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SECY-NRC

REPORT

ON

CRANKSHAFT TORSIONAL STRESSES

TRANSAMERICA DELAVAL MODEL DSR-48

Serial No. 74010/12

for

LONG ISLAND LIGHTING COMPANY

REVIEWED IN ACCORDANCE WITH ABS LETTER DATED
3 MAY 1984
AMERICAN BUREAU OF SHIPPING NEW YORK

Roland Yang.
April 4. 1984
Transamerica Delaval In
Oakland, CA.

TABLE OF CONTENTS

I N T R O D U C T I O N

This report consists of four sections and contains calculations. test data and operating experience, which Transamerica Delaval Inc. (TDI) considers relevant material to establish the adequacy of these DSR-48 engine generator sets.

The application of these units are for emergency standby service in the LILCO Shoreham Nuclear Power Plant. These units are rated at 3500 KW and have an overload rating of 3900 KW (111.4%) allowed two hours per day of twenty four hours.

Section One. Torsional Analysis.

A four page introduction is included here, explaining the method and nomenclature used in the torsional critical speed analysis.

The mass elastic system in the analysis reflects the piston skirt which has since been superceded. The extra reciprocating weights due to the heavier replacement piston shirt have been evaluated and the change concluded to be negligible. The effect of the 111.4% overload on torsional stress levels are shown in pages 12 to 15. Due to the close proximity of the calculated stresses to the ABS allowable stress, we elected to include Section Two..

Also included in this section are mass elastic system parameters of other DSR-48 engine generators of identical rating, to establish the similarity of these units, especially from a torsional standpoint.

Section Two. Torsiograph Tests.

Measurements by FaAA/SWEC and TDI are presented here , along with TDI measurements on other DSR-48 engine generators of identical rating and similar mass elastic system. Here again, the intent is to establish the similiarity between the various DSR-48 engine generators. The stresses evaluated from the torsiograph measurements are still in close proximity to the ABS allowable and therefore in Section Three, we present the actual strain gage measurements, taken on the subject shaft in January, 1984.

Section Three. Strain Gage Test.

Here the measured strains are listed along with the corresponding stresses. With the similar grade of crankshaft material, the endurance limit of the shaft is established and finally the margin of safety determined using the Goodman diagram. A factor of safety of the replacement crankshaft is 1.48 without the benefit of shot peening. The factor of safety is 1.75 as determined by FaAA, when the effect of shot peening is taken into account and taken to increase the endurance limit by 20%.

Section Four. Operating Hours Logged.

There are seventeen engine generator sets of similar configuration and identical rating in Saudi Arabia with considerable operating hours. These similar units are running regularly and generating power today. Worthy of note are the DSR-48 units at Rafha, with 5500 hours at 3200 KW on one unit and 6200 hours at 3300 KW and 8250 hours at 3200 KW for the other two. The similarity of these units are listed in page 17. Examination of the mass elastic data (I & K) and the torsional natural frequencies (N) listed, will show that there are no essential differences between LILCO and the rest of the units. The LILCO units are undergoing performance tests at their Shoreham Nuclear Power Station, the total hours logged at loads of 3500 KW and above, are shown on page 28.

Summary.

Based on the foregoing calculations, test data and operating logs, Transamerica Delaval, Inc. considers the adequacy of these DSR-48 engine generator sets to be established for the intended service at the LILCO Shoreham Nuclear Power Station.

INTRODUCTION

TO

TORSIONAL CRITICAL SPEED ANALYSIS

The engine generator system is modeled as a system of moments of inertia interconnected by torsional springs.

The standard procedure is to concentrate all the moment of inertia of each crankthrow, including rod and piston at the corresponding cylinder center position. The moment of inertia per cylinder is obtained by summing the moment of inertias of the journal, crankpin, two crankwebs, counterweights (if used), rotating part of connecting rod and the equivalent inertia of the reciprocating parts, (namely the upper end weight of the rod and the piston).

The moment of inertia (I) is calculated by dividing the WK^2 by gravitational acceleration (g). WK^2 is obtained by multiplying the weight of the part by the square of the radius of gyration (K). WK^2 therefore, would be in Lb. In.^{2} or Lb.Ft.^{2} units and inertia (I) in Lb.In.Sec.^{2} or Lb.Ft.Sec.^{2} units. The inertia values, thus represent the concentrated inertia of the moving parts at each crankthrow.

From our torsional vibration analysis data files, we obtained the appropriate values of inertia or stiffness to make up the model.

The procedure and notations used in this analysis are explained below.

NATURAL FREQUENCY EVALUATION

This is done by Holzer's method. In our Holzer's tabulation, the definition of the notations are as follows:

ω^2 EIGENVALUE (Omega squared), in 10^6 radian/second^2

NO.	Mass number, counted from the free end of the engine
INERTIA	Inertia of the various masses, in lb.in.sec.^{2}
THETA	Relative amplitude or angular position, in radians.
IOM2T	Product of INERTIA $*\omega^2*$ THETA, and is the vibratory torque due to each mass, in 10^6 ft.lb.
SIGNAM	Summation of the vibratory torques, in 10^6 ft.lb.
SHAFTK	Torsional stiffness of shaft, in 10^6 ft.lb/radian.
DTHETA	Quotient of SIGNAM divided by SHAFTK. This is the relative angular displacement between masses. The unit is radians.

STRESS EVALUATION

From the Holzer Tabulation, we determine the following, which are later used in the stress calculations.

SIGMA I * THETA ** 2 — This is the summation of the products of INERTIA and THETA squared. The first of the two printed, is that of the engine up to and including the last crankthrow and the second, that of the whole system.

T INT and T EXT — These are the maximum stresses in the shafting within the engine (INTernal) and outside of the engine (EXTernal). The stresses are evaluated for each section of shafting by this formula:

$$\text{SIGNAM} * \frac{\pi}{180} * \frac{16}{\pi D^3}$$

Only the maximums are printed out.

STRESSED DIAMETER OF EXTERNAL SHAFT — This is the diameter of the external shaft at which the maximum stress occurs.

EQUILIBRIUM AMPLITUDE — If the applied and resisting torques are applied and suddenly removed, the shaft is put into a state of free vibration and the curve of angular displacement can be analyzed into a series of normal elastic curves, each corresponding to one of the normal modes of free vibration of which the system is capable. The amplitdude of any of these modes of vibration under the above conditions is referred to as the EQUILIBRIUM AMPLITUDE, since it is the amplitude which is attained without any magnification due to resonance with an external pulsating couple. It is determined by:

$$\frac{\text{Piston Area} * \text{Crank Radius} * 180}{\omega^2 * 10^6 * \Sigma I\theta^2 \pi} \; T_N * N$$

This is left in the form- - - - Constant $_1 * T_N * \Sigma\theta_N$

F IN — The work output into the system and is determined by:
* Piston Area * Crank Radius * ϕ * $T_N \Sigma\theta_N$
This is left in the form - - - -Constant $_2 * T_N * \Sigma\theta_N \phi$

FE — Hysteresis damping due to friction, etc. and is determined by:

$$\frac{\pi * \omega^2 * 10^6 * \Sigma I\theta^2 * \phi^2}{25}$$

FD Viscous Damper damping, from this expression:

$$\frac{WR^2 \text{ Damper Ring } (\text{lb.ft.}^2) * 12\pi\omega^3 * \left(\frac{180}{\pi}\right)^2 * 33000 * \pi\omega^2}{G * 86700000 * 30 * \omega}$$

FCR Rubber coupling damping, and is determined by:

Coupling Stiffness * (.00037(D-20) (D-30)) * DTHETA of cplg^2

D is the durometer of the coupling rubber.

