BEFORE THE ATOMIC SAFETY AND LICENSING BOARD

---oOo---

In the matter of
LONG ISLAND LIGHTING COMPANY,

(Shoreham Nuclear Power
Station, Unit 1)

DOCKET NO. 50-332-OL

COPY

DEPOSITION OF MAURICE H. LOWREY

May 10, 1984

REPORTED BY:
JOAN MARIE COLUMBINI, CSR NO. 5435

TOOKER & ANTZ
CERTIFIED SHORTHAND REPORTERS
631 MARKET STREET, SUITE 925
SAN FRANCISCO, CALIFORNIA 94105
415/392-0650

COMPUTERIZED TRANSCRIPT

A. Not that I can recall.

Q. Did the lengthening of the cylinder head bosses involve a thickening of the material of the cylinder block?

A. Not to my knowledge, except in that area where the boss lengthening occurred.

Q. In that particular area, what was the extent of the thickening, approximately?

A. Again, if memory serves me correctly, approximately five and a half inches.

Q. Did the thickening of the cylinder block in that area increase the weight of the block?

A. Yes, sir.

Q. By approximately how much?

A. I don't really know.

Q. Was the change in the design of the cylinder block that you have just referred to made in response to any problems experienced with the prior design block?

MR. SMITH: I believe the question has already been asked and answered. I think this witness testified that as a result of the experience with the RV5.

MR. DYNNER: Q. Can you answer the question?

A. Would you say again please, sir?

Q. Yes.

Was the change in the design of the cylinder block made at all in response to any problems experienced with the prior designed block?

A. It was my judgment that the deepening, if you

2

will, or lengthening of the head stud bosses would reduce the possibility of cracking noticed around the head stud bosses of the previous block.

Q. To what extent was cracking in the area of the head stud bosses in the prior design cylinder block experienced?

A. You will have to give me your meaning of "experience," please.

Q. Certainly.

Were you aware of problems in cracking in that area experienced by Delaval R4 engi

A. Yes, I was.

Q. Was that problem of crack documents that you saw?

A. In documents that I prepa

Q. How did you come to prepa

A. As a result of various fi made and records that I had made du and in the normal course of recordi

Q. In approximately how many engines in the field did you note cracking in that portion of the cylinder block?

MR. SMITH: What time period are we talking about?

MR. DYNNER: Whatever time period he is referring to with respect to the reports that he prepared. Do you understand the question?

THE WITNESS: Would you repeat?

AFTERNOON SESSION 1:35 P.M.

EXAMINATION BY MR. DYNNER (Resumed)

MR. DYNNER: Q. Mr. Lowrey, do you recall this morning we talked briefly about the crankshaft for the DSR-48 engine with the nominal 12 inch crankpin in which you participated in design.

Who prepared the detailed design drawing for that crankshaft?

A. I don't know.

Q. You mentioned that the replacement crankshaft for that design for the Shoreham engines was shot peened. Do you know who performed the shot peening operations?

A. No, sir, I don't.

Q. Do you know the procedures that were followed for the shot peening process?

A. Likewise. I don't know that.

Q. Did the replacement crankshaft design specify a different grade of material than its predecessor, that is the crankshaft with the 11 inch nominal crankpin?

A. I don't recall whether it did or not, Mr. Dynner.

Q. Was the replacement design crankshaft tested here at the Delaval factory before it was supplied to any customers?

A. No, sir, it was not, because we have never built a crankshaft that had been shot peened. So in the strict sense of the word, we have never operated a crankshaft that had been shot peened other than at

4

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

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In the Matter of:

LONG ISLAND LIGHTING COMPANY (Shoreham Nuclear Power Station, Unit 1) : Docket No. 50-332-OL-3 (Low Power)

---------------------------------------x

919 Third Avenue
New York, New York
Wednesday
July 18, 1984

DEPOSITION of RICHARD WOYTOWICH, HOWARD C. BLANDING and ROBERT A. GIUFFRA, on behalf of the American Bureau of Shipping, called for examination by counsel for Suffolk County in the above-entitled action, pursuant to subpoena, the witnesses having been duly sworn by NICHOLAS J. TORRE, a shorthand reporter and notary public for the State of New York, at the 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.

