THE SOCIETY OF NAVAL ARCHITECTS AND MARINE ENGINEERS
One World Trade Center, Suite 1369, New York, N.Y. 10048

Papers to be presented at Extreme Loads Response Symposium
Arlington, VA, October 19–20, 1981

Fatigue Criteria for Ship Structure Details

W. H. Munse, University of Illinois, Urbana, IL

ABSTRACT

Fatigue criteria are presented for the design of ship structures. The criteria take into account the ship structure details, the fatigue resistance of specific locations in these details, the variable loading to which a ship is subjected, and the desired level of reliability (factor of safety). A design example indicates the simple manner in which the criteria can be used.

NOMENCLATURE

log C	= The life intercept of the S-N curve
$f_{S(s)}$	= Probability density function-Weibull distribution (Equation 2)
k	= Shape parameter for the Weibull distribution
L(n)	= The probability of no failure through a life, n (Equation 9).
m	= Negative slope of S-N curve
N	= The mean life necessary to produce a useful life n with a reliability of L(n).
(P_F)	= Probability of failure = [1-L(n)].
R_F	= Reliability factor (Equation 13).
s	= Stress range
S_D	= Constant-cycle fatigue design stress range for a useful life n and reliability L(n).
$(S_D)_{all}$	= The maximum allowable fatigue stress range (Equation 15).
$(S_{max})_N$	= The maximum stress range of a variable loading history that is expected to produce failure in 10^8 cycles.
S_N	= Mean constant-cycle stress range for failure at N cycles.
S_{10^8}	= The maximum stress range expected to occur once in 10^8 cycles, based on a Weibull distribution.
w	= Characteristic value of S (Equation 5).
γ_L	= Scatter factor (Equation 11).
Γ	= Gamma function
μ_S	= Mean stress of the Weibull distribution (Equation 3).
ξ	= Random load factor (Equation 7).
σ_S	= Standard deviation of the Weibull distribution.
Ω_N	= Total uncertainty in fatigue life (coefficient of variation).

INTRODUCTION

Fatigue cracking in ships has been a serious problem for many years. As noted by Vedeler (1)[1] in 1962, shipbuilders in Norway and Sweden considered the problem of fatigue in ships to be of more practical importance for ordinary ships than the question of brittle fracture. He noted that fatigue cracks were often found in the forepeak region, bottom amidships, at the bulwark at both ends of the bridge, and in the hatch corners. In a recent study it has been observed that ships may also have cracks at crossings of frames, longitudinals, and girders, and many other locations (2, 3 and 4).

[1]Reference numbers are indicated in parentheses.

Since such cracks may be possible points of initiation for catastrophic failures and, since the repair of fatigue cracks can be very costly, it is essential that fatigue be given adequate consideration in the design of ship structures.

In this paper criteria are presented for the fatigue design of ship structural details along with a brief discussion of the principal parameters included in the criteria.[2] An example of the application of these design criteria is then presented to indicate the simple manner in which the criteria can be used to provide a verification of the adequacy of a ship in fatigue.

FATIGUE DESIGN PARAMETERS

Fatigue design generally consists of verifying that the details of a structure have sufficient resistance to repeated loading to provide a fatigue life equal to or greater than required: a verification by checking process. To achieve this for ship structure details, criteria have been developed which takes into account, (a) the basic fatigue behavior of welded structural details, (b) the types of details found in ship structures at which cracking has been observed, (c) the loading histories to which ships may be subjected, and (d) the level of fatigue safety to be included in the ship design.

There are other factors that can affect the fatigue behavior of a structure also; however, their importance or effect is not as great as the effect of those noted above and to include them would have greatly complicated the overall design process. These neglected factors include, (a) the mean stress effect, an effect that is relatively small for welded details and is now neglected in most fatigue design specifications, (b) the tensile strength of the steel, another factor that is generally neglected in structural fatigue design specifications and considered to have relatively little effect in long life fatigue, and (c) temperature, rate of load application, residual stresses, and size effect, again factors that are generally found to be of secondary importance.

[2]The ship design criteria presented herein were developed in an investigation conducted at the University of Illinois and sponsored by the Ship Structure Committee.

Fatigue Behavior of Weldments

During the nearly 50 years of laboratory studies conducted on weldments, numerous papers, conference proceedings and books have been published wherein detailed fatigue data for welds and weldments may be found (5-11). Recently, much of this information has been re-examined to establish basic S-N relationships for numerous types of welded members and details (12). Nearly 1500 S-N curves have been produced, one example of which is presented in Fig. 1. (This is the S-N curve for axially loaded longitudinal full penetration groove welds with the reinforcement intact; based on stress range; and for mild, high strength low alloy or quenched and tempered steels. Identified by DAAAXB).

