Order	Stress, psi	Phase Angle $\emptyset_S$, degree
0.5	884	184.86
1.5	989	52.8
2.5	822	126.5
4.0	3388	65.6
4.5	901	90.0
5.5	595	201.6
SRSS	3883	

The dynamic stress amplitude (true sum) for the six (6) selected orders is shown as follows.



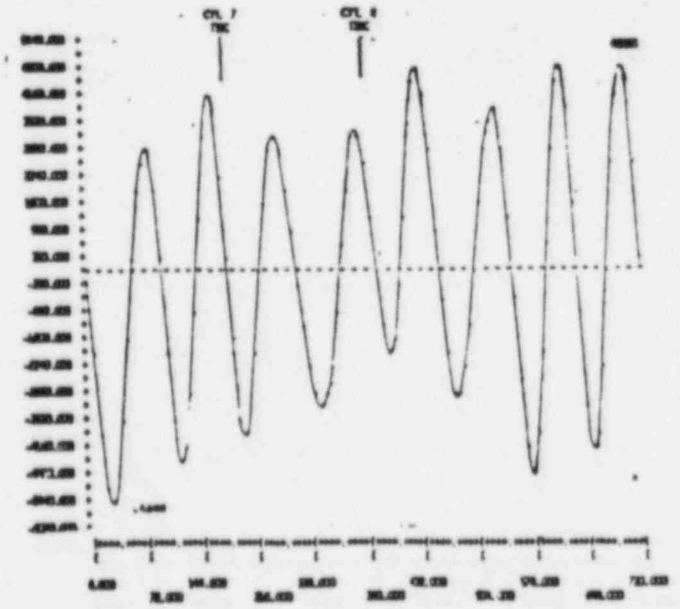
The mean stress is 5212 psi, based on a maximum 4928 psi and a minimum -5496 psi.

19. NOMINAL TORSIONAL STRESS FOR SHAFT 8-9 AT 225 bmep, 472.5 rpm

The shaft stress at 225 bmep, 472.5 rpm is shown in the following table. Six (6) selected orders were used in this series of calculations.

Order	Stress, psi	Phase Angle $\emptyset_S$, degree
0.5	627	140.5
1.5	704	92.4
2.5	626	96.25
4.0	3740	65.6
4.5	888	78.75
5.5	730	198.0
SRSS	4073	

The dynamic stress amplitude (true sum) for the six (6) selected orders is shown as follows.



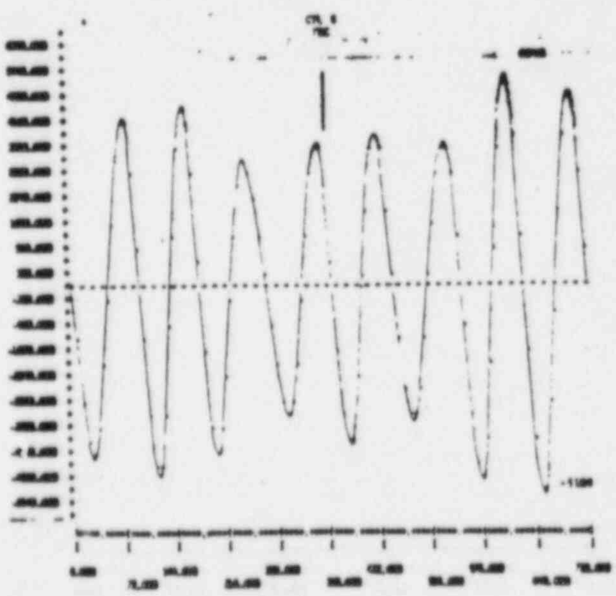
The mean stress is 5212 psi, based on a maximum 4928 psi and a minimum -5496 psi.

20. NOMINAL TORSIONAL STRESS FOR SHAFT 9-10 AT 225 bmep, 472.5 rpm

The shaft stress at 225 bmep, 472.5 rpm is shown in the following table. Six (6) selected orders were used in this series of calculations.

Order	Stress, psi	Phase Angle $\emptyset_s$, degree
0.5	5	399.6
1.5	95	204.2
2.5	228	94.4
4.0	4010	73.6
4.5	732	61.9
5.5	838	29.1
SRSS	4168	

The dynamic stress amplitude (true sum) for the six (6) selected orders is shown as follows.