TANKOOS REPORTING COMPANY, INC.
223 Jericho Turnpike
Mineola, New York 11501
(212)343-0171 (516)741-5235

Dupe 8412170284

2, were above 100 percent load?

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

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

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

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

WITNESS GIUFFRA: I don't believe they did.

WITNESS WOYTCWICH: No.

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

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

2

indicated there, for three different engines.

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

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

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

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

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

Q What do you know?

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

Q Do you have any knowledge?

WITNESS WOYTOWICH: Only what I read in the newspapers. We did not discuss the earlier

3

Exhibit No. 2 or Exhibit B--

Q Two.

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

We accepted those statements which they made.

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

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

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

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

WITNESS GIUFFRA: When something is

4

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

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

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

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

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

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

WITNESS WOYTCWICH: Yes.

Q Are there any adverse effects to shot peening?

MR. STRCUBE: Objection to the form of

5

the question.

Q Of a crankshaft?

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

I believe that is what the record shows.

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

Q What type of problems?

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

Q What effect might that have?

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

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

6

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

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

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

WITNESS WOYTCWICH: That is the first
page of Attachment 5. We have the pertinent page from
our approval record log.

And that indicates a maximum firing
pressure of 1700 psi.

Q Is that for operation at full load?

WITNESS WOYTCWICH: For operation at the
rating shown there.

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

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

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

WITNESS WOYTCWICH: Torsiograph measurements.

Q All right.

WITNESS WOYTCWICH: But these were, in a sense, supported by the calculations.

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

WITNESS WOYTCWICH: Yes.

Now, to address --

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

Q We have gone through that.

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

WITNESS WOYTCWICH: To further discuss the matter of t sub n values and our acceptance, look at handwritten notes, four, five and six of six, of Exhibit No. 3.

8

Q One moment.

All right.

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

We obtained t sub n values from that paper.

So that, we did have some verification of the t-n values, but these are of secondary importance, because we believe that the torsiograph test measurements are the primary indicator of what the stresses -- what the vibratory stresses in the crankshaft would be, rather than a theoretical prediction.

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

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

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

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

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

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

Q Did you rely on the service experience that TDI submitted to you?

WITNESS WOYTCWICH: Yes.

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

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

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

Q I believe it.

10

conclusions on.

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

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

Q Did you perform any independent verification of that?

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

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

WITNESS WOYTOWICH: No.

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

WITNESS WOYTOWICH: No.

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

MR. STRCUPE: Objection.

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

MR. SCHEIDT: A short break.

(Recess.)

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

Would operations of these engines at that speed be deleterious?

WITNESS WOYTOWICH: We have indicated that we would have no objection to operation at that speed. If we had an objection to such operation, we would have indicated that consideration should be given to the establishment of a bard speed range.

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

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

12

III

ABS Check Calcs

Ext- 3.
7/18/14

** VM/SP :RVICE LEVEL 0203 **** CPU SERIAL 014054 CPU MODEL 43.

MASYS	MASYS	USERID	ORIGIN
MACHNERY	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	