The solid line mean regression curve (50% reliability) of Fig. 1, as established by a least-square analysis, can be given by,

$$\text{Log } N = \log C - m \log S_N \qquad (1)$$

$$\text{or } S_N = \left(\frac{C}{N}\right)^{\frac{1}{m}} \qquad (1a)$$

where,

log C = the life intercept of the S-N curve

N = number of cycles to failure for a constant cycle stress range of S_N.

m = the negative slope of the S-N curve

S_N = the constant-cycle stress range for failure at N cycles.

Using such straight-line relationships, the mean fatigue stress ranges for lives of 10^5, 10^6, 10^7, 10^8 cycles, have been established for numerous welded members and details, and are presented in Table I[3]. These mean fatigue strengths are for the numerous structural fatigue details shown in Fig. 2.

For each of the details shown in Fig. 2, the location at which the fatigue strength applies is the point where the greatest stress concentration exists and, except as noted, can be

[3]Under random loadings straight-line S-N relationships are found to extend well beyond the fatigue limits often reported for constant cycle tests (10). Therefore, in this development, the straight-line S-N relationship has been assumed to extend to 10^8 cycles or more.

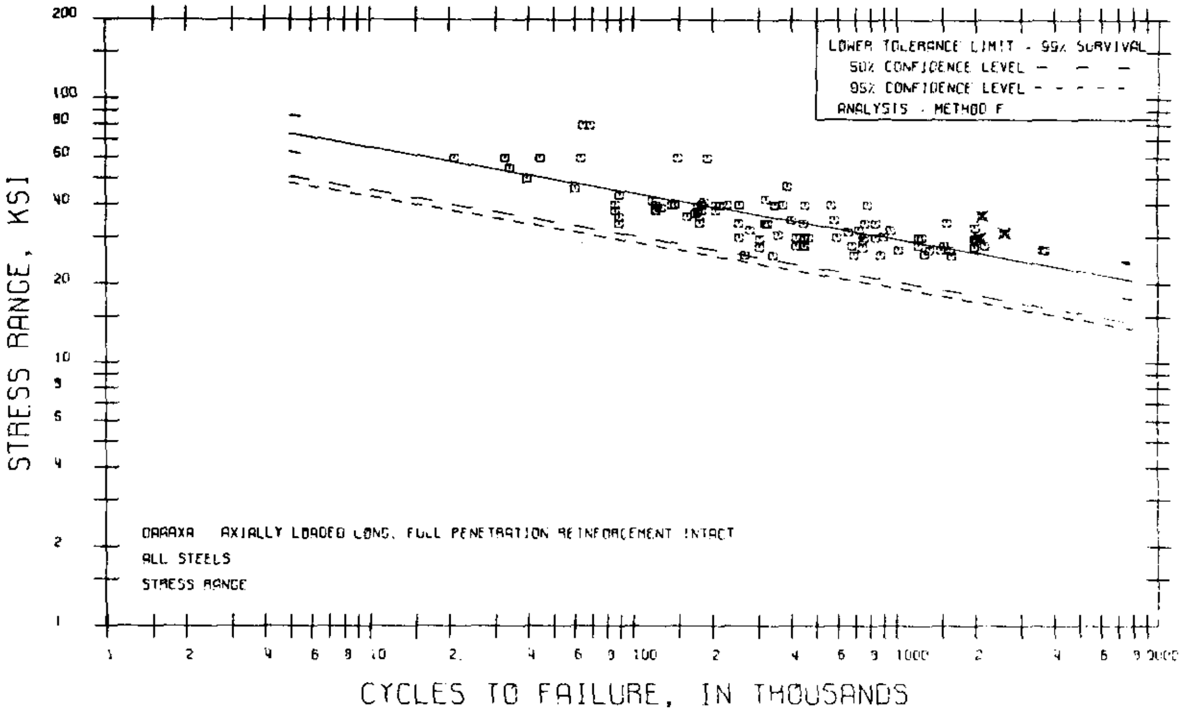


Fig. 1. S-N Relationship for a Weldment Containing a Longitudinal Groove weld in the As-Welded Condition. (Detail No. 3)

considered to be a function of the principal tensile stress at that location. This location, for example, is at the end of the cover plate for detail No. 5, is at the toe of the butt weld for detail No. 10, and is at the side of the hole in detail No. 28.