FCS Steel coupling damping, and is determined by:

Damping coefficient * DTHETA of coupling ^{2}

FP Propeller damping from this expression:

$$\frac{\pi * 12 * 1.82 * 10^6 * \omega^2 * 10^6 * \text{bhp} * \text{Gear Ratio} * \text{THETA of prop}}{\text{Harmonic} * \text{RPM}^3}$$

F INT Static Stress INTernal, which is the product of T INT and EQUILIBRIUM AMPLITUDE. This is left in the form -------- Constant $_3$ * T_N * $\Sigma\theta_N$

F EXT Static Stress EXTernal, which is the product of T EXT and EQUILIBRIUM AMPLITUDE. This is left in the form --------- Constant $_4$* T_N * $\Sigma\theta_N$

Total damping is the sum of FE + FD + FCR + FCS + FP

In the stress calculation tabulation we have the following columns, some of which are calculated from the values previously determined.

ORDER Harmonic of the mode of vibration.

RPM Resonant speed of the harmonic, or critical speed.

TN T_N, which is the harmonic component determined from the Fourier Analysis of a cylinder pressure diagram.

VEC $\Sigma\theta_N$. This is the vector summation of THETA of the crank-throws, for the engine's firing order, of the particular harmonic.

TSTINT Static stress INTernal and is determined by F INT, which is:

Constant $_3$* T_N * $\Sigma\theta_N$

TSTEXT Static stress EXTernal and is determined by F EXT, which is:

Constant $_4$ * T_N * $\Sigma\theta_N$

PHI ϕ, the front end amplitude in degrees at resonance, and is determined by:

F IN = Total Damping

TMAXI — Maximum internal resonance stress in PSI, which is:

PHI * T INT

TMAXI — Maximum external resonance stresses in PSI, which is:

PHI * T EXT

When plotting the stresses versus RPM, the off resonant stresses are determined by:

$$\frac{\text{TSINT or TSTEXT}}{\text{MAGNIFICATION FACTOR}}$$

$$\text{Where Magnification Factor} = 1 - \left(\frac{\text{RPM}}{\text{Resonant RPM}}\right)^2$$

SECTION ONE

TORSIONAL ANALYSIS

TORSIONAL AND LATERAL CRITICAL SPEED ANALYSIS

ENGINE NUMBERS 74010/12

DELAVAL-ENTERPRISE ENGINE MODEL DSR-48

3500 KW/4889 BHP AT 450 RPM

FOR

STONE & WEBSTER ENGINEERING CORP.

LONG ISLAND LIGHTING COMPANY

TRANSAMERICA DELAVAL ENGINE & COMPRESSOR DIVISION

550 - 85th AVENUE
OAKLAND, CALIFORNIA 94621

By: ROLAND YANG

AUGUST 22, 1983

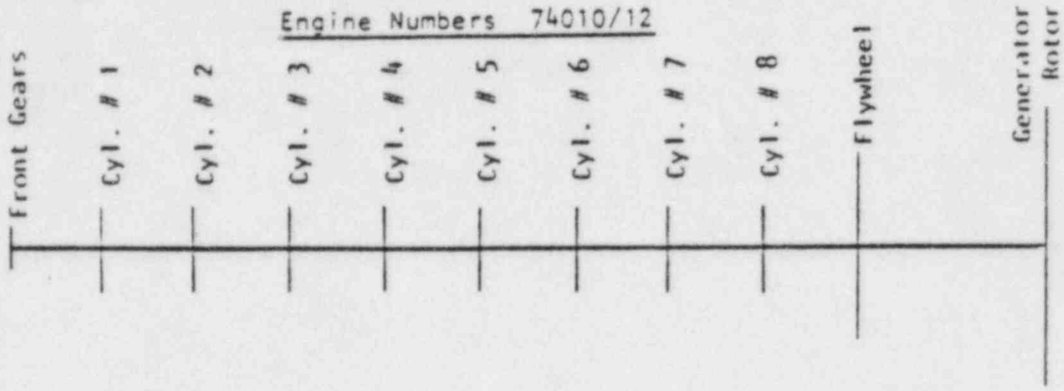
LONG ISLAND LIGHTING COMPANY

Delaval-Enterprise Engine Model DSR-48

3500 KW/4889 BHP at 450 RPM

225.6 BMEP

Engine Numbers 74010/12



Mass Elastic System

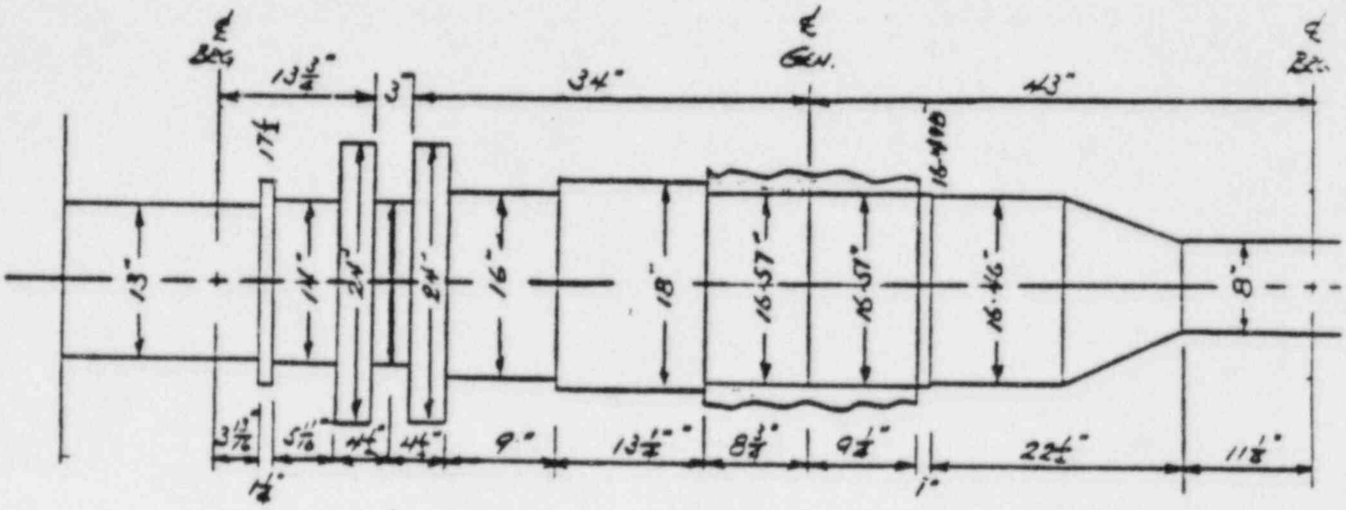
Crankshaft Gear	26.52	
Water Pump Drive	63.10	
Cams & Idlers	119.70	
Shaft	9.63	
	218.95 lb.ft.^{2}	I = 6.805 lb.ft.sec.^{2}
Crankthrow No.1		
As No.2 crankthrow	1541.85	
Shaft	41.81	
	1583.66 lb.ft.^{2}	I =49.222 lb.ft.sec.^{2}
Crankthrow No. 2 to 7		
Journal	43.13	
Crankpin	202.17	
Two Webs	679.81	
Reciprocating Wt.	309.99	
Rotating Weight	306.75	
	1541.85 lb.ft.^{2}	I =47.922 lb.ft.sec.^{2}
Crankthrow No.8		
As No. 2 Crankthrow	1541.85	
Shaft	71.63	
	1613.48 lb.ft.^{2}	I =50.149 lb.ft.sec.^{2}
Crankshaft Flange	255.08	
Flywheel 73 x 6½	34764.	
Generator Shaft	374.	
	35393.08 lb.ft.^{2}	I =1100.052 lb.ft.sec.^{2}
Generator Rotor	85275 lb.ft.^{2}	I =2650.432 lb.ft.sec.^{2}

Total WK^2 = 133335 lb.ft.^{2}

NOTE: The reciprocating weight used above are based on 800 lb. The replacement "AE" psiton skirt is 26 lb. heavier. We have evaluated the effect of this weight difference and conclude that the change in natural frequencies and stresses is less than .5% and therefore negligible.

