



1 UNITED STATES OF AMERICA  
 2 NUCLEAR REGULATORY COMMISSION  
 3 Before the Atomic Safety and Licensing Board

4 -----x  
 5 In the Matter of: :  
 6 LONG ISLAND LIGHTING COMPANY : Docket No.  
 7 (Shoreham Nuclear Power Station, : 50-332-OL-3  
 8 Unit 1) : (Low  
 9 : Power)  
 10 -----x

11 919 Third Avenue  
 12 New York, New York  
 13 Wednesday  
 14 July 18, 1984

15 DEPOSITION of RICHARD WCYTONICH, HOWARD  
 16 C. ELAMING and ROBERT A. GIUFFERA, on behalf of the  
 17 American Bureau of Shipping, called for examination by  
 18 counsel for Suffolk County in the above-entitled  
 19 action, pursuant to subpoena, the witnesses having been  
 20 duly sworn by NICHOLAS J. TORRE, a shorthand reporter  
 21 and notary public for the State of New York, at the  
 22 offices of Paul, Weiss, Rifkind, Wharton & Garrison,  
 919 Third Avenue, New York, New York, at 10:45 a.m.,  
 the proceedings being taken down by Stenotype by  
 Nicholas J. Torre, and transcribed under his direction.

TANXCO REPORTING COMPANY, INC.  
 223 Jericho Turnpike  
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ALDERSON REPORTING COMPANY, INC.  
 20 F ST., N.W., WASHINGTON, D.C. 20001 (202) 628-9300  
 TRANSCRIPTION

C-43-1

1 2, were above 100 percent load?

2 WITNESS WOYTCWICH: No, I don't believe  
3 they did.

4 WITNESS GIUFFRA: Can you rephrase the  
5 question or repeat the question?

6 Q On page 28 of Exhibit No. 2, it lists  
7 operating hours at 100 percent load and above. It  
8 doesn't break down into how many hours for 100 percent  
9 load and how many hours above 100.

10 Did TLI give you any idea or LILCO or  
11 Failure Analysis Associates, give you any idea as to  
12 how many hours of operation occurred at above 100  
13 percent load?

14 WITNESS GIUFFRA: I don't believe they  
15 did.

16 WITNESS WOYTCWICH: No.

17 Q Then how do you know, sir, whether it  
18 was one hour of overload, of the 116 or 114 or 110, or  
19 whether it is all of those hours at 110 percent load?

20 WITNESS GIUFFRA: In the other columns,  
21 further over, there is an indication, kilowatt rating,  
22 450 rpm, total hours logged, with three numbers

1 indicated there, for three different engines.

2 Then you would go over to the last  
3 column, other loads and hours reported, 3500 and above.

4 Our assumption was that you would  
5 subtract the number indicated in the last column from  
6 the first one. That would represent--that way you  
7 would find out how many were done above.

8 In actual fact, 114, 116, 110, were done  
9 at above 3500 kw.

10 Q Are you familiar with or do you know,  
11 sir, of the service experience of the crankshaft that  
12 was in these three particular engines at LILCO, before  
13 this replacement crankshaft was installed?

14 WITNESS GIUFFRA: I guess I don't,  
15 except from what I read in the newspapers.

16 Q What do you know?

17 WITNESS GIUFFRA: I understand they  
18 found a crack in them. That is about the extent of my  
19 knowledge.

20 Q Do you have any knowledge?

21 WITNESS WOYTCWICH: Only what I read in  
22 the newspapers. We did not discuss the earlier

1 Exhibit No. 2 or Exhibit B--

2 Q Two.

3 WITNESS WOYTCWICH: Two. They do  
4 indicate that the crankshafts were shot peened. They  
5 do indicate that their experience shows that shot  
6 peening contributes to the fatigue strength of the  
7 material.

8 We accepted those statements which they  
9 made.

10 Q Do you know if Transamerica Delaval was  
11 responsible for writing that section of the report that  
12 you are referring to, that should be Section 3,  
13 strain-gauge matters.

14 WITNESS WOYTCWICH: Since it is all  
15 under Roland Yang's signature on the front sheet, and I  
16 don't see any other signatures, I assume that it was  
17 done by Transamerica Delaval.

18 Q But you don't know that for sure, you  
19 are assuming?

20 WITNESS WOYTCWICH: No, we didn't see  
21 them write it.

22 WITNESS GIUFFRA: When something is

1 shot peening, on a crankshaft, would it be important to  
2 you to know whether the shot peening was done correctly  
3 or properly?