The data in Table I and the diagrams of Fig. 2 provide the basic fatigue information on which the fatigue design criteria presented herein are based.

Ship Details

After the basic fatigue data had been assembled, the second phase in the development of the fatigue design procedure was the indentification of those ship locations at which fatigue cracking might occur. Two recent reports on the in-service performance of structural ship details (3, 4) have served to define possible fatigue critical locations in ships. These reports catalog and define the types and locations of details at which failures have occured in a variety of merchant and Naval vessels.

A total of 86 ships were included in the surveys. The details examined were separated into twelve general families (see Table II) and these were in turn divided into 56 groups of 634 separate configurations. A total of 6856 failures were found in the 607584 details observed. In the investigation on which this paper is based, the locations in each of these configurations at which fatigue might develop have been identified and will provide guidance in locating the details for which possible fatigue failure should be checked (22). An example of a few of these families and configurations of details is presented in Fig. 3.

The locations in Fig. 3 at which fatigue cracking might develop are circled and identified by the corresponding structural detail number from Fig. 2. (The basic fatigue resistance for the detail is provided in Table 1).

Not all details in Fig. 2 are found in ships and not all of the details used in the ships surveyed have exhibited cracks. Only the thirty-seven details listed in Table III are those for which cracking was reported and then, some of the details exhibited only a very limited number of cracks. Those for which the largest number of cracks were found are those for which these fatigue design criteria will be of greatest value. However, in design a fatigue evaluation should be made of all details for which fatigue cracking is a possibility.

Ship Loadings

To properly evaluate the fatigue adequacy of a ship detail requires a realistic estimate of the cyclic stress history to which the detail will be subjected during its lifetime. Since the existing full-scale ship loading data are for a limited length of time, it has been necessary to extrapolate the available data to obtain an estimated lifetime spectrum (13, 14). Such

TABLE I

Mean Fatigue Stress Range for Fatigue Details in Fig. 2.

(Constant Cycle - 0.50 Reliability)

Detail No.[1] (see Fig. 2)	S-N Slope, m	Stress Range, ksi, for n cycles[2]			
		$n = 10^5$	$n = 10^6$	$n = 10^7$	$n = 10^8$
1	5.73	69.4	46.5	31.1	20.8
1(F)	4.80	67.1	41.5	25.7	15.9
2	6.05	61.5	42.0	28.7	19.62
3	5.77	44.1	29.6	19.9	13.33
3(G)	5.94	41.2	27.9	19.0	12.87
4	6.08	41.4	28.3	19.4	13.30
5	3.25	26.4	13.02	6.42	3.16
6	6.08	41.4	28.3	19.41	13.30
7	4.11	39.8	22.71	12.97	7.41
8	6.54	55.8	39.2	27.6	19.4
9	9.64	32.6	25.7	20.2	15.92
9(S)	8.85	48	37	28.5	22.0
10	7.44	39.9	29.3	21.5	15.76
10(G)	9.32	47.2	36.9	28.8	22.49
11	6.13	33.1	22.73	15.62	10.73
11(G)	8.65	29.4	22.5	17.26	13.23
12(G)	5.66	40.8	27.2	18.09	12.05
12	3.98	35.0	19.6	11.00	6.17
13	4.23	48.3	28.0	16.27	9.44
14	7.43	40.6	29.8	21.8	16.03
15	3.48	25.98	13.40	6.91	3.57
16	4.63	32.8	19.93	12.12	7.37
16(G)	6.97	32.8	23.6	16.94	12.17
17	3.73	27.8	15.00	8.10	4.37
17(S)	9.52	28.9	22.7	17.81	13.99
18	4.03	20.30	11.46	6.47	3.65
18(S)	9.22	25.7	20.02	15.60	12.15
19	7.49	23.1	17.00	12.49	9.18
19(S)	7.53	27.5	20.28	14.93	10.99
20	3.94	32.9	18.36	10.23	5.70
20(S)	6.44	28.02	19.60	13.71	9.59
21	3.94	(32.9)	(18.36)	(10.23)	(5.70)
21(S)	7.36	42.4	31.0	22.7	16.59
22	3.15	39.8	19.16	9.22	4.43
23	3.26	35.7	17.6	8.68	4.28
24	3.26	35.7	17.6	8.68	4.28
25	7.09	33.2	24.0	17.36	12.54
26A	8.53	49.9	38.1	29.1	22.20
25B	3.95	(34.9)	(19.51)	(10.89)	(6.08)
26	3.46	31.9	16.41	8.44	4.34
27	4.85	22.8	14.17	8.81	5.48
27(S)	4.48	22.8	13.83	8.16	4.88
28	7.74	40.1	29.8	22.11	16.42
28(F)	4.81	(29.4)	(18.21)	(11.28)	(6.99)
29	-	-	-	-	-
29(F)	-	-	-	-	-
29R1	-	-	-	-	-
29R2	-	-	-	-	-
30	2.83	38.0	16.83	7.46	3.31