ABS

ABS COMPUTERS

MASYS

MACHNERY

Natural Frequency calculations

Shoreham Plant

New Crankshaft

15

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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 ,NONEXISTENT
```

FILE: FORM NUFRONT * ABS COMPUTERS, INC.

17

FILE: LILCODAT BASDATA A ABS COMPUTERS, INC.

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

1	6	-.476696	3.37849E+08
1	7	-.808996	1.79254E+08
1	8	-.985305	-1.39052E+07
1	9	-.971626	-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

159-END

PRINT COMPLETED AT 17:09:42 FOR USER: MASYS   DIST: MACHINERY

END-159

20

** VM/SP SERVICE LEVEL 0203 **** CPU SERIAL 014054 CPU MODEL 4341

ZCPT7781	ZCPT7781	USERID	ORIGIN
=ZCPT1=	ABSNYA	DISTCODE	SYSTEM
XEROX	OUTPUT2	FILENAME	FILETYPE
04/09/84	16:56:30	FILE CREATION DATE	
9727	00002514	SPOOLID	COUNT
04/09/84	17:09:42	FILE PRINT DATE	
H	406	CLASS	DEVICE
STANDARD		FORMS	

160

ABS

ABS COMPUTERS

ZCPT7781

ZCPT1

Check of Torsional Vibration

Hysteresis Damping –

Delaval

$$\frac{\pi(\omega^2 \times 10^6)(\varepsilon I A^2)\,\phi^2}{25}$$

We know

$$D.M. = \frac{\pi\,\omega^2\,\varepsilon I A^2}{\text{work } \cancel{\text{[illegible]}}^{\text{function}} (= F(\phi^2))}$$

or

$$\text{work } \frac{\cancel{\text{[illegible]}}^{\text{function}}}{(F(\phi^2))} = \frac{\pi\,\omega^2\,\varepsilon I A^2}{DM}$$

~~Since~~ work absorbed = $\dfrac{\pi\,\omega^2\,\varepsilon I A^2\,\phi^2}{DM}$

Therefore, Delaval assumes dynamic magnifier of 25.

$BMEP = 225.6$ psi

Insert cont. Cont

For a 4-cycle engine, use Porter Q2 curves

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

MIP, psi	T_n, psi
109.2	10.601
129.5	11.199
150.0	11.774
225.6	calculated $= 13.948$

MIP: $\Delta = 20.3$, $\Delta = 20.5$ (avg $= 20.4$), $\Delta = 75.6$

T_n: $\Delta = .598$, $\Delta = .575$, avg $= .5865$; calculated $\Delta = 2.1735$

$$\theta_{eq} = \frac{T_n \, A \, R \, \Sigma \vec{a}}{\omega^2 \, \Sigma I A^2}$$

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

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

R = crank radius $= \frac{1}{2} \times 21$ (stroke) $= 10.5$

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

If D.M. $= 25$, $\theta_{resonance} = 0.414$ degrees
" " " $= 50$, " " $= 0.828$ "

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

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

crankpins to flywheel = 12"

generator shaft = 16"

Worst stresses for each size shaft:

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

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