EQUIVALENT LENGTH $D_e = 1''$

Front Gear to Cylinder No.1	.0001661''	$K = 58.121 \times 10^6$ ft.lb./rad.
Between Cylinders	.0011394''	$K = 84.727 \times 10^6$ ft.lb./rad.
Cylinder No. 8 to Flywheel	.0012547''	$K = 76.941 \times 10^6$ ft.lb./rad.
Flywheel to Generator		



$$L_e = \frac{3 - .0515 \times 16.25}{20^4} + \frac{.055 \times 16 + 9 + .03 \times 16}{16^4}$$

$$+ \frac{- .03 \times 16 + 13.25 - .021 \times 16.572}{18^4} + \frac{.021 \times 16.572 + 16.5/4}{16.572^4}$$

$$= .0000132 + .0001580 + .0001183 + .000593 = .0003488''$$

$$K = 276.773 \times 10^6 \text{ ft.lb./radian}$$

Shafting Diameters

Front gear ro Cylinder No. 1	8''
Crankpin	12''
Cylinder No. 8 to Flywheel	12''
Generator Shaft	16''

Flywheel Weight	6935 lb.
Gen. Rotor Weight	17150 lb.

Torsional Natural Frequencies

First Mode	2323 vpm
Second Mode	5575 vpm
Third Mode	7000 vpm

Electrical Natural Frequency

$$N = \frac{35200}{450}\sqrt{\frac{11260 \times 60}{133335}} = 175 \text{ vpm}$$

Generator Shaft Lateral Natural Frequency 3191 vpm

MODE 1
OMEGA SQUARED IN (RADIANS/SECOND)**2 = .05918702
NATURAL FREQUENCY IN V.P.M. = 2323.19

NO.	INERTIA	THETA	IOM2T	SIGMA M	SHAFT K	DTHETA
1	6.8	1.00000	.403	.403	58.1	.00693
2	49.2	.99307	2.893	3.296	84.7	.03890
3	47.9	.95417	2.706	6.002	84.7	.07084
4	47.9	.88333	2.505	8.508	84.7	.10041
5	47.9	.78291	2.221	10.728	84.7	.12662
6	47.9	.65629	1.861	12.590	84.7	.14859
7	47.9	.50770	1.440	14.030	84.7	.16559
8	47.9	.34211	.970	15.000	84.7	.17704
9	50.1	.16507	.490	15.490	76.9	.20132
10	1100.1	-.03625	-2.360	13.130	276.8	.04744
11	2650.4	-.08369	-13.129	.001		

MODE 1
OMEGA SQUARED .0592
NATURAL FREQUENCY 2323.1900
SIGMA I*THETA**2 2385.7810
SIGMA I*THETA**2 2707.4896
T INT 9561.73
T EXT 3419.27
STRESSED DIAMETER OF EXTERNAL SHAFT 16.00
EQUILIBRIUM AMPLITUDE .00065213:052
F IN 7487.33
F INT 8.15
F EXT 2.91
F E 17744629.
F D 0.
F CR 0.
F CS 0.
F P 0.

ORDER	RPM	TN	VEC	TSTINT	TSTEXT	PHI	TMAXI	TMAXE
.5	4646	155.86	.701	889.8	318.2	.738	7053.	2522.
1.0	2323	94.58	.146	112.7	40.3	.158	1513.	541.
1.5	1548	129.52	1.394	1471.3	526.1	1.454	13904.	4972.
2.0	1161	41.05	.376	125.7	45.0	.670	6408.	2292.
2.5	929	71.71	1.394	814.6	291.3	1.050	10036.	3589.
3.0	774	16.16	.146	19.3	6.9	.044	423.	151.
3.5	663	42.79	.701	244.3	87.4	.376	3596.	1286.
4.0	580	27.66	5.285	1191.1	425.9	2.066	19750.	7063.
4.5	516	23.76	.701	135.7	48.5	.216	2063.	738.
5.0	464	17.37	.146	20.7	7.4	.061	587.	210.
5.5	422	12.84	1.394	145.8	52.2	.433	4138.	1480.
6.0	387	5.68	.376	17.4	6.2	.052	494.	176.
6.5	357	4.49	1.394	51.0	18.2	.151	1447.	517.
7.0	331	3.69	.146	4.4	1.6	.013	125.	45.
7.5	309	3.05	.701	17.4	6.2	.052	494.	176.
8.0	290	2.52	5.285	108.4	38.8	.322	3076.	1100.
8.5	273	2.26	.701	12.9	4.6	.038	366.	131.
9.0	258	1.97	.146	2.3	.8	.007	67.	24.
9.5	244	1.53	1.394	17.3	6.2	.051	492.	176.
10.0	232	1.27	.376	3.9	1.4	.012	110.	39.
10.5	221	1.14	1.394	13.0	4.6	.038	368.	132.
11.0	211	1.02	.146	1.2	.4	.004	34.	12.
11.5	202	.89	.701	5.1	1.8	.015	144.	51.
12.0	193	.79	5.285	33.9	12.1	.100	961.	344.

MODE 2
OMEGA SQUARED IN (RADIANS/SECOND)**2 = .34090021
NATURAL FREQUENCY IN V.P.M. = 5575.52

NO.	INERTIA	THETA	IOMET	SIGMA M	SHAFT K	DTHETA
1	6.8	1.00000	2.320	2.320	58.1	.03991
2	49.2	.96009	16.110	18.430	84.7	.21752
3	47.9	.74257	12.131	30.561	84.7	.36070
4	47.9	.38187	6.238	36.799	84.7	.43433
5	47.9	-.05246	-.857	35.942	84.7	.42421
6	47.9	-.47667	-7.787	28.155	84.7	.33230
7	47.9	-.80898	-13.216	14.939	84.7	.17632
8	47.9	-.98530	-16.096	-1.157	84.7	-.01366
9	50.1	-.97164	-16.611	-17.768	76.9	-.23093
10	1100.1	-.74070	-277.770	-295.538	276.8	-1.06780
11	2650.4	.32710	295.542	.004		

MODE 2
OMEGA SQUARED .3409
NATURAL FREQUENCY 5575.5200
SIGMA I*THETA**2 2578.9556
SIGMA I*THETA**2 13306.1514
T INT 22715.43
T EXT 76363.02
STRESSED DIAMETER OF EXTERNAL SHAFT 16.00
EQUILIBRIUM AMPLITUDE .000030103703
F IN 7487.33
F INT .68
F EXT 2.32
F E 110479321.
F D 0.
F CR 0.
F CS 0.
F P 0.