TABLE I (CONTINUED)

Detail No.[1] (See Fig. 2)	S-N Slope, m	Stress Range, ksi, for n cycles[2] $n = 10^5$	$n = 10^6$	$n = 10^7$	$n = 10^8$
30A	(6.23)	(34.0)	(23.5)	(16.1)	(11.16)
31	4.35	(20.16)	(11.87)	(6.99)	(4.12)
32	-	-	-	-	-
32A	3.48	(25.98)	(13.40)	(6.91)	(3.57)
32B	3.53	21.50	11.21	5.84	3.04
32C	-	-	-	-	-
33	3.66	21.34	11.38	6.07	3.23
33(S)	10.39	25.5	20.45	16.38	13.12
34	3.66	(21.34)	(11.38)	(6.07)	(3.23)
34(S)	10.39	(25.5)	(20.45)	(16.38)	(13.12)
35	3.55	33.6	17.56	9.18	4.80
36	3.95	34.9	19.51	10.89	6.08
36A	3.95	34.9	19.51	10.89	6.08
37	-	-	-	-	-
37S	-	-	-	-	-
38	3.71	32	17.2	9.24	4.97
38(S)	10.23	16.27	13.00	10.37	8.28
39	-	-	-	-	-
39A	-	-	-	-	-
39B	-	-	-	-	-
39C	-	-	-	-	-
40	3.53	(21.50)	(11.21)	(5.84)	(3.04)
41	-	-	-	-	-
42	5.02	110.2	69.8	44.08	27.88
43	-	-	-	-	-
43A	-	-	-	-	-
44	-	-	-	-	-
45	-	-	-	-	-
46	4.35	(20.16)	(11.87)	(6.99)	(4.12)
47	-	-	-	-	-
47A	-	-	-	-	-
48	-	-	-	-	-
48R	-	-	-	-	-
49	-	-	-	-	-
50	-	-	-	-	-
51	-	-	-	-	-
52	-	-	-	-	-
53	-	-	-	-	-

[1](S) Indicates shear stress on fasteners or welds.

(F) Indicates flame cut surfaces.

(G) Indicates the surfaces have been ground flush.

[2]Estimated values are shown in parentheses ( ).
A dash is provided where no data are available.