generator shaft - $\frac{\text{Stress}}{\text{deg}} = \frac{1.57544\text{E}8}{11.25(16)^3} = 3419.03$ 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 magnific...

[illegible]

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

MIP, psi		Tn, psi	
109.2	avg 20.4; Δ=20.3	19.945	Δ= 1.379; avg 1.365
129.5	Δ=20.5	21.324	Δ= 1.354
150	Δ=75.6	22.678	
225.6			

calculated Δ= 5.064

calculated T_n= 27.742 psi

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

$$\theta_{eq} = \frac{(27.742)(226.48)(10.5)(5.285)}{(243.283)^2(2708.68)} = 2.1796\times10^{-3}\text{ rad}$$

$= 0.12489$ deg

≈ 0.125 deg

ω = 243.283 rad/sec = 2323.2 VPM

Exciting frequency = 450 RPM × 4th order = 1800 VPM

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

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

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

At MCR, Stress = (.31274)(9561.67 stress/deg) = 2990 psi

Resultant Stress:

4 order 2990 psi

$5\frac{1}{2}$ order 3958 psi

resultant = $\sqrt{\Sigma\,\text{stress}^2}$

= 4961 psi

25

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

~~4304.5 / 5.5 = 782.6 RPM~~

2323.2 VPM $\div 5.5 = 422.4$ RPM, $= 93$

At this speed, the 4th order excitation freq is

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

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

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

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

$$\text{Stress} = (.2654 \text{ deg.})(4561.67 \text{ psi/deg}) = 2537 \text{ psi}$$

Resultant stress (RMS sum)

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

Overall amplitude __should__ be $\frac{4701.4}{9561.67} = 0.492$ deg.

Stresses →

Rule allowable ($0.95R$) = 2134 psi for 11.8 in. dia.
1707 psi for 23.6 " "
= 2127 psi for 12 in. dia.

STEAR AFG

Material correction for ABS Grade 4 (UTS = 83000)

$$\text{Allowable} = 2127 \times \left(\frac{83000 + 30000}{90000}\right) = 2670.3 \text{ psi}$$

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

7%k.

mcy

3S
EEDS
:E
JWABL

:OE

31

Sheet 3 of ...

Actual material is better than Grade 4 -

min. UTS 100,700 psi

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

For all orders, Allowable = 3072 × 1.5 = 4608.5

Calculated stresses still exceed allowable.

By 1984 Rules, allowable = 3357 psi × 1.5 = 5035 psi - OK

Torsiograph results:

Calculated		Measured
4th Order ~~(pk-pk)~~ (single pk) -	.3127 degrees	.325° - .339°
$5\frac{1}{2}$ Order ~~(pk-pk)~~ (single pk)	.414	not measured
Combined $\left(\sqrt{\Sigma \text{stress}^2}\right)$, (single pk)	.492	.427° - .454° (stress < 4341 psi)