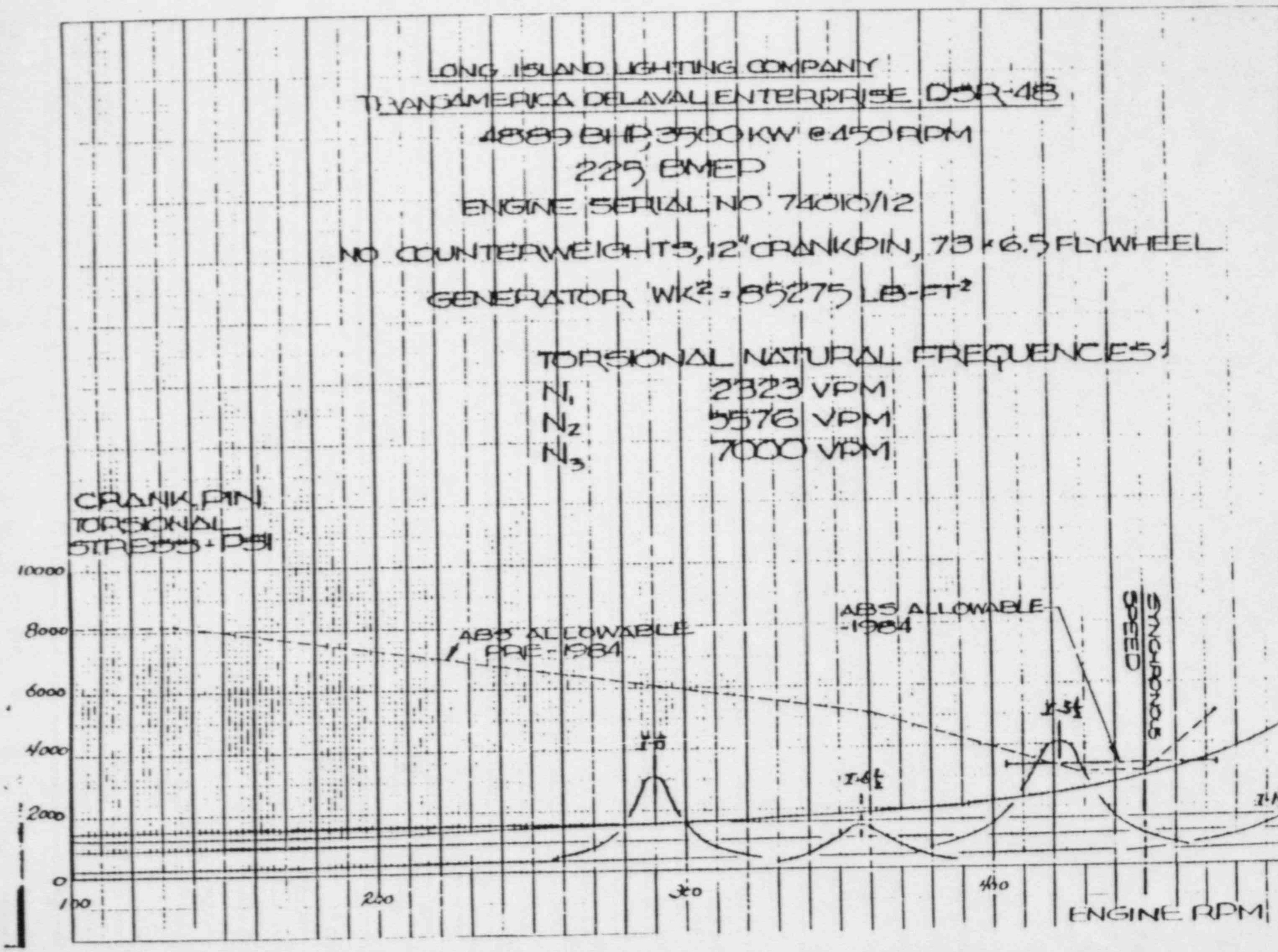
ORDER	RPM	TN	VEC	TSTINT	TSTEXT	PHI	TMAXI	TMAXE
.5	11151	155.86	1.508	160.7	544.6	.255	5792.	19624.
1.0	5575	94.58	.253	16.3	55.4	.044	997.	3379.
1.5	3717	129.52	3.789	335.6	1137.0	.635	14418.	48851.
2.0	2787	41.05	.702	19.7	66.8	.201	4567.	15474.
2.5	2230	71.71	3.789	185.8	629.5	.458	10407.	35260.
3.0	1858	16.16	.253	2.8	9.5	.012	279.	945.
3.5	1593	42.79	1.508	44.1	149.5	.130	2953.	10006.
4.0	1393	27.66	1.211	22.9	77.6	.076	1726.	5849.
4.5	1239	23.76	1.508	24.5	83.0	.075	1694.	5739.
5.0	1115	17.37	.253	3.0	10.2	.009	209.	709.
5.5	1013	12.84	3.789	33.3	112.7	.105	2386.	8085.
6.0	929	11.42	.702	5.5	18.6	.015	352.	1192.
6.5	857	9.08	3.789	23.5	79.7	.066	1500.	5084.
7.0	796	7.61	.253	1.3	4.5	.004	82.	278.
7.5	743	6.17	1.508	6.4	21.6	.018	405.	1373.
8.0	696	5.00	1.211	4.1	14.0	.012	269.	911.
8.5	655	4.74	1.508	4.9	16.6	.013	300.	1018.
9.0	619	4.08	.253	.7	2.4	.002	44.	149.
9.5	586	2.58	3.789	6.7	22.6	.022	510.	1728.
10.0	557	1.87	.702	.9	3.0	.003	79.	266.
10.5	531	1.69	3.789	4.4	14.8	.017	382.	1293.
11.0	506	1.42	.253	.2	.8	.001	23.	77.
11.5	484	1.15	1.508	1.2	4.0	.005	118.	400.
12.0	464	.96	1.211	.8	2.7	.005	103.	349.

MODE 3
OMEGA SQUARED IN (RADIANS/SECOND)**2 = .53738427
NATURAL FREQUENCY IN V.P.M. = 7000.26

NO.	INERTIA	THETA	IOM2T	SIGMA M	SHAFT K	DTHETA
1	6.8	1.00000	3.657	3.657	58.1	.06292
2	43.2	.93708	24.787	28.444	84.7	.33571
3	47.9	.60137	15.487	43.931	84.7	.51850
4	47.9	.08288	2.134	46.065	84.7	.54369
5	47.9	-.46081	-11.867	34.198	84.7	.40362
6	47.9	-.86443	-22.261	11.936	84.7	.14088
7	47.9	-1.00531	-25.889	-13.953	84.7	-.16488
8	47.9	-.84063	-21.648	-35.601	84.7	-.42019
9	50.1	-.42044	-11.331	-46.932	76.9	-.60997
10	1100.1	.18953	112.041	65.109	276.8	.23524
11	2650.4	-.04571	-65.109	.000		

MODE 3
OMEGA SQUARED .5374
NATURAL FREQUENCY 7000.2600
SIGMA I*THETA**2 2375.2154
SIGMA I*THETA**2 2997.4770
T INT 28970.37
T EXT 16955.47
STRESSED DIAMETER OF EXTERNAL SHAFT 16.00
EQUILIBRIUM AMPLITUDE .000084773254
F IN 7487.33
F INT 2.46
F EXT 1.44
F E 160397580.
F D 0.
F CR 0.
F CS 0.
F P 0.

ORDER	RPM	TN	VEC	TSTINT	TSTEXT	PHI	TMAXI	TMAXE
.5	14000	155.86	.955	365.7	214.0	.111	3223.	1887.
1.0	7000	94.58	.857	199.0	116.4	.102	2969.	1738.
1.5	4666	129.52	3.103	986.9	577.6	.358	10371.	6070.
2.0	3500	41.05	2.525	254.6	149.0	.498	14432.	8446.
2.5	2800	71.71	3.103	546.4	319.8	.258	7486.	4381.
3.0	2333	16.16	.857	34.0	19.9	.029	831.	486.
3.5	2000	42.79	.955	100.4	58.8	.057	1644.	962.
4.0	1750	27.66	1.970	133.9	78.3	.085	2468.	1445.
4.5	1555	23.76	.955	55.8	32.6	.033	943.	552.
5.0	1400	17.37	.857	36.5	21.4	.022	623.	365.
5.5	1272	12.84	3.103	97.8	57.3	.059	1717.	1005.
6.0	1166	11.42	2.525	70.8	41.4	.038	1111.	651.
6.5	1076	9.08	3.103	69.2	40.5	.037	1079.	632.
7.0	1000	7.61	.857	16.0	9.4	.008	245.	143.
7.5	933	6.17	.955	14.5	8.5	.008	226.	132.
8.0	875	5.00	1.970	24.2	14.2	.013	384.	225.
8.5	823	4.74	.955	11.1	6.5	.006	167.	98.
9.0	777	4.08	.857	8.6	5.0	.005	131.	77.
9.5	736	2.58	3.103	19.7	11.5	.013	367.	215.
10.0	700	1.87	2.525	11.6	6.8	.009	248.	145.
10.5	666	1.69	3.103	12.9	7.5	.009	275.	161.
11.0	636	1.42	.857	3.0	1.7	.002	67.	39.
11.5	608	1.15	.955	2.7	1.6	.002	66.	38.
12.0	583	.96	1.970	4.7	2.7	.004	120.	70.