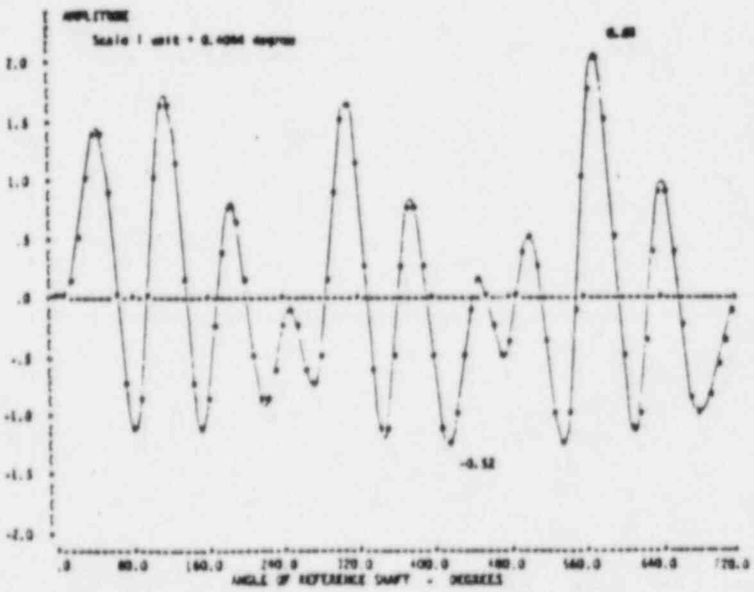
The mean stress is 5673 psi, based on a maximum 6243 psi and a minimum -5104 psi.

9. FREE END TORSIONAL AMPLITUDE AT 225 bmep, 427.5 rpm

TORVAP C gives torsional amplitudes at 225 bmep, 427.5 rpm for each order and for the sum of orders. Six (6) significant orders were selected for calculation. The square root of the sum of square (SRSS) amplitude was also calculated to compare with TDI data, as shown in the following table.

Order	Amplitude Θ, degree	Phase Angle ϕ_m, degree
0.5	0.07	37.3
1.5	0.14	83.5
2.5	0.11	22.0
4.0	0.27	28.7
4.5	0.05	21.6
5.5	0.33	51.1
SRSS	0.47	

The dynamic free end torsional amplitude (true sum) for the six (6) selected orders is shown as follows.



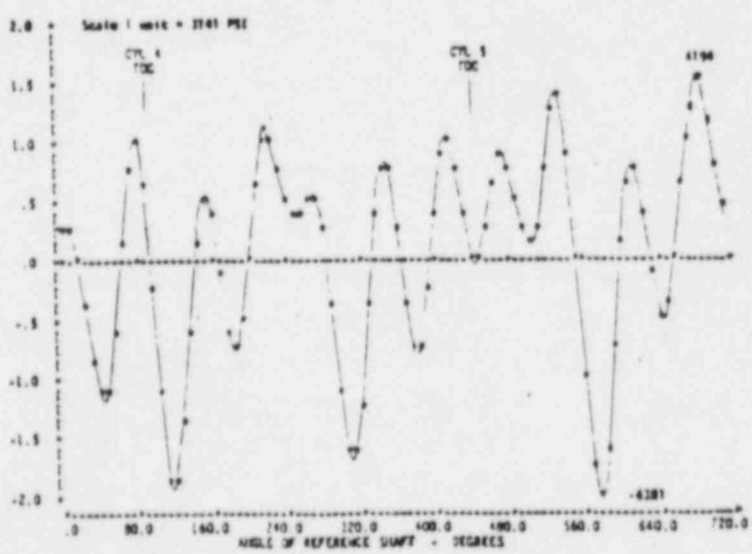
The mean amplitude is 0.67 degree, based on a maximum 0.81 degree and a minimum -0.52 degree.

10. NOMINAL TORSIONAL STRESS FOR SHAFT 5-6 AT 225 bmep, 427.5 rpm

The shaft stress at 225 bmep, 427.5 rpm is shown in the following table. Six (6) selected orders were used in this series of calculations. The SRSS torsional stress is also listed to compare with TDI data.