4 WITNESS WOYTCWICH: If the crankshaft  
5 were being installed in a classed vessel, the shot  
6 peening would presumably be done under the cognizance  
7 of our attending surveyors.

8 Their judgment would come into play in  
9 that area. So that, if it were being done on a classed  
10 vessel, yes, we would want to know at least whether the  
11 surveyor had any objection to the process being applied.

12 Q And in considering a crankshaft not to  
13 be installed in a classed vessel--

14 WITNESS WOYTCWICH: In this case we  
15 could only review what was presented to us. We didn't  
16 make judgments on information not presented to us.

17 Q so that, is it correct to say that you  
18 assumed it was properly done in this case?

19 WITNESS WOYTCWICH: Yes.

20 Q Are there any adverse effects to shot  
21 peening?

22 MR. STIRCUFF: Objection to the form of

1 the question.

2 Q Of a crankshaft?

3 WITNESS BLANDING: If you assume that  
4 the shot peening is carried out correctly, you mount a  
5 strip outside and that is meant to curl up a certain  
6 amount, if that is done correctly, there presumably  
7 would be no adverse effects.

8 I believe that is what the record shows.

9 If you continue it too long or if you  
10 don't do it correctly, then you certainly could cause  
11 some problems.

12 Q What type of problems?

13 WITNESS BLANDING: Well, you could  
14 create some distortion of the metal surface.

15 Q What effect might that have?

16 WITNESS BLANDING: Might increase stress  
17 concentration factors. It would have metallurgical  
18 effects.

19 I would not be prepared to comment on  
20 that. That is a metallurgical problem. There is an  
21 accepted technique for doing shot peening, and accepted  
22 criteria as well, for the results of the work.



1 application or with the submission, the particulars  
2 that are listed in paragraph 34.3.1?

3 WITNESS GIUFFRÀ: Just at that time.  
4 Over the years, that information would be in our files.

5 Q Do you know the value that Transamerica  
6 Delaval conveyed to you as to the maximum firing  
7 pressure of this engine?

8 WITNESS WOYTCHWICH: That is the first  
9 page of Attachment 5. We have the pertinent page from  
10 our approval record log.

11 And that indicates a maximum firing  
12 pressure of 1700 psi.

13 Q Is that for operation at full load?

14 WITNESS WOYTCHWICH: For operation at the  
15 rating shown there.

16 You would have to convert the horsepower  
17 to kilowatts to see what that involved.

18 Q Immediately to the left of that, sir,  
19 the first page of Exhibit 5, there is a reference to  
20 1600 psi. To what does that refer?

21 A At the time the drawings were submitted,  
22 they asked for both ratings to be approved for whatever

1 WITNESS WOYTCWICH: Torsiograph  
2 measurements.

3 Q All right.

4 WITNESS WOYTCWICH: But those were, in a  
5 sense, supported by the calculations.

6 Q Didn't both the calculations and the  
7 torsiograph measurements indicate that ABS allowable  
8 stress limits were possibly exceeded?

9 WITNESS WOYTCWICH: Yes.

10 Now, to address --

11 WITNESS GIUFFRA: I think they indicated  
12 that they exceeded stress values which were indicated  
13 in our tables. I want to add, though, that there are  
14 provisions to consider values higher in the notes to  
15 those tables.

16 Q We have gone through that.

17 WITNESS GIUFFRA: Fine. From your  
18 questioning, that wasn't too clear.

19 WITNESS WOYTCWICH: To further discuss  
20 the matter of the stress values and our acceptance, look  
21 at handwritten notes, four, five and six of six, of  
22 Exhibit No. 3.



1 Q One moment.

2 All right.

3 WITNESS WOYTCWICH: In these  
4 calculations, we considered the submitted values and  
5 values which we found in our check process, using a  
6 paper by F. F. Porter, which is a longstanding  
7 reference document in torsional vibration analysis,  
8 published in the 1940's.

9 We obtained t sub n values from that  
10 paper.

11 So that, we did have some verification  
12 of the t-n values, but those are of secondary  
13 importance, because we believe that the torsigraph  
14 test measurements are the primary indicator of what the  
15 stresses -- what the vibratory stresses in the  
16 crankshaft would be, rather than a theoretical  
17 prediction.

18 Q How did the TDI submission of t-n values  
19 compare with the values in Porter?

20 WITNESS WOYTCWICH: We didn't compare  
21 them directly. We only looked at the overall results,  
22 which were comparable.

1           Q       Is it correct that you relied on, for  
2 your May 3rd letter, strain-gauge test measurements  
3 that TII provided?

