



ABS Check Cales

EXH- 3.
7/16/14

8412170310 841001
PDR ADOCK 05000322
G PDR

C-47-1

MASYS	MASYS	USERID	ORIGIN
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XEROX	OUTPUT2	FILENAME	FILETYPE
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9710	00000113	SPOOLID	COUNT
04/09/84	17:09:26	FILE PRINT DATE	
H	406	CLASS	DEVICE
STANDARD		FORMS	

SSS
SSS SSS

C-47-2

FILE: FORM XVO6 • ABS COMPUTERS, INC.

PAGE 001

C-47-3

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	,	NONEXISTENT

C-47-4

FILE: FORM

NUFRONT *

ABS COMPUTERS, INC.

C-47-5

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	.937082	3.41321E+08
1	3	.601367	5.2716E+08
1	4	8.28666E-02	5.52768E+08
1	5	-.460822	4.10362E-09
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

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ZCPT1	ABSNYA	DISTCODE	SYSTEM
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9727	00002514	SPOOLID	COUNT
04/09/84	17:09:42	FILE PRINT DATE	
H	406	CLASS	DEVICE
STANDARD		FORMS	

Figure 1 shows a 10x10 grid representing a 100-cell system. The grid is divided into four quadrants by a vertical line at column 5 and a horizontal line at row 5. The top-left quadrant (columns 1-4, rows 1-4) contains 16 cells, with 15 cells filled with asterisks. The top-right quadrant (columns 6-10, rows 1-4) contains 16 cells, with 15 cells filled with asterisks. The bottom-left quadrant (columns 1-4, rows 6-10) contains 16 cells, with 15 cells filled with asterisks. The bottom-right quadrant (columns 6-10, rows 6-10) contains 16 cells, with 15 cells filled with asterisks. The center of the grid (column 5, rows 5-10) contains 10 cells, with 9 cells filled with asterisks. The cells at (5,5), (5,6), (5,7), (5,8), (5,9), and (5,10) are empty.

[illegible]

ABS COMPUTERS

Check of Fractional Vibration

Hysteresis Damping -

DeLaval

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

1750 rpm

12 April 54

Sheet 1 of 6

Eng. C. W. S.

We know

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

or

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

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

Therefore, DeLaval assumes dynamic magnifier of 25.

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

Insert into
Chart

For a 4-cycle engine, use Portin Q2 curves

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

	<u>MIP, psi</u>	<u>T_n, psi</u>	
$\Delta = 20.3$ $\Delta = 20.5$ $\Delta = 75.6$	109.2	10.601	$\Delta = .598$ $\Delta = .575$
	129.5	11.199	
	150.0	11.774	
	225.6	calculated = 13.948	calculated $\Delta = 2.1735$

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

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

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

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

$$\Theta_{eq} = \frac{(13.948)(226.98)(10.5)(1.394)}{(243.283)^2 (2708.68)} = 2.89 \times 10^{-4} \text{ rad} = 0.0165^\circ$$

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

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

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

max = 4.0
Dist

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

crankpins to flywheel = 12"

generator shaft = 16"

Worst stresses for each size shaft:

$$\text{front gear to cyl. \#1} - \frac{\text{Stress}}{\text{deg}} = \frac{4,83317 E^6}{11.25 (8)^3} = 839 \text{ psi/deg}$$

$$\text{crankpins to flywheel - (max. after mass \#9)} - \frac{\text{Stress}}{\text{deg}} = \frac{1,85879 E^8}{11.25 (12)^3} = 9561.67 \text{ psi/deg}$$

$$\text{generator shaft} - \frac{\text{Stress}}{\text{deg}} = \frac{1,57544 E^8}{11.25 (16)^3} = 3419.03 \text{ psi/deg}$$

Worst stress is in engine.

Equilibrium amplitude = .01656 degrees

$$\text{stress} = (.01656)(9561.67) = 158.34 \text{ psi}$$

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

Sheet 4 of 6
 2/27/71

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
	129.5	21.324	
	150	22.678	
$\Delta = 20.5$	225.6		
$\Delta = 75.6$			
		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)

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

$$= 0.12489 \text{ deg}$$

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

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

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

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

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

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

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

Resultant Stress:

4 order 2990 psi

5 1/2 order 3958 psi

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

= 4961 psi

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

$$400 \times 5.5 = 2200 \text{ RPM}$$

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

At this speed, ~~the~~ $\frac{\omega}{\omega_n}$ 4th order excitation freq
is

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

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

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

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

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

Resultant stress (RMS am.)