Order	Stress, psi	Phase Angle $\emptyset_s$, degree
0.5	674	484.7
1.5	1711	219.3
2.5	1275	85.3
4.0	1813	73.8
4.5	394	66.0
5.5	2255	17.7
SRSS	3678	

The dynamic stress amplitude (true sum) for the six (6) selected orders is shown as follows.



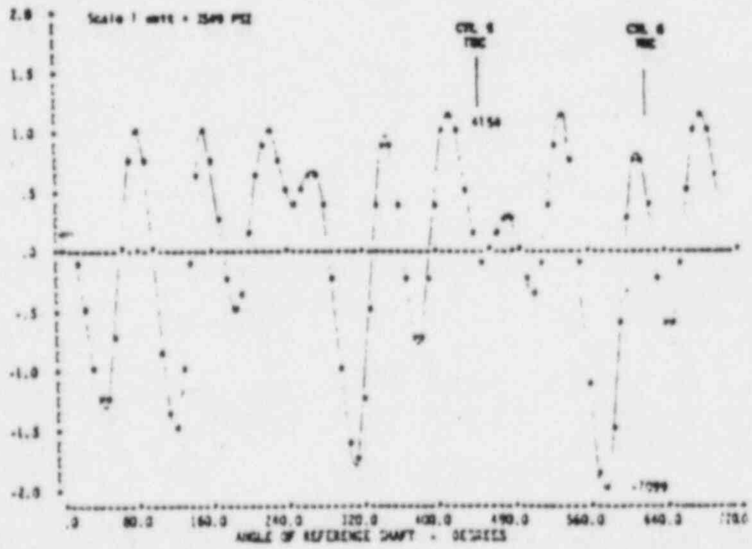
The mean stress is 5288 psi, based on a maximum 4194 psi and a minimum -6381 psi.

11. NOMINAL TORSIONAL STRESS FOR SHAFT 6-7 AT 225 bmep, 427.5 rpm

The shaft stress at 225 bmep, 427.5 rpm is shown in the following table. Six (6) selected orders were used in this series of calculations.

Order	Stress, psi	Phase Angle $\emptyset_s$, degree
0.5	260	350.7
1.5	1596	204.9
2.5	1217	93.3
4.0	2192	73.7
4.5	322	60.7
5.5	2555	18.0
SRSS	3941	

The dynamic stress amplitude (true sum) for the six (6) selected orders is shown as follows.



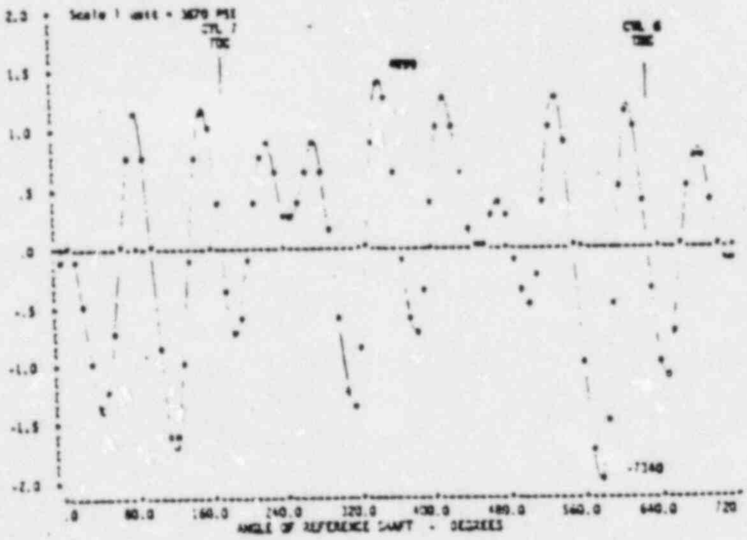
The mean stress is 5627 psi, based on a maximum 4154 psi and a minimum -7099 psi.

12. NOMINAL TORSIONAL STRESS FOR SHAFT 7-8 AT 225 bmep, 427.5 rpm

The shaft stress at 225 bmep, 427.5 rpm is shown in the following table. Six (6) selected orders were used in this series of calculations.