4                   WITNESS WOYTCWICH: We relied on that to  
5 form part of our judgment, yes.

6           Q       Did you rely on the fact that the  
7 crankshafts were shot peened for that same letter?

8                   WITNESS WOYTCWICH: We relied on their  
9 statements in that regard.

10          Q       Did you rely on the service experience  
11 that TII submitted to you?

12                  WITNESS WOYTCWICH: Yes.

13          Q       Anything else that you relied on in  
14 reaching your statement of no objection?

15                  WITNESS WOYTCWICH: One other aspect  
16 that wasn't considered yet, and since we will need to  
17 explain it sooner or later, might as well look at that  
18 now.

19                    Sheets four, five and six, same three,  
20 deal with another aspect of our rules which, believe it  
21 or not, you haven't brought up.

22          Q       I believe it.

1 conclusions on.

2 Q Did you rely on the TDI submittal for  
3 your information as to the strain-gauge tests?

4 WITNESS WOYTOWICH: Yes. We had no  
5 other source of that information.

6 Q Did you perform any independent  
7 verification of that?

8 WITNESS WOYTOWICH: Not experimentally;  
9 only in the fact that we did an independent fatigue  
10 analysis to compare theoretical results against  
11 strain-gauge measured results.

12 Q Did you independently verify any service  
13 experience submitted by TDI?

14 WITNESS WOYTOWICH: No.

15 Q Did you independently verify anything  
16 about the shot peening performed on the crankshafts?

17 WITNESS WOYTOWICH: No.

18 Q If it were shown to you that any of this  
19 information you relied on in reaching your conclusion  
20 in the May 3rd letter was incomplete, incorrect or  
21 misleading, would you have to reconsider your  
22 conclusion in that letter?

1 MR. STRCUPE: Objection.

2 WITNESS BLANDING: I think the answer is  
3 obvious. Yes.

4 MR. SCHEIDT: A short break.

5 (Recess.)

6 MR. SCHEIDT: The May 3rd letter  
7 identified as Exhibit 4, refers to torsional vibrations  
8 at five and a half order critical speed.

9 Would operations of these engines at  
10 that speed be deleterious?

11 WITNESS WCYTCWICH: We have indicated  
12 that we would have no objection to operation at that  
13 speed. If we had an objection to such operation, we  
14 would have indicated that consideration should be given  
15 to the establishment of a hard speed range.

16 Based on the submitted information, and  
17 having come to the assessment of the safety factors and  
18 service experience with comparable engines, our  
19 conclusion was that on that basis, then, we would have  
20 no objection to such operation.

21 As to whether or not it is deleterious,  
22 operation of any engine at any speed is somewhat



ABS Check Cales

EXH- 3.  
7/18/14

MASYS	MASYS	USERID	ORIGIN
MACHINERY	ABSNYA	DISTCODE	SYSTEM
XEROX	OUTPUT2	FILENAME	FILETYPE
04/09/84	16:47:14	FILE CREATION DATE	
9710	00000113	SPOOLID	COUNT
04/09/84	17:09:26	FILE PRINT DATE	
H	406	CLASS	DEVICE
STANDARD		FORMS	

The diagram shows a 10x10 grid of squares. The top and bottom rows are fully filled with black dots. The middle rows contain scattered black dots, representing occupied sites in a lattice. The grid is enclosed by a solid line on the top and bottom, and dashed lines on the left and right, indicating periodic boundary conditions.

[illegible]

The figure shows a 10x10 grid of dots. The dots are arranged in a regular pattern, with some dots missing to form a shape resembling a large 'X' or a cross. The grid is used to illustrate the concept of a 100-dot array and how it can be partitioned into groups of 10 and 100.

Natural Frequency curve  
Shoeborn Plant  
New Cambridge

C-43-14



FILE: FORM

XV06

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ABS COMPUTERS, INC.

PAGE 001

C-43-15

FILE: LILCOGEN BASDATA A

ABS COMPUTERS, INC.

BRANCHED HOLZER TABLE INPUT DATA FOR :

LILCO Shoreham Plant - Emergency Diesel Gen - DeLaval/Enterprise DSR-48

# OF BRANCHES-

1

# OF MASSES IN EACH BRANCH (ONE ENTRY PER LINE)-

11

BRANCH#, MASS#, INERTIA, STIFFNESS

(FOUR ENTRIES PER LINE- SEPARATE WITH COMMAS):

1	,	1	,	81.66	,	6.97452E+08
1	,	2	,	590.664	,	1.0167E+09
1	,	3	,	575.064	,	1.0167E+09
1	,	4	,	575.064	,	1.0167E+09
1	,	5	,	575.064	,	1.0167E+09
1	,	6	,	575.064	,	1.0167E+09
1	,	7	,	575.064	,	1.0167E+09
1	,	8	,	575.064	,	1.0167E+09
1	,	9	,	601.788	,	9.233E+08
1	,	10	,	13200.6	,	3.3213E+09
1	,	11	,	31805.2	,	NONEEXISTENT

FILE: FORM

NUFRONT \*

ABS COMPUTERS, INC.