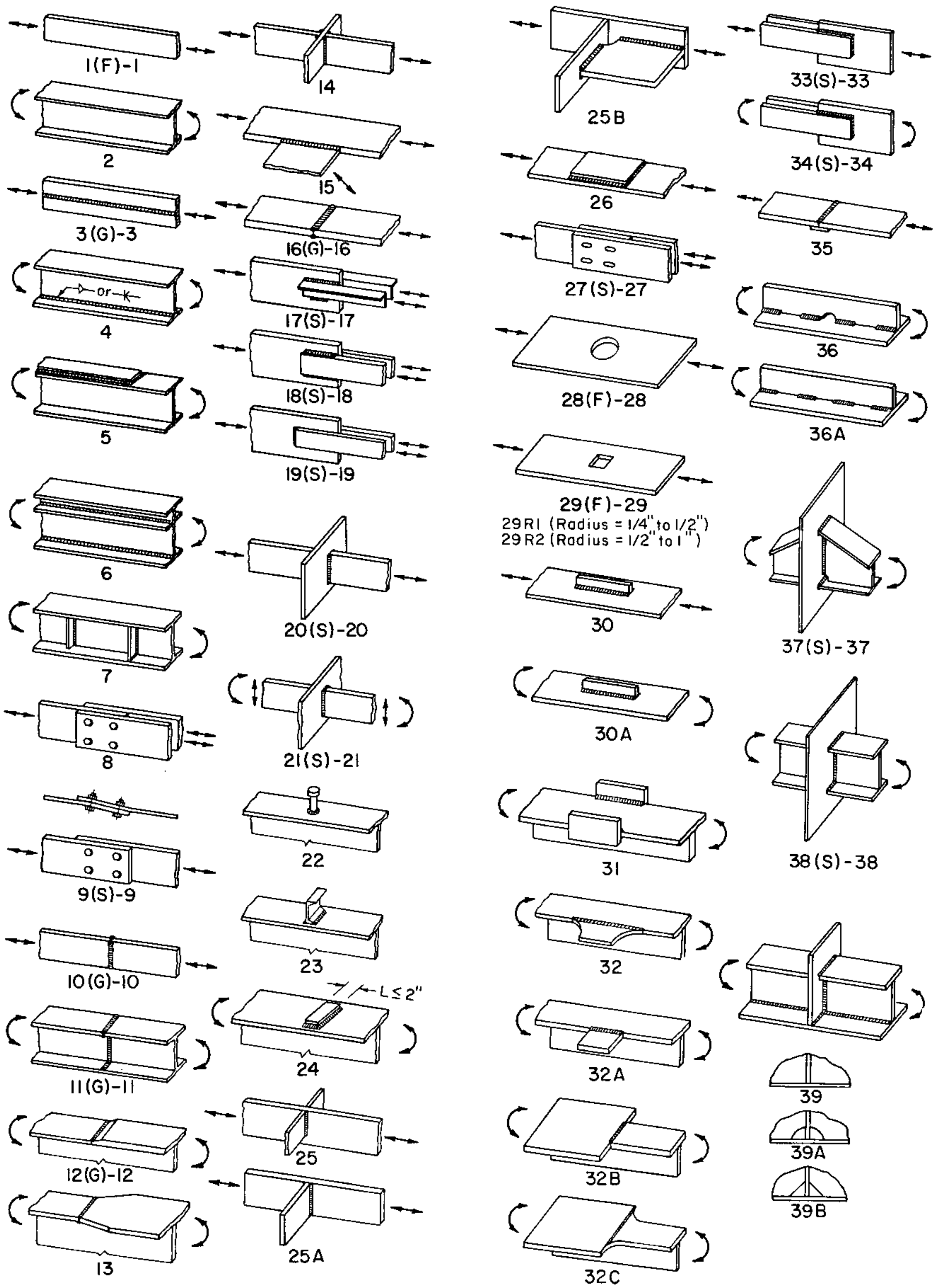
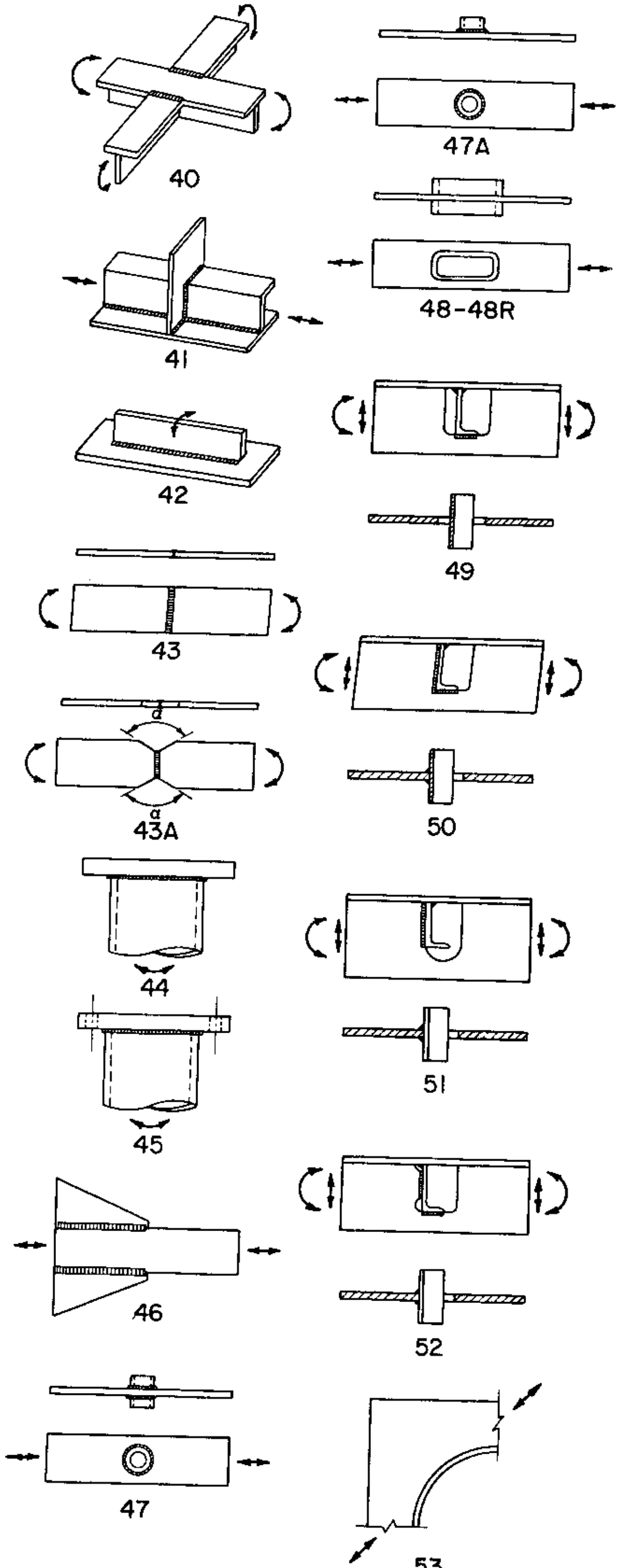