Order	Stress, psi	Phase Angle ϕ_s, degree
0.5	881	350.2
1.5	968	204.9
2.5	789	93.5
4.0	2532	73.7
4.5	516	59.0
5.5	2908	18.5
SRSS	4179	

The dynamic stress amplitude (true sum) for the six (6) selected orders. is shown as follows.



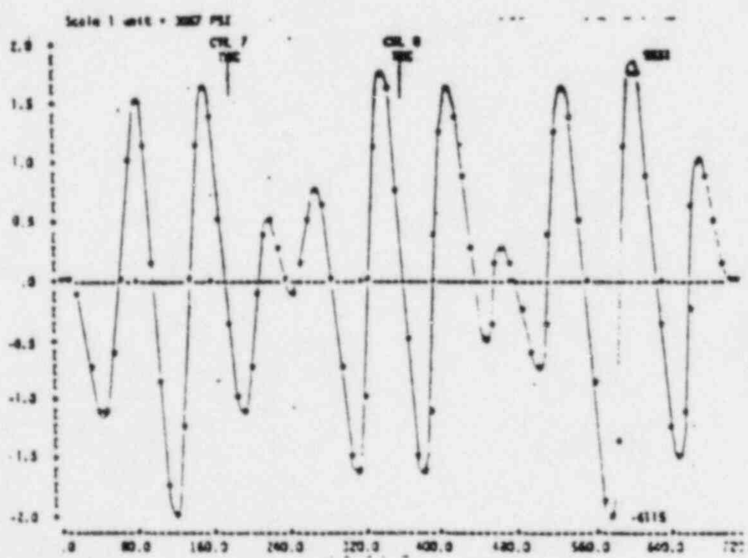
The mean stress is 6120 psi, based on a maximum 4899 psi and a minimum -7340 psi.

13. NOMINAL TORSIONAL STRESS FOR SHAFT 8-9 AT 225 bmep, 427.5 rpm

The shaft stress at 225 bmep, 427.5 rpm is shown in the following table. Six (6) selected orders were used in this series of calculations.

Order	Stress, psi	Phase Angle $\emptyset_s$, degree
0.5	624	439.8
1.5	687	177.9
2.5	582	106.7
4.0	2826	73.6
4.5	475	63.4
5.5	3088	18.9
SRSS	4352	

The dynamic stress amplitude (true sum) for the six (6) selected orders is shown as follows.



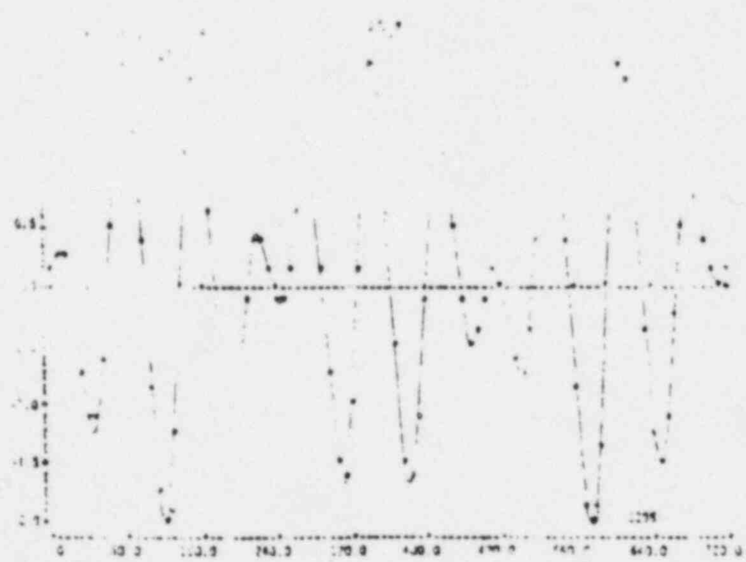
The mean stress is 5823 psi, based on a maximum 5531 psi and a minimum -6115 psi.

14. NOMINAL-TORSIONAL STRESS FOR SHAFT 9-10 AT 225 bmep, 427.5 rpm

The shaft stress at 225 bmep, 427.5 rpm is shown in the following table. Six (6) selected orders were used in this series of calculations.