C-43-17

## HOLZER TABLE FOR 1 BRANCHES

FOR:

LILCO Shoreham Plant - Emergency Diesel Gen - DeLaval/Enterprise DSR-48

BRANCH# 1 (MAIN TRUNK)

BRANCH#	MASS#	INERTIA	STIFFNESS
1	1	81.66	6.97452E+08
1	2	590.664	1.0167E+09
1	3	575.064	1.0167E+09
1	4	575.064	1.0167E+09
1	5	575.064	1.0167E+09
1	6	575.064	1.0167E+09
1	7	575.064	1.0167E+09
1	8	575.064	1.0167E+09
1	9	601.788	9.233E+08
1	10	13200.6	3.3213E+09
1	11	31805.2	

FOR OMEGA 1  
 FREQUENCY STEP = 10  
 TOLERANCE = .001

OMEGA = 243.283

BRANCH#	MASS#	AMPLITUDE	TORQUE AFTER MASS
1	1	1	4.83317E+06
1	2	.99307	3.95502E+07
1	3	.95417	7.20264E+07
1	4	.883326	1.02091E+08
1	5	.782912	1.28738E+08
1	6	.656288	1.51076E+08
1	7	.507694	1.68356E+08
1	8	.342103	1.8E+08
1	9	.16506	1.85879E+08
1	10	-3.62596E-02	1.57549E+08
1	11	-8.36955E-02	-2414.43
			(REMAINDER TORQUE)

SUM OF I\*A SQUARED = 2708.68

OMEGA = 583.867

BRANCH#	MASS#	AMPLITUDE	TORQUE AFTER MASS
1	1	1	2.78379E+07
1	2	.960086	2.21159E+08
1	3	.74256	3.6673E+08
1	4	.381854	4.41588E+08
1	5	-5.24807E-02	4.313E+08

FILE: LILCODAT BASDATA A

ABS COMPUTERS, INC.

1	6	-.476696	3.37849E+08
1	7	-.808996	1.79254E+08
1	8	-.985305	-1.39052E+07
1	9	-.971628	-2.13235E+08
1	10	-.74068	-3.54637E+09
1	11	.327085	18588.5

(REMAINDER TORQUE)

SUM OF I\*A SQUARED = 13306.7

OMEGA = 733.061

BRANCH#	MASS#	AMPLITUDE	TORQUE AFTER MASS
1	1	1	4.38823E+07
1	2	.937062	3.41321E+08
1	3	.601367	5.2716E+08
1	4	8.28666E-02	5.52768E+08
1	5	-.460822	4.10362E+08
1	6	-.864443	1.43226E+08
1	7	-1.00532	-1.67444E+08
1	8	-.840623	-4.27218E+08
1	9	-.420422	-5.63177E+08
1	10	.18954	7.81365E+08
1	11	-.045719	-36823.

(REMAINDER TORQUE)

SUM OF I\*A SQUARED = 2998.74







# Check of Torsional Vibration

1700 rev/min

12 April 84

Sheet 1 of 6

Eng. C.W.

Hysteresis Damping -

DeLaval

$$\frac{\pi (\omega^2 \times 10^6) (\epsilon - \epsilon^2) \phi^2}{25}$$

We know

$$D.M. = \frac{\pi \omega^2 \epsilon I A^2}{\text{work ~~absorbed~~ function } (= F(\phi^2))}$$

or

$$\text{work ~~absorbed~~ function } (F(\phi^2)) = \frac{\pi \omega^2 \epsilon I A^2}{D.M.}$$

$$\text{Since work absorbed} = \frac{\pi \omega^2 \epsilon I A^2 \phi^2}{D.M.}$$

Therefore, DeLaval assumes dynamic magnifier of 25.