Fig. 2. Structural Fatigue - Details



(G) - Designates a ground surface
(S) - Designates shear on weld or fastener
(F) - Designates flame-cut edges for comparison with machined edges

Fig.2. Structural Fatigue-Details (cont.)

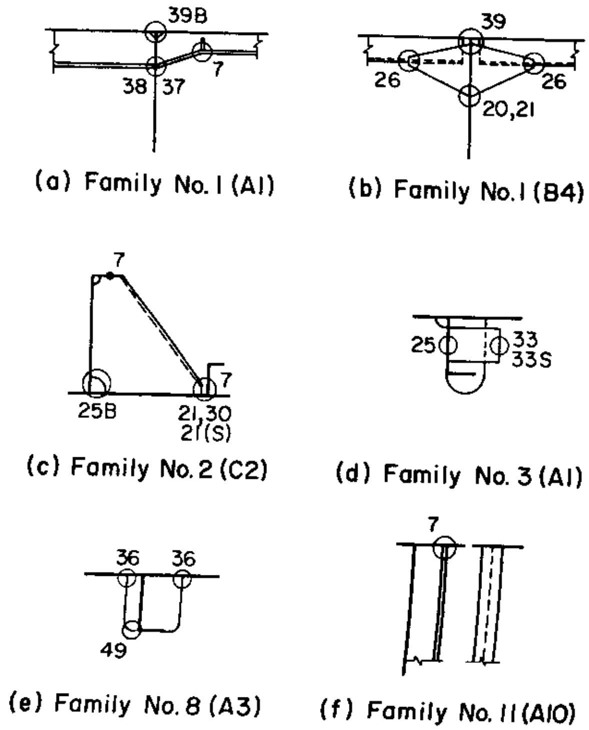


Fig. 3. Examples of Configurations in ship family details (3,4)

TABLE III

Details Exhibiting Failures in Ships

Detail No.	Total No. of Cracks	Detail No.	Total No. of Cracks
7	272	32B	2
9	7	33	36
14	7	33S	20
17	2	34	23
17S	2	34S	17
19	42	36	600
19S	40		
20	318	37	462
		38	8
21	1300	40	2
21S	54	41	11
26	155	42	7
28	208	43	75
28F	222	44	14
29	9	47	29
29R	3		
29F	7	48R	25
		50	2
30	142	51	687
30A	672	52	105
		53	8

TABLE II

Summary of Data for 12 Detail Families (Ref. 4)

Family No.	Detail Family Name	Totals Observed		
		Total No. Details	Total No. Failures	% Failures
1	Beam Bracket	68,586	2,252	3.28
2	Tripping Bracket	34,012	1,587	4.67
3	Non-Tight Collar	20,974	33	0.16
4	Tight Collar	20,654	46	0.22
5	Gunwale Connection	172	5	2.91
6	Knife Edges	0	0	-
7	Miscellaneous Cutouts	296,689	853	0.29
8	Clearance Cutouts	57,307	843	1.47
9	Deck Cutouts	7,534	29	0.38
10	Stanchion Ends	7,090	122	1.72
11	Stiffener Ends	40,729	298	0.73
12	Panel Stiffeners	53,827	788	1.46
	Totals	607,584	6,856	1.13

extrapolations, plotted on a semi-log cumulative distribution basis for large tankers and dry cargo vessels are presented in Fig. 4 and are based on the wave-induced longitudinal bending stresses. A complete loading history should include also the high-frequency dynamic loadings. However, since these are generally of a relatively small stress-range and would produce little damage they have been neglected in the present study. (The somewhat conservative previous assumption that the S-N curve is linear to 10^8 cycles tends to compensate for this neglect of the high-frequency stresses.) Nevertheless, if it can be demonstrated that the stress ranges for the expected high-frequency loadings will not be small (above about 6 ksi), then they should be included in the total loading history.