Order	Stress, psi	Phase Angle [illegible], degree
0.5	4	402.6
1.5	76	[illegible]5.3
2.5	174	[illegible]
4.0	3071	[illegible]
4.5	306	[illegible]
5.5	3294	[illegible]19
SRSS	4517	

The dynamic stress amplitude (true sum) for the six (6) selected orders [illegible]



The mean stress is 6232 psi, based on a maximum 6067 psi and a minimum -6396 psi.

Torsional Stress at 3900 KW

TORSIONAL STRESS AT 3900 KW, 450 RPM

The overload condition of 3900 kw at 450 rpm is investigated and the results are tabulated as follows.

Shaft / Order	Torsional Stress, psi 5-6	6-7	7-8	8-9	9-10	Free End Degree
4.0	744	287	973	690	5	0.08
1.5	1866	1742	1061	753	93	0.15
2.5	1398	1340	879	653	215	0.12
4.0	2251	2713	3119	3463	3740	0.34
4.5	528	465	682	655	477	0.07
5.5	753	869	1109	1256	1405	0.12
SRSS	3450	3639	3777	3932	4030	0.42
True Sum	4757	5401	5228	4883	5192	0.59

In this case, fourth order Tn used is 35.0 psi corresponding to the 3900 kw, 251 bmep or 280 imep condition. The maximum single order torsional stress is at shaft section 9-10, with an amplitude of 3740 psi, safely below the DEMA allowable of 5000 psi.

The maximum sum of order SRSS figure is 4030 psi at shaft 9-10, and the mass sum of order true sum is 5401 psi at shaft 6-7. Both are based on TORVAP C six (6) order summation. They are again safely below the DEMA allowable of 7000 psi.

DEMA recommendation does not require calculation at overload condition. This is included here for our own reference only.

TORVAP Program Inputs and Outputs

TORVAP Methods

TORVAP (Torsional Vibration Analysis Program) is a computer program licensed from CAD Centre, U.K., and is available from Comshare, Inc., Ann Arbor, Michigan.

Program R provides Holzer natural frequency and determines the significant orders and stresses caused by f

Program C provides torque, stress and amplitude at each mass number, as well as sum of stress and amplitude for the six major orders evaluated. TORVAP C is basically a modal superposition method.

Input Data for TORVAP and Output Requested

Bore = 17", stroke = 21", crankpin = 12"
m = mass number
I = mass moment of inertia, lb·ft·sec^2
Rated speed = 450 rpm
Limits = 1000 to 8000 cpm, for search natural frequency
Excoder (orders to be studied) = 0.5, 1.0, 1.5, 2.0, 2.5, 3.5, 4.0, 4.5, 5.0, 5.5, 6.5, and 8.0
Engine cycle 4, banks 1
Full load, BHP/cyl = 611, rpm = 450, mech. eff. = 0.9 corresponding to imep = 250 psi
Torque-speed constant = constant at 250 imep
Gas tangential efforts reference Lloyd's
Firing order = 2-5-8-4-9-6-3-7 given mass number corresponding to cylinder number 1-4-7-3-8-5-2-6
Stress limit = 5000 psi based on DEMA guidelines for single order torsional vibratory stress, 7000 psi for sum of orders.
The reference shaft angle is assumed to be zero when the piston of cylinder 1 is at TDC, firing condition.

The output requested is:

Maximum amplitude of single order at main mass 1
True sum amplitude at main mass 1
True sum torsional stress at shaft 5-6, 6-7, 7-8, 8-9, 9-10