Question - are measured results single-peak?
Per telcon 4/12 w/ Gene Montgomery, LILCO, all ~~-~~ submitted measurements are single amplitude (Ø-peak or Ø-RMS). Therefore, agreement w/ calculations is good.
Per telcon 4/18 w/ Gene Montgomery, LILCO, cyl. exhaust temperatures are monitored - max. allowable diff. is 75°F.

16 April, 1984

CIECO

Hull No. Shoreham Plant

BY [illegible]

MANUFACTURER Transamerica Delaval

MODEL DSR/DSR-48

CRANKSHAFT SAFETY FACTOR BY CIMAC METHOD

(SI UNITS ONLY)

		*
		- (READ SIDE A)
		38R - (READ SIDE B)
1		J
[illegible]	S	- CYL. BORE, mm
11.7[illegible]	S	- MAX. FIRING PRESSURE, N/mm²
157.167[illegible]	S	- EFFECTIVE MOMENT ARM, mm (from Prelim. Calc
175	S	- W (Web Thickness) } mm
5[illegible]3.[illegible]	S	- B (Web Width) } mm
[illegible]		◇ - σ_{BN} (nominal bending stress, N/mm²)
		J
[illegible]	S	- D (crankpin dia.) } mm
17[illegible]	S	- W (Web Thickness) } mm
[illegible]	S	- S (pin overlap) } mm
533.[illegible]	S	- B (Web Width) } mm
1[illegible]	S	- R_H (Crankpin Fillet Radius) } mm
73.[illegible]	S	- D_{BG} (Journal Bore Dia.) } mm
73.[illegible]	S	- D_{BH} (Crankpin Bore Dia.) } mm
2.7[illegible]		◇ - α_B (bending stress concentration factor)
1.[illegible]		◇ - α_T (torsional " ")
17.[illegible]	S	- TH (Web undercut at crankpin)
21.[illegible]1	S	- TG (" " " journal)
1.297[illegible]		◇ - F_{HH} (Recess Factor)
3.[illegible]11		◇ - $\alpha_B \times F_{HH}$
		J
45.54[illegible]	S	- σ_{BN}, N/mm²
3.5[illegible]11	S	- $\alpha_B \times F_{HH}$
0	S	- σ addition, N/mm² (20 mid size engine, 30 for [illegible])
[illegible].73	S	- τ_N (Nominal Torsional Vibration Stress) N/mm²
1.84[illegible]45	S	- α_T (Calc. from NOTES & UTS)
=28,362 psi ← 195.46607		◇ - σ_V (Reference Stress, N/mm²)
		J
694.3	S	- σ_B (Crankshaft Ultimate Tensile Strength, N/mm²)
1	S	- K (Manufacturing Factor)
19.05	S	- R (Crankpin Fillet Radius, mm)
304.[illegible]	S	- D (Crankpin Diameter, mm)
[illegible] ← 227.99697		◇ - σ_{DW} (Fatigue Strength, N/mm²)
195.46607	S	- σ_V
1.16643		◇ - SAFETY FACTOR (FOR WEB IN WAY OF CRANKPIN FILLET)

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

AED SHIPYARD SHEET 2.5
,19 HULL BY AED
MANUFACTURER
MODEL

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/mm² Divide by stress concentration
S - WEB THICKNESS OR PIN DIAMETER mm
S - WEB WIDTH (1J ONLY) mm
A/◊ - BENDING MOMENT
3J
S - BORE mm
S - MAX. FIRING PRESSURE N/mm² (mult. by 1.2 for V engine)
A/◊ - FIRING FORCE
4J
S - BENDING MOMENT (FROM 1J OR 2J)
S - FIRING FORCE (FROM 3J)
S - DISTANCE TO CRITICAL SECTION (x) from main brg ℄
S - ℄ TO ℄ DISTANCE BETWEEN MAIN BRGS. (ℓ)