To use the fatigue design criteria developed herein requires that the loading history be represented in probabilistic terms by a probability distribution function. A variety of distribution functions were investigated, including the Beta, Lognormal, Weibull, Exponential and Rayleigh distributions; however, the evaluation clearly indicated that the Weibull distribution would most effectively define existing ship loading data.

The two parameter Weibull probability distribution function can provide many shapes to model the loading history, including those of the Exponential and Rayleigh distributions, and is given by:

$$f_{S(s)} = \frac{k}{w} \left(\frac{s}{w}\right)^{k-1} \exp \left[-\left(\frac{s}{w}\right)^{k}\right] \qquad (2)$$

where:

k = shape parameter

w = characteristic value of s

s = stress

The mean stress of the distribution is,

$$\mu_S = w\, \Gamma\, (1 + 1/k) \qquad (3)$$

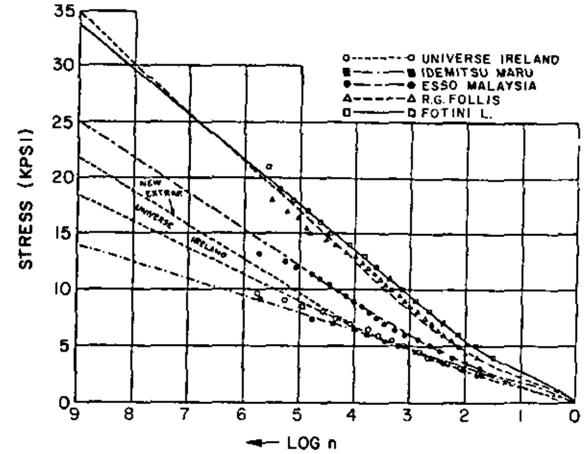
and the standard deviation is,

$$\sigma_S = w\, [\Gamma(1 + 2/k) - \Gamma^2(1 + 1/k)]^{1/2} \qquad (4)$$

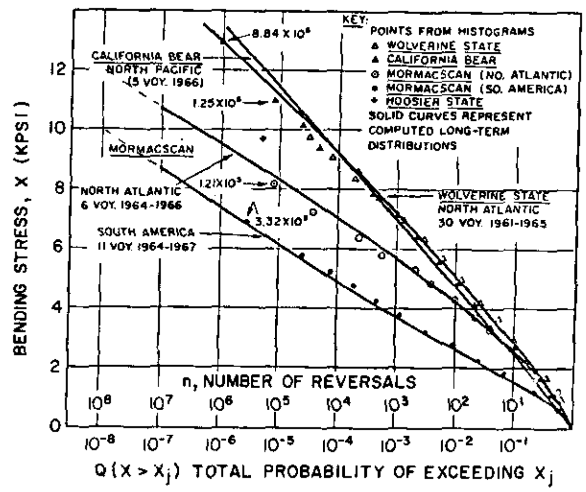
where;

Γ = Gamma function

The general configurations of several Weibull distributions (various values of k) are shown in Fig. 5. These are similar to the distributions often reported for strain measurements made on ships at sea. An indication of the excellent fit of a Weibull distribution with k = 1.2 and the data obtained from Sea-Lands SL-7 shipboard measurements (15) is presented in Fig. 6.



(a) Service Stresses In Large Tankers. (13)



(b) Service Stresses In Dry Cargo Vessels. (14)

Fig. 4. Long-Term Trends in Service Stresses for Large Tankers and Dry Cargo Vessels (13,14)