Input Sheets Required for TORVAP Program

```
(DATA) 100% LOAD, 100% SPEED, LLOYD Tm, NONE CYL MISFIRING
TITLE PEI 8I17*21 GEN SET 12 INCH PINS 100% LOAD 450 SPEED
UNITS 2 2 (US INDUSTRIAL UNIT)
THETAREF 'MAIN' 1
RANGE 'SPEED' 450 . 450 . 10
LIMITS 1000 . 7000 . 2
GENERATOR 'MAIN' 11
BRANCH 'MAIN' DESCRIPTION
1 'GEAR' 6.8 0. 0. 1. 58.1E6 0. 0. 0. 12.
2 'CYL 1' 49.2 0. 50. 1. 84.7E6 0. 0. 0. 12.
3 'CYL 2' 47.9 0. 50. 1. 84.7E6 0. 0. 0. 12.
4 'CYL 3' 47.9 0. 50. 1. 84.7E6 0. 0. 0. 12.
5 'CYL 4' 47.9 0. 50. 1. 84.7E6 0. 0. 0. 12.
6 'CYL 5' 47.9 0. 50. 1. 84.7E6 0. 0. 0. 12.
7 'CYL 6' 47.9 0. 50. 1. 84.7E6 0. 0. 0. 12.
8 'CYL 7' 47.9 0. 50. 1. 84.7E6 0. 0. 0. 12.
9 'CYL 8' 50.1 0. 50. 1. 76.9E6 0. 0. 0. 12.
10 'F.W.' 1100. 0. 0. 1. 276.7E6 0. 0. 0. 16.
11 'GEN' 2650. 0. 0. 1. 0. 0. 0. 0. 0.
EXCORDER 4.0 4.5 5.5 3.5 2.5 1.5 0.5 6.5 8.0 5.0 2.0 1.0
ENGINE CYCLE 4 'CSA'
BORE 17.0 STROKE 21.0 BANKS 1
FULLLOAD 611. 450. 0.9
TORCUESPEED CONSTANT 'C'
HARMONICS LLOYDS 'LLOYD'
EXCITATION
2 'CYL 1" 0.0 0. 0. 0. 816.4 0. 4.2 0.
3 'CYL 2' 540.0 0. 0. 0. 816.4 0. 4.2 0.
4 'CYL 3' 270.0 0. 0. 0. 816.4 0. 4.2 0.
5 'CYL 4' 90. 0. 0. 0. 816.4 0. 4.2 0.
6 'CYL 5' 450. 0. 0. 0. 816.4 0. 4.2 0.
7 'CYL 6' 630. 0. 0. 0. 816.4 0. 4.2 0.
8 'CYL 7' 180. 0. 0. 0. 816.4 0. 4.2 0.
9 'CYL 8' 360. 0. 0. 0. 816.4 0. 4.2 0.
LIMSTRESS LLOYDSAUX 'B'
OUTPUT
MAXAMP 'MAIN' 1
SCANTORQUE 'MAIN' 5 'MAIN' 6 'MAIN' 7 'MAIN' 8 'MAIN' 9
SUMAMP 'MAIN' 1
SUMTORQUE 'MAIN' 8
SUMTORQUE 'MAIN' 9
SUMTORQUE 'MAIN' 7
SUMTORQUE 'MAIN' 6
SUMTORQUE 'MAIN' 5
(END)
```

Torsional Stress at Rated Condition for
Second and Third Modes per TORVAP R

Torsional Stress of Shaft 9-10 at 225 bmep, for Second Mode

ORDER	NC	NC/NS	Σβ	Tn	ϵ_0	AT NC SPEED		AT 225 bmep	
						Mr	Sr	Mf	Sf
0.5	11150.5	24.78	1.51	88.5	0.00	41.5	1826	1.00	44
1.0	5575.2	12.39	.25	95.4	0.02	29.5	5092	1.01	9
1.5	3716.8	8.26	3.79	90.1	0.01	32.8	3692	1.01	114
2.0	2787.6	6.20	.70	74.5	0.08	19.3	18228	1.03	11
2.5	2230.1	4.96	3.79	62.0	0.01	36.1	2792	1.04	81
3.0	1858.4	4.13	.25	51.2	0.01	39.8	2076	1.06	4
3.5	1592.9	3.54	1.51	41.5	0.00	50.0	1030	1.09	22
4.0	1393.8	3.10	1.21	32.7	0.00	50.0	489	1.12	11
4.5	1238.9	2.75	1.51	25.3	0.00	50.0	628	1.15	14
5.0	1115.0	2.48	.25	19.2	0.00	50.0	80	1.19	2
5.5	1013.7	2.25	3.79	14.7	0.00	50.0	915	1.25	23
6.0	929.2	2.07	.70	11.3	0.00	50.0	130	1.31	3
6.5	857.7	1.91	3.79	9.0	0.00	50.0	563	1.38	16
7.0	796.5	1.77	.25	7.3	0.00	50.0	30	1.47	1
7.5	743.4	1.65	1.51	5.9	0.00	50.0	147	1.58	5
8.0	696.9	1.55	1.21	4.7	0.00	50.0	94	1.71	3
8.5	655.9	1.46	1.51	3.9	0.00	50.0	97	1.89	4
9.0	619.5	1.38	.25	3.4	0.00	50.0	14	2.12	1
9.5	586.9	1.30	3.79	3.1	0.00	50.0	191	2.43	9
10.0	557.5	1.24	.70	2.7	0.00	50.0	32	2.87	[illegible]