$$BMEP = 225.6 \text{ psi}$$

Sheet 2 of 3  
C43

For a 4-cycle engine, use Porter Q2 curves

$$\text{Order of interest} = 5 \frac{1}{2}$$

	<u>MIP, psi</u>	<u>Tn, psi</u>		
$\Delta = 20.3$ $\Delta = 20.5$ $\Delta = 79.6$	109.2	10.601	$\Delta = .598$ $\Delta = .575$ $\Delta = 2.1735$	
	129.5	11.199		
	150.0	11.774		
	225.6	calculated = 13.948	calculated = 2.1735	

$$\Theta_{eq} = \frac{T_n A R \epsilon \vec{a}}{\omega^2 \epsilon I A^2}$$

$$\epsilon \vec{a} = \text{vector sum} = 1.394 \text{ (from submitted calc.)}$$

$$A = \text{piston area} = \frac{\pi}{4} \times (17)^2 = 226.98$$

$$R = \text{crank radius} = \frac{1}{2} \times 21 = 10.5$$

$$\Theta_{eq} = \frac{(13.948)(226.98)(10.5)(1.394)}{(243.283)^2 (2708.68)} = 2.89 \times 10^{-4} \text{ rad} = 0.01656 \text{ deg}$$

$$\text{If } D.M. = 25, \Theta_{\text{resonance}} = 0.414 \text{ degrees}$$

$$\text{" " " } = 50, \text{ " " } = 0.828 \text{ "}$$

$$\text{Stress per degree} = \frac{EI\omega^2 A}{11.25 d^3}$$

max. stress  
Cyl.

Shaft diameter: front gear to cyl. #1 = 8"

crankpins to flywheel = 12"

generator shaft = 16"

Worst stresses for each size shaft:

front gear to cyl. #1 -  $\frac{\text{Stress}}{\text{deg}} = \frac{4.83317 \times 10^6}{11.25 (8)^3} = 839 \text{ psi/deg}$

crankpins to flywheel -  $\frac{\text{Stress}}{\text{deg}} = \frac{1.85879 \times 10^8}{11.25 (12)^3} = 9561.67 \text{ psi/deg}$   
(max. after mass #9)

generator shaft -  $\frac{\text{Stress}}{\text{deg}} = \frac{1.57544 \times 10^8}{11.25 (16)^3} = 3419.03 \text{ psi/deg}$

Worst stress is in engine.

Equilibrium amplitude = .01656 degrees

" stress = (.01656)(9561.67) = 158.34 psi

If D.M. = 25, stress = 3958.53 psi (vs. 4138 submitted)

If D.M. = 50, " " 7917.06 psi

Conclusion: Adequacy of analysis depends on dynamic magnifier

inset 4. f. c.  
 20.3

It also appears that off-resonance harmonics of 4th order will be important.

	<u>MIP, psi</u>	<u>Tn, psi</u>	
avg 20.3	109.2	19.945	$\Delta = 1.379$ $\Delta = 1.354$ avg 1.3665
$\Delta = 20.5$	129.5	21.324	
$\Delta = 75.6$	150	22.678	
	225.6		
		calculated $\Delta = 5.064$	
		calculated $T_n = 27.742$ psi	

Vector Sum = 5.285 (from submitted calc - confirmed by check calc - check result 5.2846)

$$\Theta_{eq} = \frac{(27.742)(226.48)(10.5)(5.285)}{(243.283)^2 (2708.68)} = 2.1796 \times 10^{-3} \text{ rad}$$

$$= 0.12489 \text{ deg}$$

$$\approx 0.125 \text{ deg}$$

$$\omega = 243.283 \text{ rad/sec} = 2323.2 \text{ VPM}$$

$$\text{Exciting frequency} = 450 \text{ RPM} \times 4^{\text{th}} \text{ order} = 1800 \text{ VPM}$$

$$r = \frac{\omega}{\omega_n} = .7748$$

$$DM = \frac{1}{1-r^2} = 2.5$$

$$\Theta_{res} = (2.5)(.125) = 0.31274 \text{ deg}$$

$$\text{At MCR, } S_{stress} = (.31274)(9561.67) = 2990 \text{ psi}$$

Resultant Stress:  
 4 order 2990 psi  
 5 1/2 order 3958 psi  
 resultant =  $\sqrt{S_{stress}^2}$   
 = 4961 psi

Critical speed for  $5\frac{1}{2}$  order is

$$4202525 = 2242524 \text{ RPM}$$

$$2323.2 \text{ VPM} \div 5.5 = 422.4 \text{ RPM, } \Rightarrow = 93$$

At this speed, ~~the~~ 4th order excitation frequency

$$4 \times 422.4 = 1689.6 \text{ VPM}$$

$$r = \frac{\omega}{\omega_n} = \frac{1689.6}{2323.2} = .7273$$

$$DM = \frac{1}{1-r^2} = 2.123$$

$$\theta_{res} = (2.123 \times .125) = .2654 \text{ deg.}$$

$$\text{Stress} = (.2654 \text{ deg.} \times 9581.67 \text{ }^{\circ}/\text{deg.}) = 2537 \text{ psi.}$$