A◊ - END MOMENT CONSTANT (FOR ABS METHOD)**
A◊ - 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

5	J	5J
3.50911	S	S - $\alpha_B \times F_{HH}$ (CIMAC BENDING STRESS CONC. FACTOR)
304.8	S	S - PIN DIAMETER
125	S	S - WEB THICKNESS (w)
533.4	S	S - WEB WIDTH (B)
7.02238	A ◊	A◊ - α_B (CORRECTED - BASED ON PIN - FOR ABS METHOD)

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

,19

Hull
Manufacturer
Model

Crankpin Safety Factor
(Steady + Alternating Load)
Metric, English, or SI Units

			1 J
	1		S - Bore
	131.[illegible]		S - ℄ to ℄ distance between main bearings
	[illegible]	S	S - end moment constant [1] (from prelim calc)
	[illegible]	S	S - Distance to critical section (from main brg ℄ to edge of web)
From [illegible]	214.3125	S	S - Maximum Firing Pressure (adjusted for V-engine [illegible])
	11.721	S	
			2J
	304.[illegible]	S	S - Crankpin Diameter
	26.[illegible]	A	-A◇ Nominal bending stress (If program starts at 2J, program will print)
	[illegible]	S	S - Estimated torsional vibration stress
	1.[illegible]	S	S - Torsional stress concentration Factor from other sheet
	26.[illegible]	S	S - Nominal alternating or steady bending stress
	7.[illegible]	S	S - Bending stress concentration factor [3]
	694.3	S	S - Crankshaft Ultimate Tensile Strength
From →	277.[illegible]	S	S - Crankshaft Fatigue Limit
[illegible] MAC = 33083 psi	9542[illegible]	S	S - proportionality constant [2] (see below)
	3500	S	S - Rated Power
	450	S	S - Rated RPM
	13.[illegible]	A	A◇ - Steady Torsional Stress
psi	36.[illegible]	A	A◇ - Combined Steady Stress
←	206.[illegible]	A	A◇ - Combined Alternating Stress
	19.[illegible]	A	A◇ - Steady Stress Safety Factor
	1.10695	A	A◇ - Alternating Stress Safety Factor
	1.04695	A	A◇ - Overall Safety Factor
	3500	S	S - Power
	450	S	S - RPM
	14.67487	A	(This portion of program repeats)
5313.7 psi ←	36.61925	A	
29440 psi ←	206.34127	A	
	18.95999	A	$.589 \frac{d^3}{BW^2}$
	1.10195	A	
	1.04410	A	

Based on LTL ...

[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	HP (English)	63025
mm	kg	HP (Metric)	716200
mm	N	KW (SI)	9542800

By AJD

Safety Factors – desired minimum = 1.34 – lowest for special review of another mfgr.

Fatigue Stress = 29440 psi

Fatigue Strength = 33083 psi (theoretical CIMAC) → S.F. = 1.105

39200 psi (submitted – no shot peen) → S.F. = 1.309

47000 psi (shot peened) → S.F. = 1.570

UTS = 100,7[illegible] psi

Steady Stress = ~~[illegible]~~ 5313.5 psi → S.F. = ~~[illegible]~~ 18.960

Combined S.F. – Goodman method $\left(\frac{1}{SF} = \frac{1}{SF_{steady}} + \frac{1}{SF_{alt}} \right)$

For CIMAC (theoretical) fatigue limit → 1.044 (LOW)