LONG ISLAND LIGHTING COMPANY
TRANSAMERICA DELAVAL ENTERPRISE DSR-48
4889 BHP, 3500 KW @ 450 RPM
225 BMEP
ENGINE SERIAL NO 74010/12
NO COUNTERWEIGHTS, 12" CRANKPIN, 73 x 6.5 FLYWHEEL
GENERATOR WK² = 85275 LB-FT²
TORSIONAL NATURAL FREQUENCIES:
N1 2323 VPM
N2 5576 VPM
N3 7000 VPM
CRANKPIN TORSIONAL STRESS - PSI
10000
8000
6000
4000
2000
0
ABS ALLOWABLE PRE-1984
ABS ALLOWABLE -1984
SYNCHRONOUS SPEED
I-4
I-4½
I-3½
I-5½
I-6½
100
200
300
400
500
ENGINE RPM

The preceeding pages show the analysis based on the nameplate rating of the engines. To illustrate the changes in amplitudes and stress levels when operating at 111.4% (3900 KW) or overload condition allowed for two hours per day, we supplemented the analysis with the following pages.

The changes in the amplitude and stress between the two load conditions are reflected in the "T sub n" (T_n) values. The following is a listing of the 111.4% T_n divided by the 100% T_n for the orders or harmonics calculated. The ratios show the increase in amplitude and stress level due to the extra load at 111.4% load operation.

ORDER	FIRST MODE	SECOND MODE	THIRD MODE
.5	1.1040	1.1040	1.1040
1.0	1.0777	1.0777	1.0777
1.5	1.0963	1.0963	1.0963
2.0	.8877	.8877	.8877
2.5	1.0817	1.0817	1.0817
3.0	1.1733	1.1733	1.1733
3.5	1.0694	1.0694	1.0694
4.0	1.0600	1.0600	1.0600
4.5	1.0669	1.0669	1.0669
5.0	1.0662	1.0662	1.0662
5.5	1.0639	1.0639	1.0639
6.0	1.0370	1.0727	1.0727
6.5	1.0379	1.0727	1-0727
7.0	1.0379	1.0749	1.0749
7.5	1.0361	1.0729	1.0729
8.0	1.0357	1.0700	1.0700
8.5	1.0398	1.0738	1.0738
9.0	1.0406	1.0735	1.0735
9.5	1.0196	1.0581	1.0581
10.0	1.0157	1.0481	1.0481
10.5	1.0175	1.0473	1.0473
11.0	1.0098	1.0423	1.0423
11.5	1.0112	1.0261	1.0261
12.0	1.0000	1.0313	1.0313

From the above, the increase in amplitude or stress level due to the fourth order at 111.4% load will be about 6% higher than that at 100%. If we calculate the stress levels at 450 RPM due to the .5, 1.5, 2.5, 4.0, 4.5, 5.0, 5.5 orders and sum these by the Square Root of the Sum of the Squares, the overall amplitude at 111.4% (4154 psi) will be about 7% higher than that (3879 psi) at 100% load.

MODE 1
OMEGA SQUARED IN (RADIANS/SECOND)**2 = .05918702
NATURAL FREQUENCY IN V.P.M. = 2323.19

NO.	INERTIA	T-ETA	IOMET	SIGMA M	SHAFT K	DT-ETA
1	6.8	1.00000	.403	.403	58.1	.00533
2	49.2	.99307	2.893	3.296	84.7	.03890
3	47.9	.95417	2.706	6.002	84.7	.0708-
4	47.9	.88333	2.505	8.506	84.7	.10041
5	47.9	.78291	2.221	10.728	84.7	.12662
6	47.9	.65629	1.861	12.590	84.7	.14859
7	47.9	.50770	1.440	14.030	84.7	.16559
8	47.9	.34211	.970	15.000	84.7	.1770-
9	50.1	.16507	.490	15.490	76.9	.20132
10	1100.1	-.03625	-2.360	13.130	276.8	.047-4
11	2650.4	-.08369	-13.129	.001		

MODE 1
OMEGA SQUARED .0592
NATURAL FREQUENCY 2323.1900
SIGMA I*THETA**2 2385.7810
SIGMA I*T-ETA**2 2707.4696
T INT 9561.73
T EXT 3419.27
STRESSED DIAMETER OF EXTERNAL SHAFT 16.00
EQUILIBRIUM AMPLITUDE .000852131052
F IN 7487.33
F INT 6.15
F EXT 2.91
F E 17744629.
F D 0.
F CP 0.
F CS 0.
F F 0.

ORDER	RPM	TN	VEC	TSTINT	TSTEXT	PHI	TMAXI	TMAXE
.5	4646	172.07	.701	983.4	351.3	.805	7710.	2757.
1.0	2323	101.93	.146	121.5	43.4	.164	1571.	562.
1.5	1548	141.99	1.394	1612.9	576.8	1.559	14909.	5331.
2.0	1161	36.44	.376	111.6	39.9	.659	6305.	2255.
2.5	929	77.57	1.394	881.1	315.1	1.099	10506.	3756.
3.0	774	18.96	.146	22.6	8.1	.042	404.	145.
3.5	663	45.76	.701	261.3	93.4	.389	3717.	1329.
4.0	580	29.32	5.285	1262.5	451.5	2.119	20257.	7244.
4.5	516	25.35	.701	144.7	51.8	.222	2127.	761.
5.0	464	18.52	.146	22.1	7.9	.066	626.	224.
5.5	422	13.66	1.394	155.2	55.5	.460	4403.	1574.
6.0	387	5.89	.376	18.0	6.4	.053	512.	183.
6.5	357	4.66	1.394	52.9	18.9	.157	1500.	536.
7.0	331	3.83	.146	4.6	1.6	.014	129.	46.
7.5	309	3.16	.701	18.0	6.5	.054	512.	183.
8.0	290	2.61	5.285	112.3	40.2	.333	3186.	1139.
8.5	273	2.35	.701	13.4	4.8	.040	380.	136.
9.0	258	2.05	.146	2.4	.9	.007	69.	25.
9.5	244	1.56	1.394	17.8	6.4	.053	504.	180.
10.0	232	1.29	.376	4.0	1.4	.012	112.	40.
10.5	221	1.16	1.394	13.2	4.7	.039	374.	134.
11.0	211	1.03	.146	1.2	.4	.004	35.	12.
11.5	202	.90	.701	5.1	1.8	.015	145.	52.
12.0	193	.79	5.285	34.1	12.2	.101	969.	346.

MODE 2
OMEGA SQUARED IN (RADIANS/SECOND)**2 = .34089903
NATURAL FREQUENCY IN V.P.M. = 5575.51

NO.	INERTIA	THETA	IOMST	SIGMA M	SHAFT K	DTHETA
1	6.8	1.00000	2.320	2.320	58.1	.03993
2	49.2	.96009	16.110	18.430	84.7	.21751
3	47.9	.74257	12.131	30.561	84.7	.36070
4	47.9	.38187	6.238	36.799	84.7	.43433
5	47.9	-.05246	-.857	35.942	84.7	.42421
6	47.9	-.47667	-7.787	28.155	84.7	.33230
7	47.9	-.80897	-13.216	14.939	84.7	.17632
8	47.9	-.98530	-16.096	-1.157	84.7	-.01366
9	50.1	-.97164	-16.611	-17.768	76.9	-.23033
10	1100.1	-.74071	-277.771	-295.539	276.8	-1.06760
11	2650.4	.32709	295.538	.000		

MODE 2
OMEGA SQUARED .3409
NATURAL FREQUENCY 5575.5100
SIGMA I*THETA**2 2578.9463
SIGMA I*THETA**2 13306.1296
T INT 22715.43
T EXT 76963.28
STRESSED DIAMETER OF EXTERNAL SHAFT 16.00
EQUILIBRIUM AMPLITUDE .000030103857
F IN 7487.33
F INT .66
F EXT 2.32
F E 110478540.
F D 0.
F CA 0.
F CS 0.
F P 0.