Resultant stress (RMS sum)

$$= \sqrt{(2537)^2 + (3958)^2} = 4701.4 \text{ psi}$$

Overall amplitude should be  $\frac{4701.4}{9581.67} = 0.492 \text{ deg.}$

Stress -

Rule allowable (0.95R) = 2134 psi for 11.8 in. dia.  
1707 psi for 236 " "

= 2127 psi for 12 in. dia.

Material correction for ABS Grade 4 (UTS = 83000)

$$\text{Allowable} = 2127 \times \left( \frac{83000 + 30000}{80000} \right) = 2670.3 \text{ psi}$$

Allowable for all orders =  $1.5 \times 2670.3 = 4005 \text{ psi}$



2  
-

79% L.

uncy

35  
FEDS  
E  
JWA  
1  
05

Actual material is better than Grade 4 -  
min. UTS 100,700 psi

Sheet =  
Civ

$$\text{Single Harmonic Allowable} = 2127 \times \left( \frac{100,000 + 30000}{90000} \right) = 3072 \text{ psi}$$

$$\text{For all orders, Allowable} = 3072 \times 1.5 = 4608.5$$

Calculated stresses still exceed allowable.

$$\text{By 1984 Rules, allowable} = 3357 \text{ psi} \times 1.5 = 5035 \text{ psi} - \underline{\underline{OK}}$$

Torsigraph results:

<u>Calculated</u>	<u>Measured</u>
4th Order (pk/pk) (single pk) - .3127 degree	.325° - .339°
5 1/2 Order (pk/pk) (single pk) .414	not measured
Combined ( $\sqrt{\epsilon \text{ stress}^2}$ ), (single pk) .492	.424° - .454° (stress < 4341 psi)

Question - are measured results single-peak?

Per telcom 4/12 w/ Gene Montgomery, LILCO,  
all submitted measurements are single  
amplitude ( $\phi$ -peak or  $\phi$ -RAUS). Therefore  
agreement w/ calculation is good.

Per telcom 4/18 w/ Gene Montgomery, LILCO, cyl. exhaust  
temperatures are monitored - max. allowable diff. is 75°F.

C-413-28

16 April, 1984

WILCO  
Hull No. Shorham Plant  
MANUFACTURER Panameric Solar  
MODEL DSM/DSR-48

BY [signature]

# CRANKSHAFT SAFETY FACTOR BY CIMAC METHOD (SI UNITS ONLY)

			*
			- (READ SIDE A)
			38 R - (READ SIDE B)
		J	
			- CYL. BORE, mm
			- MAX. FIRING PRESSURE, N/mm <sup>2</sup>
			- EFFECTIVE MOMENT ARM, mm (from Prelim. Calc)
			- W (Web Thickness) } mm
			- B (Web Width) }
			◇ - $\sigma_{BN}$ (nominal bending stress, N/mm <sup>2</sup> )
		J	
			- D (crankpin dia.)
			- W (Web Thickness)
			- S (pin overlap)
			- B (Web Width)
			- R <sub>H</sub> (Crankpin Fillet Radius)
			- D <sub>JO</sub> (Journal Bore Dia.)
			- D <sub>CM</sub> (Crankpin Bore Dia.)
			◇ - $\alpha_B$ (bending stress concentration factor)
			◇ - $\alpha_T$ (torsional " " " " )
			- TH (Web undercut at crankpin)
			- TG ( " " " journal)
			◇ - F <sub>HN</sub> (Recess Factor)
			◇ - $\alpha_B \times F_{HN}$
		J	
			- $\sigma_{BN}$ , N/mm <sup>2</sup>
			- $\alpha_B \times F_{HN}$
			- $\sigma_{addition}$ , N/mm <sup>2</sup> (20 mds. max. calc. for Calc.)
			- $\tau_N$ (Nominal Torsional Vibration Stress) N/mm <sup>2</sup>
			- $\alpha_T$
			◇ - $\sigma_V$ (Reference Stress, N/mm <sup>2</sup> )
		J	
			- $\sigma_B$ (Crankshaft Ultimate Tensile Strength, N/mm <sup>2</sup> )
			- K (Manufacturing Factor)
			- R (Crankpin Fillet Radius, mm)
			- D (Crankpin Diameter, mm)
			◇ - $\sigma_{DW}$ (Fatigue Strength, N/mm <sup>2</sup> )
			- $\sigma_V$
			◇ - SAFETY FACTOR (FOR WEB IN WAY OF CRANKPIN FILLET)