Torsional Stress of Shaft 9-10 at 225 bmep, for Third Mode

ORDER	NC	NC/NS	Σβ	Tn	θ_0	AT NC SPEED		AT 225 bmep	
						Mr	Sr	Mf	Sf
0.5	14000.8	31.11	.96	88.5	0.01	35.9	7460	1.00	208
1.0	7000.4	15.56	.86	95.4	0.23	15.0	101827	1.00	227
1.5	4666.9	10.37	3.10	90.1	0.02	26.6	18284	1.01	693
2.0	3500.2	7.78	2.52	74.5	1.39	9.6	387547	1.02	298
2.5	2800.2	6.22	3.10	62.0	0.02	29.2	13826	1.03	485
3.0	2333.5	5.19	.86	51.2	0.07	20.1	42911	1.04	103
3.5	2000.1	4.45	.96	41.5	0.00	43.4	4226	1.05	103
4.0	1750.1	3.89	1.97	32.7	0.01	38.2	6213	1.07	174
4.5	1555.6	3.46	.96	25.3	0.00	49.1	2916	1.09	65
5.0	1400.1	3.11	.86	19.2	0.00	50.0	2020	1.12	45
5.5	1272.3	2.83	3.10	14.7	0.00	41.9	4688	1.14	128
6.0	1166.7	2.59	2.52	11.3	0.00	47.2	3296	1.17	82
6.5	1077.0	2.39	3.10	9.0	0.00	47.4	3256	1.21	83
7.0	1000.1	2.22	.86	7.3	0.00	50.0	766	1.25	19
7.5	933.4	2.07	.96	5.9	0.00	50.0	693	1.30	18
8.0	875.1	1.95	1.97	4.7	0.00	50.0	1135	1.36	31
8.5	823.6	1.83	.96	3.9	0.00	50.0	460	1.43	13
9.0	777.8	1.73	.86	3.4	0.00	50.0	359	1.50	11
9.5	736.9	1.64	3.10	3.1	0.00	50.0	1166	1.59	37
10.0	700.0	1.56	2.52	2.7	0.00	50.0	848	1.70	29

PEI

CONS[redacted]

APPENDIX V

Reference List

Page [redacted]

Reference List

1. "Standard Practices for Low and Medium Speed Stationary Diesel and Gas Engines," Sixth Edition, Diesel Engine Manufacturers Association DEMA, 1972.

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3. "Torsional Vibration Analysis Program (TORVAP)," Computer Aided Design Centre, Madingley Road, Cambridge, UK CB3 OHB. June, 1975. Available from Comshare Inc., Ann Arbor, Michigan.

4. "Evaluation of Emergency Diesel Generator Crankshafts at Shoreham and Grand Gulf Nuclear Power Stations," Failure Analysis Associates, Palo Alto, California. March 30. 1984.

5. "Vibration Analysis for a Sound Generator-Set Desi[redacted]," Dr. Simon [redacted] Chen September 19, 1978. [redacted]

6. "Torsional Vibration Analysis Program (TORVAP)," Computer Aided Design Centre, Madingley Road, Cambrid[redacted] CB3 [redacted], 1975.

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8. "Field Test of Emergency Diesel Generator 103," Bercel and Hall, St and Webster Engineering Corporation. Figure 8-33. February, 1984.

9. "Emergency Diesel Generator Crankshaft Failure Investigation Shoreham Nuclear Power Station," Failure Analysis Associates, Palo Alto, California, October 31, 1983.

10. "Evalution Of Emergency Diesel Generator Crankshafts At Shoreham And Grand Gulf Nuclear Power Stations." FaAA, April 19, 1984.