ORDER	RPM	TN	VED	TSTINT	TSTEXT	PHI	TMAXI	TMAXE
.5	11151	172.07	1.508	177.4	601.2	.279	6331.	21451.
1.0	5575	101.93	.253	17.6	59.7	.046	1036.	3509.
1.5	3717	141.99	3.789	367.9	1246.5	.681	15460.	52381.
2.0	2787	36.44	.702	17.5	59.3	.198	4493.	15224.
2.5	2230	77.57	3.789	201.0	681.0	.460	10896.	36919.
3.0	1858	18.96	.253	3.3	11.1	.012	266.	903.
3.5	1593	45.76	1.508	47.2	159.9	.134	3052.	10341.
4.0	1393	29.32	1.211	24.3	82.2	.078	1771.	5999.
4.5	1239	25.35	1.508	26.1	88.6	.077	1747.	5919.
5.0	1115	18.52	.253	3.2	10.8	.009	216.	731.
5.5	1013	13.66	3.789	35.4	119.9	.108	2455.	8316.
6.0	929	12.25	.702	5.9	19.9	.016	365.	1235.
6.5	857	9.74	3.789	25.2	85.5	.068	1556.	5271.
7.0	796	8.18	.253	1.4	4.8	.004	85.	289.
7.5	743	6.62	1.508	6.8	23.1	.019	430.	1434.
8.0	696	5.35	1.211	4.4	15.0	.012	278.	943.
8.5	655	5.09	1.508	5.3	17.8	.014	312.	1058.
9.0	619	4.38	.253	.8	2.6	.002	46.	155.
9.5	586	2.73	3.789	7.1	24.0	.023	523.	1771.
10.0	557	1.96	.702	.9	3.2	.004	80.	271.
10.5	531	1.77	3.789	4.6	15.5	.017	388.	1316.
11.0	506	1.48	.253	.3	.9	.001	23.	78.
11.5	484	1.18	1.508	1.2	4.1	.005	119.	404.
12.0	464	.99	1.211	.8	2.8	.005	106.	358.

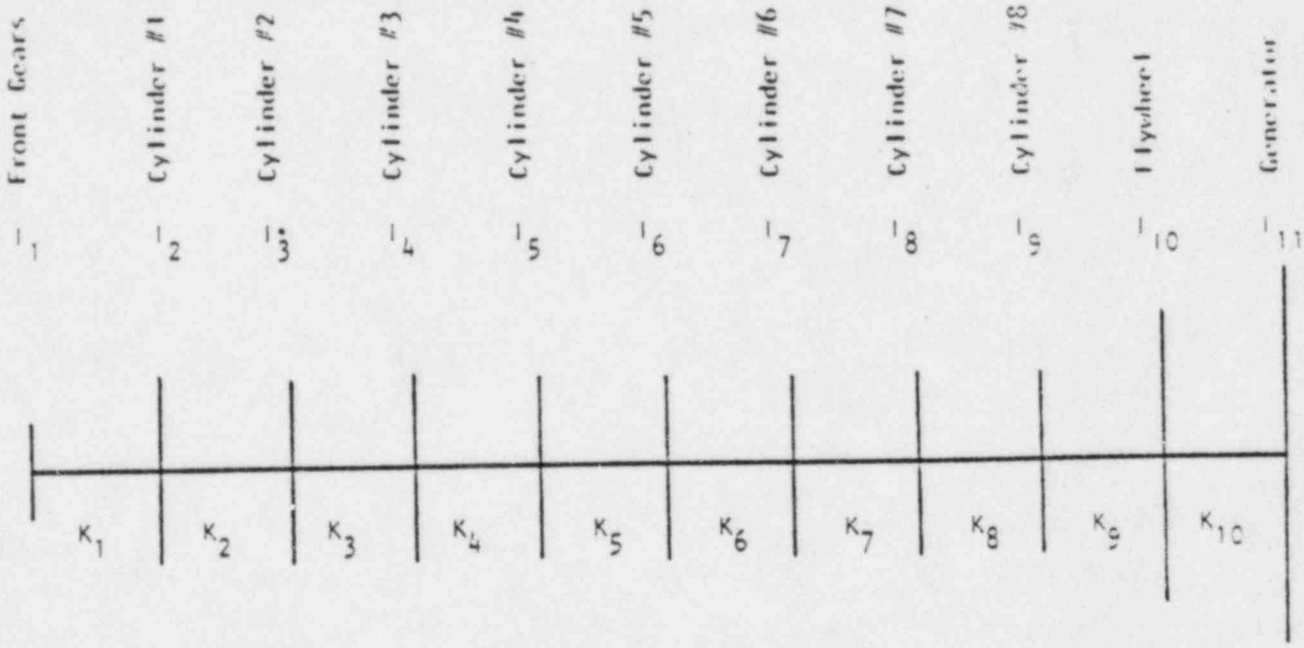
MODE 3
OMEGA SQUARED IN (RADIANS/SECOND)**2 = .53738424
NATURAL FREQUENCY IN V.P.M. = 7000.26

NO.	INERTIA	THETA	IOMBT	SIGMA M	SHAFT K	IT-ETA
1	6.6	1.00000	3.657	3.657	58.1	.06232
2	49.2	.93708	24.787	28.444	84.7	.22571
3	47.9	.60137	15.487	43.931	84.7	.51850
4	47.9	.08288	2.134	46.065	84.7	.54369
5	47.9	-.46081	-11.867	34.198	84.7	.40362
6	47.9	-.86443	-22.261	11.936	84.7	.14086
7	47.9	-1.00531	-25.889	-13.953	84.7	-.16488
8	47.9	-.84063	-21.646	-35.601	84.7	-.42219
9	50.1	-.42044	-11.331	-46.932	76.9	-.60397
10	1100.1	.18953	112.041	65.109	276.8	.23524
11	2650.4	-.04571	-65.109	.000		

MODE 3
OMEGA SQUARED .5374
NATURAL FREQUENCY 7000.2600
SIGMA I*THETA**2 2375.2154
SIGMA I*THETA**2 2997.4770
T INT 28370.37
T EXT 16355.47
STRESSED DIAMETER OF EXTERNAL SHAFT 16.00
EQUILIBRIUM AMPLITUDE .000064773258
F IN 7487.33
F INT 2.46
F EXT 1.44
F E 160237571.
F D 0.
F CR 0.
F CS 0.
F P 0.

ORDER	RPM	TN	VEC	TSTINT	TSTEXT	PHI	TMAXI	TMAXE
.5	14000	172.07	.955	403.7	236.3	.122	3523.	2061.
1.0	7000	101.93	.857	214.4	125.5	.106	3084.	1805.
1.5	4666	141.99	3.103	1081.9	633.2	.364	11121.	6509.
2.0	3500	36.44	2.525	226.0	132.3	.490	14199.	8310.
2.5	2800	77.57	3.103	591.1	345.9	.271	7836.	4587.
3.0	2333	18.96	.857	39.9	23.3	.027	793.	464.
3.5	2000	45.76	.955	107.4	62.8	.059	1699.	994.
4.0	1750	29.32	1.970	141.9	83.0	.087	2531.	1482.
4.5	1555	25.35	.955	59.5	34.8	.034	972.	569.
5.0	1400	18.52	.857	39.0	22.8	.022	642.	376.
5.5	1272	13.66	3.103	104.1	60.9	.061	1766.	1034.
6.0	1166	12.25	2.525	75.9	44.4	.040	1152.	674.
6.5	1076	9.74	3.103	74.2	43.4	.039	1119.	655.
7.0	1000	8.18	.857	17.2	10.1	.009	254.	149.
7.5	933	6.62	.955	15.5	9.1	.008	234.	137.
8.0	875	5.35	1.970	25.9	15.2	.014	398.	233.
8.5	823	5.09	.955	12.0	7.0	.006	174.	102.
9.0	777	4.38	.857	9.2	5.4	.005	136.	80.
9.5	736	2.73	3.103	20.8	12.2	.013	376.	220.
10.0	700	1.96	2.525	12.2	7.1	.009	252.	146.
10.5	666	1.77	3.103	13.5	7.9	.010	279.	163.
11.0	636	1.48	.857	3.1	1.8	.002	68.	40.
11.5	608	1.18	.955	2.8	1.6	.002	66.	39.
12.0	583	.99	1.970	4.8	2.8	.004	121.	71.