1  
491.1  
11.721  
157.162  
175  
573.1  
45.5475

104.3  
127  
573.1  
17.46  
73.4  
73.8  
2.70439  
1.84445  
17.46  
21.11  
1.29796  
3.51911

45.5475  
3.51911  
0  
75.73  
1.84445

= 28,362 psi ← 195.46607

694.3  
1  
14.05  
304.0

33.24 ← 227.99697  
195.46607  
1.16643

MAY BE STARTED AT 1J, 2J, 3J, OR 4J,  
DEPENDS ON AVAILABLE DATA. ONCE THE PROGRAM IS  
STARTED, IT CAN BE FINISHED WITHOUT ANY FURTHER  
"J" COMMANDS.

HDS

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SHIPYARD  
HULL  
MANUFACTURER  
MODEL

SHEET 2.6  
BY ASD

# PRELIMINARY CALCULATIONS FOR CRANKSHAFT SAFETY FACTOR ANALYSIS

## FIND END MOMENT CONSTANT OR EFFECTIVE $M$

1J (web) OR 2J (pin)

S - BENDING STRESS (SUBMITTED VALUE)  $N/mm^2$  Divide by stress concentration

S - WEB THICKNESS OR PIN DIAMETER mm

S - WEB WIDTH (1J ONLY) mm

A/D - BENDING MOMENT

3J

S - BORE mm

S - MAX. FIRING PRESSURE  $N/mm^2$  (mult. by 1.2 for V engine)

A/D - FIRING FORCE

4J

S - BENDING MOMENT (FROM 1J OR 2J)

S - FIRING FORCE (FROM 3J)

S - DISTANCE TO CRITICAL SECTION (X) from main brg  $\phi$ S -  $\phi$  TO  $\phi$  DISTANCE BETWEEN MAIN BRGS. (L)

A/D - END MOMENT CONSTANT (FOR ABS METHOD) \*\*

A/D - EFFECTIVE MOMENT ARM (FOR CIMAC METHOD) \*

\*(APPLIES ONLY TO "X" VALUE ENTERED ABOVE)

\*\*(VALID FOR ALL "X" VALUES BETWEEN 0 AND L/2)

## CONVERSION OF CIMAC STRESS CONC. FACTOR

5J

S -  $\phi_B \times F_{HM}$  (CIMAC BENDING STRESS CONC. FACTOR)

S - PIN DIAMETER

S - WEB THICKNESS (W)

S - WEB WIDTH (B)

A/D -  $\phi_B$  (CORRECTED - BASED ON PIN - FOR ABS METHOD)

3.50911

304.0

125

533.4

7.07233

## NOTES:

IF STRESS AT A POINT IS GIVEN - START AT 1J IF KNOWN STRESS IS IN  
" " " 2J " " " " "

IF MOMENT AT A POINT IS GIVEN - " " 3J

IF MOMENT AND FIRING FORCE ARE KNOWN - " " 4J

PROGRAM CONTINUES THROUGH 4J ROUTINE WITH NO FURTHER COMMANDS.  
5J ROUTINE MUST BE CALLED OUT SEPARATELY

C-43-30

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UNIVERSITY  
HULL  
MANUFACTURER  
MODEL

By *AD*

# CRANKPIN SAFETY FACTOR

(STEADY + ALTERNATING LOAD)

Met in ENGLISH, OR SI UNITS

1  
31.1  
61.1  
214.3125  
11.721  
304.1  
26.2276  
21.93  
1.14 45  
26.2276  
7.2233  
644.3  
From → 277.94633  
C/MAC 7547.03  
= 33083.1  
3500  
450

13.34921  
34.31077  
206.34117  
19.94497  
1.10495  
1.04495  
3400  
450  
13.87487  
36.61925  
206.34117  
18.95399  
1.10495  
1.04410

1 J  
S - Bore  
S -  $\phi$  to  $\phi$  distance between main bearings  
S - end moment constant [1] (from prelim calc)  
S - Distance to critical section (from main brg  $\phi$ )  
S - Maximum Firing Pressure (adjusted for V-engine) <sup>to edge of crank</sup>  
2J  
S - Crankpin Diameter  
-A◇ Nominal bending stress (If program starts at 2J, program will print  
S - Estimated torsional vibration stress  
S - Torsional stress concentration factor <sup>from other sheet</sup>  
S - Nominal alternating or steady bending stress  
S - Bending stress concentration factor [3]  
S - Crankshaft Ultimate Tensile Strength  
S - Crankshaft Fatigue Limit  
S - proportionality constant [2] (see below)  
S - Rated Power  
S - Rated RPM