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

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)

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

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

7706

may

SS

FEDS

2

SW486

OE

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

Sheet =
OK

$$\text{Single Harmonic Allowable} = 2127 \times \left(\frac{100,000 + 30,000}{90,000} \right) = 3072 \text{ psi}$$

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

Calculated stresses still exceed allowable.

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

Torsigraph results:

<u>Calculated</u>	<u>Measured</u>
4th Order (pk pk) (single pk) - .3127 degree	.325° - .339°
5½ Order (pk pk) (single pk) .414	not measured
Combined ($\sqrt{\epsilon_{stress}^2}$), (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 (ϕ -peak or ϕ -RAIS). Therefore
agreement w/ calculations is good.

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

Model DSM/DSR-48

By _____

CRANKSHAFT SAFETY FACTOR BY CIMAC METHOD
(SI UNITS ONLY)

191.1		*	- (READ SIDE A)
11.721		38 R	- (READ SIDE B)
157.167		J	- CYL. BORE, mm
175			- MAX. FIRING PRESSURE, N/mm ²
575.1			- EFFECTIVE MOMENT ARM, mm (from Prelim. Calc)
45.5475			- W (Web Thickness) } mm
			- B (Web Width) }
		◇	- σ_{BN} (nominal bending stress, N/mm ²)
		J	- D (crankpin dia.)
			- W (Web Thickness)
			- S (pin overlap)
			- B (Web Width)
			- R _n (Crankpin Fillet Radius)
			- D _{JE} (Journal Bore Dia.)
			- D _{BN} (Crankpin Bore Dia.)
		◇	- α_B (bending stress concentration factor)
		◇	- α_T (torsional " " " ")
			- TH (Web undercut at crankpin)
			- T _G (" " " journal)
		◇	- F _{HN} (Recess Factor)
		◇	- $\alpha_B \times F_{HN}$
		J	- σ_{BN} , N/mm ²
			- $\alpha_B \times F_{HN}$
			- $\sigma_{addition}$, N/mm ² (20 md sine wave, 30 for curve)
			- τ_N (Nominal Torsional Vibration Stress) N/mm ²
			- α_T
		◇	- σ_V (Reference Stress, N/mm ²)
		J	- σ_B (Crankshaft Ultimate Tensile Strength, N/mm ²)
			- K (Manufacturing Factor)
			- R (Crankpin Fillet Radius, mm)
			- D (Crankpin Diameter, mm)
		◇	- σ_{DW} (Fatigue Strength, N/mm ²)
			- σ_V
		◇	- 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.

647-17

PRELIMINARY CALCULATIONS FOR
CRANKSHAFT SAFETY FACTOR ANALYSISFIND END MOMENT CONSTANT OR EFFECTIVE μ

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 μ S - μ TO μ 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 - $\Delta_B \times F_{MH}$ (CIMAC BENDING STRESS CONC. FACTOR)

S - PIN DIAMETER

S - WEB THICKNESS (W)

S - WEB WIDTH (B)

A/D - Δ_B (CORRECTED - BASED ON PIN - FOR ABS METHOD)3.50911
304.0
123
633.4
7.02233

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

CRANKPIN
HULL
MANUFACTURER
MODEL

By ASD

CRANKPIN SAFETY FACTOR
(STEADY + ALTERNATING LOAD)
Metric ENGLISH, OR SI UNITS

131.3
214.3125
11.721
274.7276
27.92
1.74
26.7276
7.2232
644.3
From → 277.94611
JMAC 7547.00
33083
3500
450
19.32021
34.31277
206.34147
19.32497
1.10695
1.04695

1 J
S - Bore
S - ϕ to ϵ distance between main bearings
S - end moment constant [1] (from prelim calc)
S - Distance to critical section (from main brg ϕ)
S - Maximum Firing Pressure (adjusted for V-engine) ^{to engine spec}
2 J
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 ^{from other sheet}
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)

PS - Rated Power
PS - Rated RPM

Based on L11

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	[1] - End Moment Constant			
S - RPM	0 = simple support (limiting value)			
(This portion of program repeats)	0.67 = typical value			
	2 = built-in ends (limiting value)			
	[2] - Length Force Power Proportionality Constant			
.589 $\frac{d^3}{BW^2}$	inch	pound	HP (English)	63025
	mm	kg	HP (Metric)	716200
	mm	N	KW (SI)	9542800

MMANDS.