For submitted F.L. – no shot peen → 1.224 (LOW/MARGINAL)

For submitted F.L. – w/shot ~~[illegible]~~ peening → 1.450 (OK)

Combined S.F. – Elliptical method $\left(\frac{1}{SF^2} = \frac{1}{SF_{st}^2} + \frac{1}{SF_{alt}^2} \right)$

For CIMAC (theoretical) fatigue limit → 1.103 (LOW)

For submitted F.L. – no shot peen → 1.306 (OK) (but marginal)

For submitted F.L. – shot peened → 1.565 (OK)

NOTE: SEE FOLLOWING PAGES FOR FURTHER ~~[illegible]~~ INFO

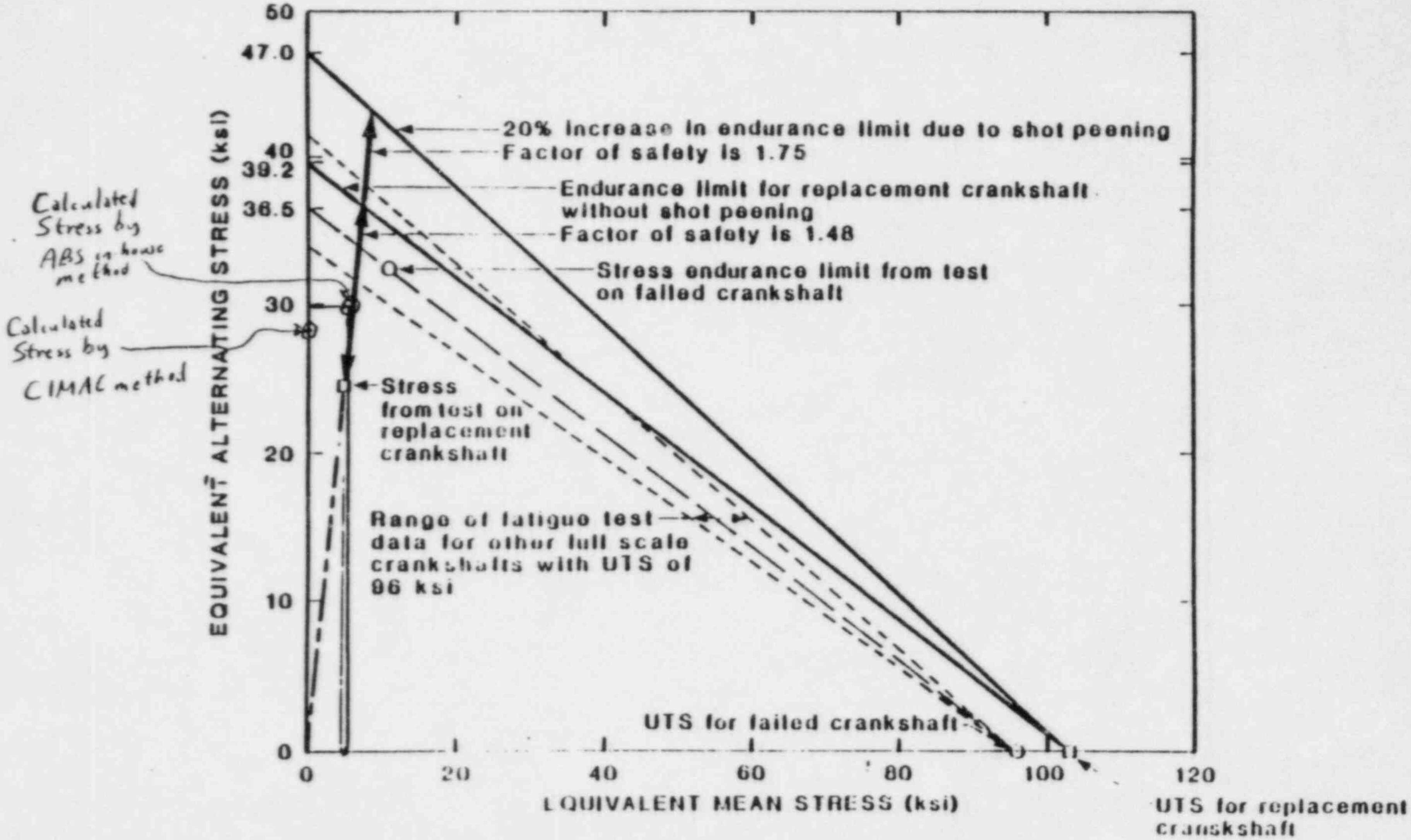


Figure 3-1. Goodman diagram for replacement crankshafts.

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 test results show ~~too~~ lower stresses than those calculated by either CIMAC or ABS formulas

Check crankshaft diameter by rules

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

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

$L = 609.6$ mm between brg. centers

$C = 1.0$ (eight cylinders)

$M = 1.86\ PD^2L$

$P = 1.196\ kg/mm^2 = 119.6\ kg/cm^2$ (≈ 1700 psi)

$D = 431.8$ mm

$L = 609.6$ mm

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

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

$H = 5300$ HP, $R = 450$ RPM

$T = 12013333$

1900 60,000

410 23000

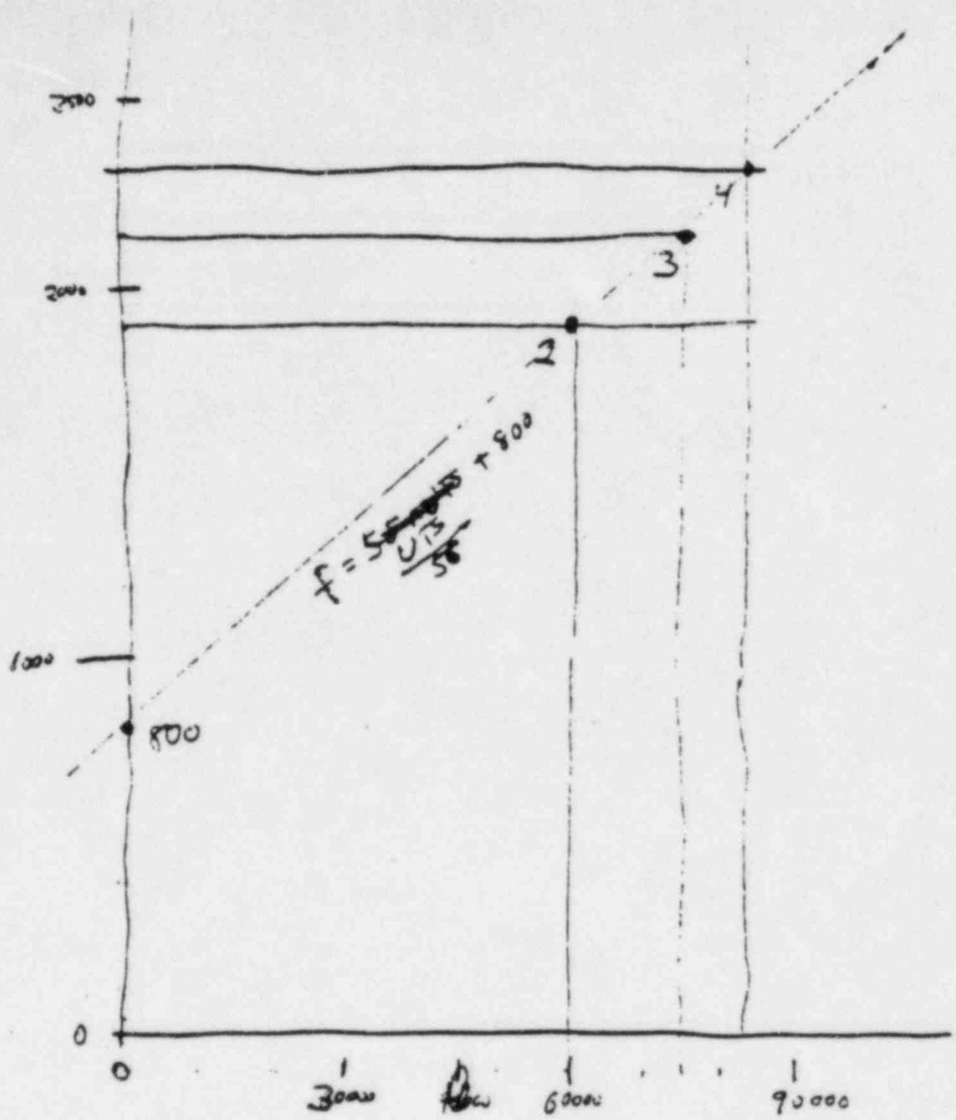
2310 63000

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

$= 264$ mm

$= 10.59$ inches

OK


2500
2000
1000
0
800
$f = 5\frac{\cancel{U}}{56} + 800$
2
3
4
0
30000
flow
60000
90000

KOBELCO
KOBE STEEL, LTD., TAKASAGO PLANT
2-3-1 SHINHAMA ARAI-CHO
TAKASAGO SHI HYOGO KEN 676 JAPAN
TELEPHON TAKASAGO 07944 3 4111
CABLE: KOBE STEEL TAKASAGO
TELEX: 5655-293 KOBSTL J

17 February 1984

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NOV 27 1984
DOCKETING & SERVICE BRANCH
SECY-NRC

Mr. Gregory M. Beshouri
Research & Development Engineer
Engine and Compressor Division
Transamerica Delaval Inc.
550 85th Avenue
Oakland, California 94621
U.S.A.

Dear Mr. Beshouri

It was my great pleasure to have your letter of December 21, 1983 on our crankshafts with fillet cold rolling.

First of all, I would like to stress that our fillet cold rolling technology will be very promising way for the cost reduction of your R5-type crankshafts. However, I also have to confess that at this moment Kobe Steel does not produce solid crankshafts with fillet cold rolling.