Based on LIL

A◇ - Steady Torsional Stress  
A◇ - Combined Steady Stress  
A◇ - Combined Alternating Stress  
A◇ - Steady Stress Safety Factor  
A◇ - Alternating Stress Safety Factor  
A◇ - Overall Safety Factor

S - Power  
S - RPM  
(This portion of program repeats)  
[1] - End Moment Constant  
0 = simple support (limiting value)  
0.67 = typical value  
2 = built-in ends (limiting value)  
[2] - Length Force Power Proportionality Constant  
inch pound IP(English) 63025  
mm kg IP(Metric) 716200  
mm N kW(SI) 9542900

MANDS.

C-43-31



Safety Factors - desired minimum = 1.34 - lowest  
for spec  
review of  
another  
mfg.

Fatigue Stress = 29940 psi

Fatigue Strength = 33083 psi (theoretical CIMAC)  $\rightarrow$  S.F. = 1.105

39200 psi (submitted - no shot peen)  $\rightarrow$  S.F. = 1.309

47000 psi (shot peened)  $\rightarrow$  S.F. = 1.570

UTS = 100,744 psi

Steady Stress = ~~5313.5~~ psi  $\rightarrow$  S.F. = ~~18.960~~ 18.960  
5313.5

Combined S.F. - Goodman method  $\left( \frac{1}{SF} = \frac{1}{SF_{fatig}} + \frac{1}{SF_{ult}} \right)$

For CIMAC <sup>(theoretical)</sup> fatigue limit  $\rightarrow$  1.044 (LOW)

For submitted F.L. - no shot peen  $\rightarrow$  1.224 (LOW/MARGINAL)

For submitted F.L. - w/shot ~~peening~~ <sup>peening</sup>  $\rightarrow$  1.450 (OK)

Combined S.F. - Elliptical method  $\left( \frac{1}{SF^2} = \frac{1}{SF_{fat}^2} + \frac{1}{SF_{ult}^2} \right)$

For CIMAC <sup>(theoretical)</sup> fatigue limit  $\rightarrow$  1.103 (LOW)

For submitted F.L. - no shot peen  $\rightarrow$  1.306 (OK) (but marginal)

For submitted F.L. - shot peened  $\rightarrow$  1.565 (OK)

NOTE: SEE FOLLOWING PAGES FOR FURTHER ~~INFO~~ INFO





## Conclusions

- 1) Based on submitted fatigue strength data and test results, proposed torsional vibration arrangements are OK.
- 2) Note that submitted fatigue strength is higher than that obtained by CIMAC formula.
- 3) Note also that submitted stress <sup>test</sup> results show ~~the~~ lower stresses than those calculated by either CIMAC or ABS formulas

Check crankshaft diameter by SAE

$$d = C \sqrt[3]{\frac{M + (M^2 + 4T^2)^{1/2}}{F}}$$

$$F = \frac{2598}{1.6} \text{ for } UTS = 100,700 \text{ psi } (70.8 \text{ kg/mm}^2)$$

$$L = 609.6 \text{ mm between brg. centers}$$

$$C = 1.00 \text{ (eight cylinders)}$$

$$M = 1.86 PD^2 L$$

$$P = 1.196 \text{ kg/mm}^2 = 119.6 \text{ kg/cm}^2 (\approx 1700 \text{ psi})$$

$$D = 431.8 \text{ mm}$$

$$L = 609.6 \text{ mm}$$

$$M = 2.5285 \times 10^{10}$$

$$T = 1.02 \times 10^6 \text{ H/R}$$

$$H = 5300 \text{ HP}, R = 450 \text{ RPM}$$

$$T = 1201^{\circ} .33$$

$$d = 1.0 \sqrt[3]{\frac{M + \sqrt{M^2 + 4T^2}}{2598}}$$

$$= 269 \text{ mm}$$

$$= 10.59 \text{ inches}$$

$$\begin{matrix} 1400 & 60,000 \\ 410 & \\ 2310 & 83000 \end{matrix} \left. \vphantom{\begin{matrix} 1400 \\ 410 \\ 2310 \end{matrix}} \right\} 23000$$

$$P = 5 \times (UTS + 60,000) + 1300$$

OK