The next page (p.17) is a Tabulation of Mass Elastic Data showing other DSR-48 engines of similar rating. Except for minor differences in flywheel and generator, the torsional characteristic of these units are very similar, as indicated by the similarities between the torsional natural frequencies. A listing of operating hours accumulated for several of these units are included in Section Four of this submittal.



Typical Torsional Mass Elastic System (DSR-48/Generator)

TABULATION OF MASS ELASTIC DATA OF DSR-48 ENGINES.

	LILCO 74010/ 74012	Kousheng Taiwan 75005/ 75008	Gulf States 74039/ 74042	U. of Texas 78029/ 78030	SMUD 81015/ 81016	Saudi Arabia ****
I_1	6.805	8.250	6.805	8.560	6.805	6.805
I_2	49.222	49.222	49.222	54.389	49.222	54.389
I_3	47.922	47.922	47.922	53.090	47.922	53.090
I_4	47.922	47.922	47.922	53.090	47.922	53.090
I_5	47.922	47.922	47.922	53.090	47.922	53.090
I_6	47.922	47.922	47.922	53.090	47.922	53.090
I_7	47.922	47.922	47.922	53.090	47.922	53.090
I_8	47.922	47.922	47.922	53.090	47.922	53.090
I_9	50.149	50.841	50.149	58.257	50.149	55.316
I_{10}	1100.520	1009.604	426.527	368.572	1280.633	1280.633
I_{11}	2650.432	2828.371	4976.067	2756.882	2650.430	2552.371
K_1	58.121	58.121	58.121	58.121	58.121	58.121
K_2	84.727	84.727	84.727	84.727	84.727	84.727
K_3	84.727	84.727	84.727	84.727	84.727	84.727
K_4	84.727	84.727	84.727	84.727	84.727	84.727
K_5	84.727	84.727	84.727	84.727	84.727	84.727
K_6	84.727	84.727	84.727	84.727	84.727	84.727
K_7	84.727	84.727	84.727	84.727	84.727	84.727
K_8	84.727	84.727	84.727	84.727	84.727	84.727
K_9	76.941	67.327	76.941	76.941	76.941	76.941
K_{10}	276.773	326.100	309.720	229.770	254.397	252.445
N_1	2323	2280	2277	2143	2317	2219
N_2	5576	5988	6421	3669	5146	5125
N_3	7000	7064	8792	6307	6913	6619
CTWT	-	-	-	1	-	1

Note:

I	Inertia values of mass elastic system (Lb. ft. sec.^{2})
K	Stiffness values of mass elastic system (Million ft. lb. per Radian)
CTWT	Counterweight used at each crankthrow
****	Saudi Arabia Installations are:
	Dhuba 76010 to 76014
	Oneiza 76026 to 76028
	Wadi 78044 to 78046
	Rafha 79002 to 79004
	Rabigh 80001 to 80003
N	Torsional Natural Frequencies of the first three modes (V.P.M.)

SECTION TWO

TORSIOGRAPH TESTS

TORSIOGRAPH TESTS

One of the three engine (S/N 74010/12) was torsiographed by both Failure Analysis Associates (FaAA) jointly ith Stone & Webster (SWEC) as well as by Transamerica Delaval (TDI). TDI only measured overall torsional amplitudes.

On page 19 are measured results by FaAA/SWEC and those by TDI. FaAA's measurement reflects actual peak to peak amplitudes of the overall composite waveform. FaAA also provided the Square Root of the Sum of the Squares (SRSS), which would be comparable to the overall amplitudes measured by TDI using the Bell & Howell C.E.C. Vibration Meter. Also included in page 19 are test results of other DSR-48 engine generator sets with similar mass elastic system and of identical rating. It can be readily observed that the Fourth Order amplitudes measured by FaAA/SWEC on the subject crankshaft are similar to values measured by TDI on similar DSR-48 engine generators. Also the FaAA/SWEC SRSS amplitudes and TDI measured overall amplitudes are very similar. It may also be noted here that the Bell & Howell C.E.C. instrumentation used by TDI is widely used in the trade and although the measurement do not represent a theoretically correct peak to peak amplitude, it does however provide a common method in measuring the combined effects of the various harmonics.

On page 20, reference is made to the approval of crankshaft drawing 03-310-05-AC, as well as the corresponding ABS forging report number and physical properties. For comparison of the actual crankshaft minimum tensile property between LILCO, Kousheng and Rafha, we have included the values for the latter two.

On page 21, allowable stress levels due to single harmonic and overall are calculated using 1982 and current 1984 ABS Rules.

TORSIONAL TEST DATA ON DSR-48 ENGINE GENERATOR SETS.

A. Torsiograph Test Results by FaAA/SWEC

	3500 KW		3800 KW	
Fourth Order	.3[illegible]°	3106 psi	.339°	3242 psi
Overall, Pk to Pk	.693°	6626 psi	.719°	6875 psi
Overall, Calculated by SRSS	.424°	4054 psi	.454°	4341 psi

B. Torsiograph Test Results by TDI, Using Bell & Howell C.E.C. Model 1-117-0001 Vibration Meter.

	3500 KW		3800 KW	
Overall	.425°	4064 psi	.450°	4302 psi

C. Torsiograph Test Results by TDI, Using Bell & Howell C.E.C. Model 1-117-0001 Vibration Meter, on other DSR-48 Engine Generators.

S/N 76014	3500 KW		3850 KW	
Fourth Order	.350°	3365 psi	.370°	3557 psi
Overall	.420°	4038 psi	.460°	4423 psi
S/N 74039				
Fourth Order	.362°	3385 psi	.372°	3478 psi
Overall	.450°	4208 psi	.480°	4488 psi

PHYSICAL PROPERTIES OF CRANKSHAFTS

ABS REPORT ON CASTINGS OR FORGINGS REPORT NO.	LOWER VALUE YIELD POINT - PSI	LOWER VALUE TENSILE STRENGTH - PSI
LILCO		
83-ES 85280-1031	58291	100777
83-ES 85279-1031	57276	101792
83-ES 86290-365	48576	100777
RAFHA, 79004		
79-K078165-232	50500	93700
KOUSHENG, 75006/08		
	55000	93000
	54000	93500
	57000	98000

Crankshaft material specified is ABS Grade 4 which is A668 - Class E. Minimum yield point is 43000 psi and minimum tensile strength is 83000 psi.

The crankshafts for the DSR-48 carry a part or drawing number 03-310-05-AC, and has ABS approval stamped, dated 2-26-1976.

ALLOWABLE TORSIONAL STRESS CALCULATION.

Based on Para. 34.47 of 1984 ABS Rules.

$$S = \left(\frac{U + 23180}{18} \right) C_k \quad C_d \quad C_r$$

where U = Minimum Tensile Strength of Shaft Material 100000 PSI

C_k is .55 for propeller shafts and crankshafts

C_d is size factor, $.35 + 0.487 / \sqrt[5]{12} = .6463$

C_r is speed ratio factor, 1.38 for 90% to 105% rated RPM.

$$S = \left(\frac{100000 + 23180}{18} \right) (.55)(.6463)(1.38)$$

=3357 PSI due to single order

Total Allowable Stress = 150% of 3357 = 5035 PSI

ALLOWABLE TORSIONAL STRESS CALCULATION.

Based on Table 34.3 of 1982 ABS Rules.

Engine Speed	.3 × 450 RPM	.8 × 450 RPM	.95 to 1.0 × 450RPM	1.05 × 450RPM
	= 135 RPM	= 360 RPM	427.5 to 450	472.5 RPM
Grade 2, 60000 psi	5689 psi	3556 psi	2134 psi	3556 psi
Grade 4, 100000 psi	8217 psi	5136 psi	3082 psi	5136 psi

$$\text{Stress limit multiplier} = \frac{2}{3} \left(\frac{100000 - 60000}{60000} \right) + 1 = 1.4444$$

for adjustment from 60000 psi to 100000 psi material.