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

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 = ~~5000~~ 5313.5 psi \rightarrow S.F. = ~~14.83~~ 18.960

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

For CIMAC ^(theoretical) 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~~ ^{peening} \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 ^(theoretical) 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

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Calculated Stress by ABS in house method

Calculated Stress by CIMAC method

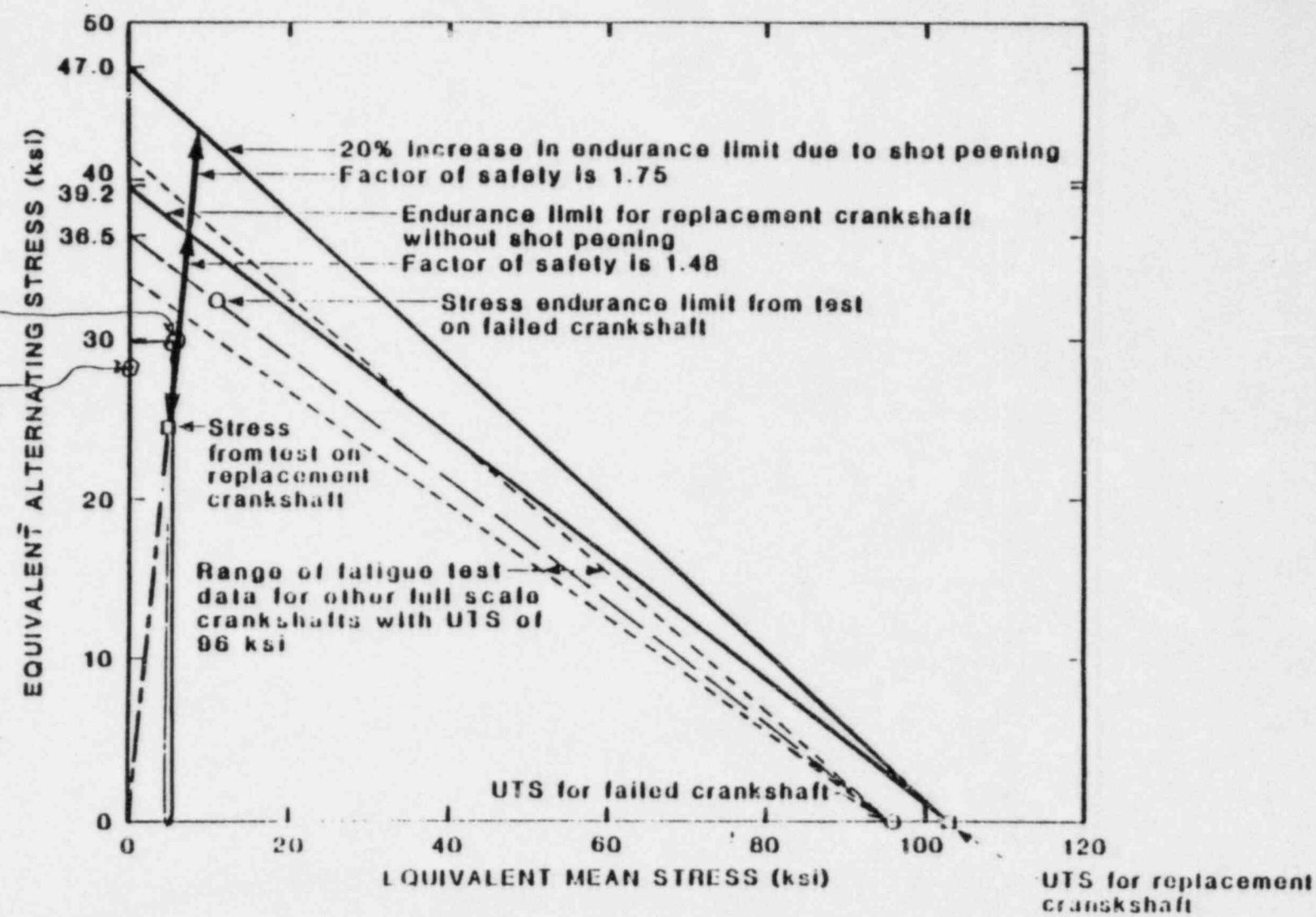


Figure 3-1. Goodman diagram for replacement crankshafts.

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 ~~the~~ lower stresses than those calculated by either CIMAC or ABS formulas

Check crankshaft diameter by formula

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

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

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

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

$$M = 1.86 PD^2 L$$

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

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

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

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

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

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

$$T = 12013333$$

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

$$= 269 \text{ mm}$$

$$= 10.59 \text{ inches}$$

OK

$$\begin{matrix} 1900 & 60,000 \\ \times & (418) & 23000 \\ \hline 2310 & 83000 \end{matrix}$$

$$P = 5.8 \times (UTS + 60,000) + 1300$$