The reason is that the fillet cold rolling technology for solid crankshafts was developed more than ten years ago simultaneously with that for shrink fitting crankshafts. As the latter was applied to cast steel crankshaft, the technology got many interests of engine builders from the first stage and we have already delivered numerous shrink fit cast steel crankshafts with fillet cold rolling. On the other hand, requirements to solid crankshafts were not so severe that the fillet cold rolling technology for solid crankshafts could not have any actual demand of engine builders and was not extended to actual production. Consequently, we have enough data to get approval of ship classification societies on the improvement of fatigue strength of forged solid crankshafts,

DAR 8412170318

but have not a fillet cold rolling machine for the actual production.

I recieved similar questions with yours from some of our clients last year and we have decided to restart the development of fillet cold rolling machine for solid crankshafts. It probably requires about one year for the development. However, if your demand for this technology is very high, the time schedule will be rearranged.

As for the fatigue strength, a carbon steel crankshaft with fillet cold rolling (base metal tensile strength σ_B=60kg/mm^2) is higher than a heat treated low alloy steel crankshaft with the tensile strength of 70kg/mm^2. And the cost is estimated to be between those of non-treated carbon steel and low alloy steel crankshafts.

I estimate that shot peening on this size of crankshafts is a waste of time. Because, the hardened depth by shot peening is estimated to be quite shallow comparing with the depth of highly stressed area at fillets.

ABS Rule Book

The copies of Nippon Kaiji Kyokai Rule " Rules and Detailed Rules for Diesel Engine Crankshafts and Those Explanations" and the latest IACS's draft "Rules for the Calculation of Crankshafts for Diesel Engines" will be sent by a separate mail.

As we do not have NKK's rule book "Rules for the Survey and Construction of Steel Ships", I recommend you to contact with NKK's New York office. The address is:

Nippon Kaiji Kyokai
17 Battery Place, Room No.210,
New York, N.Y. 10004

212-425-3799

CRANKSHAFT BOOK
ENGINE BOOK -- 4
PRICE:
IF WE HAVE QUESTIONS:-
T. OKAYAMA

Please note that the IACS's draft mentioned above was editted bacing on CIMAC proposal "Rules on Calculation of

Crankshafts for Diesel Engines (4.Draft)" and is still under discussion among IACS members and between CIMAC and IACS. If you are interested in the detailed discussions on this matter, I recommend you to contact with Dr. Günter Donath who is the chairman of CIMAC Sub Group "Crankshaft Dimentions". His address is

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Cordially,

S. Denoh

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