SECTION THREE

STRAIN GAUGE TESTS

Submittal to the American Bureau of Shipping
for the DSR-48 13-Inch by 12-Inch Replacement Crankshafts

Fatigue Analysis of the Replacement Crankshafts

The factor of safety against fatigue failure in the replacement (12-inch crank pins) 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 crank pins) 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.

Stresses in Replacement Crankshafts

The replacement crankshaft was instrumented with strain gages in the fillet locations of crank pins 5 and 7 and tested under operational conditions at both 3500 kW (100% rated load) and 3800 kW (109% rated load), 450 RPM synchronous speed. The highest stresses were measured in crank pin 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 crank pin. The following strains were measured at 3500 kW:

Strain Gage		Maximum	Minimum
5-1	(compression)	-195 με	288 με
5-2	(bending)	695 με	-410 με
5-3	(tension)	737 με	-610 με

PRJ:S-03310A-d/slw 3/27/84

To account for the simultaneous effects of shear and bending, the stress state is represented by equivalent stresses using Sine's method [1]. 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 [2], the equivalent alternating stress, S_{qa}, and equivalent mean stress, S_{qm}, on crank pin 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.

Endurance Limit for Failed Crankshaft

The failed crankshaft was instrumented with strain gages in the fillet location of crank pin 5. This fillet on the failed crankshaft had previously experienced a fatigue crack during performance testing. After the test, the three-dimensional finite element models of a quarter crank throw showed that the strain gage location on the failed crankshaft 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 percent higher at a nearby location. From the test report [3], the following strains were measured at 3500 kW:

Strain Gage	At Maximum Torque	At Minimum Torque
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 ksi
S_{qm} = 10.9 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 [4] in Figure 3.1.

This line is bounded by two lines showing the endurance limit for full scale crankshafts based on other test data [5].

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 crankshaft is shown in Figure 3.1.

The fillet regions of the replacement crankshafts have been shot peened. The effect of shot peening may produce widely differing increases in fatigue endurance limit; however, a conservative minimal value of this increase is 20% [6]. The endurance limit for the replacement crankshafts, including the effect of shot peening, is shown in Figure 3.1.

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, and is 1.75 when the effect of shot peening is assumed to increase the endurance limit by 20%.

At 3800 kW, the strain gage test data [1] 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.

PRJ:S-03310A-d/slw 3/27/84

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PRJ:S-03310A-d/slw 3/27/84

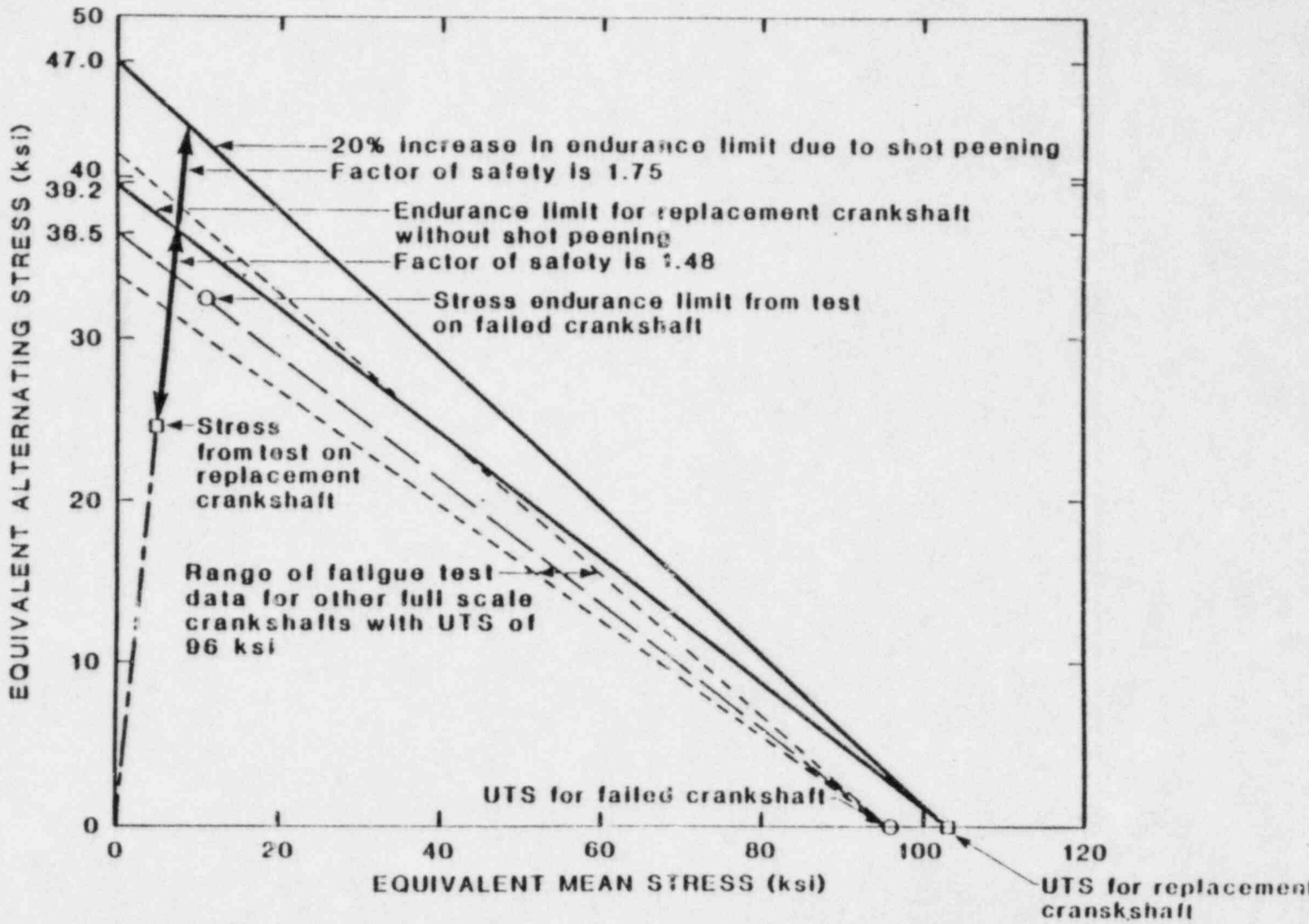

Figure 3-1. Goodman diagram for replacement crankshafts.

SECTION FOUR

OPERATING HOURS LOGGED

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	AV. LOAD REPORTED	OTHER LOADS & HOURS REPORTED
74010	LILCO, Shoreham	3500	368	4-01-84	--	3500 KW & Above 114 Hrs.
74011			430	4-01-84		3500 KW & Above 116 Hrs.
74012			345	4-01-84		3500 KW & Above 110 Hrs.
75005	KOUSHENG, TAIWAN	3600	246	3-15-84	Mostly 100%	--
75006			221	3-15-84		
75007			368	3-15-84		
75008			299	3-15-84		
76010	DHUBA, SAUDI ARABIA	3500	19800	3-17-84	--	--
76011			23300	3-17-84		
76012			23800	3-17-84		
76013			19700	3-17-84		
76014			23500	3-17-84		
76026	ONEIZA, SAUDI ARABIA	3515	16204	3-17-84	--	3000/3200 KW for 9000 Hrs.
76027			12428	3-17-84		
76028			14978	3-17-84		
78029	U. OF TEXAS	3500	8180	3-15-84	1100 KW	--
78030			5385	3-01-84	1100 KW	--
78044	WADI DAWASIR, SAUDI ARABIA	3515	10882	3-17-84	2200/3000 KW	--
78045			10832	3-17-84	2200/3000 KW	--
78046			11212	3-17-84	2200/3000 KW	--
79002	RAFI SAUDI ARABIA	3515	12667	3-16-84	--	3300 KW for 6200 Hrs.
79003			11655	3-16-84	--	3200 KW for 8250 Hrs.
79004			13186	3-16-84	--	3200 KW for 5500 Hrs.
80001	RABIGH, SAUDI ARABIA	3515	10196	3-16-84	2700 KW	--
80002			10245	3-16-84	2800 KW	--
80003			11602	3-16-84	2800 KW	--