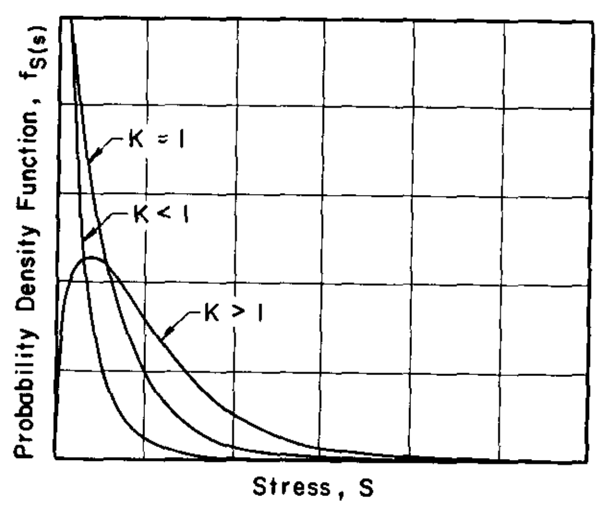


Fig. 5. General Shapes of Weibull Distributions

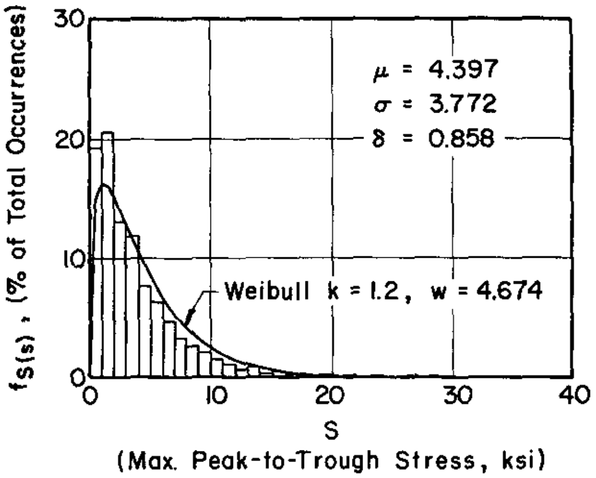


Fig. 6. SL-7 Scratch Gage Data with Corresponding Weibull Distribution (16)

To obtain Weibull distributions that represent ship loading stress histories, the mean, the standard deviation and the coefficient of variation for the distributions are made equal to the corresponding values for the ship data. First, the appropriate value of the Weibull distribution parameter k can be obtained from Table IV by making the coefficient of variation for the Weibull distribution (standard deviation/mean) equal to the coefficient of variation for the histrograms. Next, the value of parameter w is obtained from the following:

$$w = \frac{\mu_S}{\Gamma(1 + 1/k)} \qquad (5)$$

TABLE IV

Table of Weibull Shape Parameter Values and Corresponding Coefficients of Variation

Weibull Shape Parameter k	Coefficient of Variation δ
0.5	2.236
0.6	1.758
0.7	1.462
0.8	1.261
0.9	1.113
1.0	1.000
1.1	0.910
1.2	0.837
1.3	0.776
1.4	0.724
1.5	0.679
1.6	0.640
1.7	0.605
1.8	0.575
1.9	0.547
2.0	0.523

The above procedure has been followed to establish values of k for a variety of ships for which loading history data are available. These values are given in Table V and range from 0.7 to 1.3. The shapes of the functions are presented in Fig. 7 and are similar to those for the ships shown in Fig. 4. It should be noted that the larger ships (tankers and bulk carriers) tend to have loading shape parameters equal to or greater than 1.0.

For the Weibull distribution functions, the maximum stress-range expected to be reached once in a ship's lifetime of 10^8 cycles (estimated to be a 20 year life) can be determined from the following equation:

$$S_{10^8} = w\,[\ln(52000)]^{1/k} \qquad (6)$$

The predicted values of S_{10^8} for the ships for which loading history data are available are given in Table V and range from 10 ksi to 34.1 ksi.

Random Load Factor

In the fatigue design procedure presented herein the constant-cycle fatigue stress range in Eq. 1 must be modified to account for the random loading represented by the Weibull distribution functions. This can be done readily and simply by introducing a random load factor, ξ (16).

The random load factor is based on the assumption that the stress range history can be modeled by the two-parameter Weibull distribution function with a maximum stress range equal to S_{10^8} and a minimum stress range equal to zero and is given by:

$$\xi = (\ln N)^{1/k}\,[\Gamma(1 + m/k)]^{-1/m} \qquad (7)$$

This factor, as developed by Ang and Munse (16), is based on the application of Miner's linear damage hypothesis to the random loading represented by the Weibull probability density function of Eq. 2. A tabulation of the random load factors for various values of k and m are presented in Table VI for a life of 10^8 cycles. (A comparable table could readily be prepared for any other life.)

Using the random stress factor from Eq. 7 (Table VI) (a factor that corresponds to a specific Weibull distribution and S-N curve slope), and the mean fatigue stress range from Table I for specific detail, one can readily obtain the maximum stress of the variable loading that can be expected to produce failure at 10^8 cycles. This maximum stress is:

$$(S_{max})_N = S_N \cdot \xi \qquad (8)$$

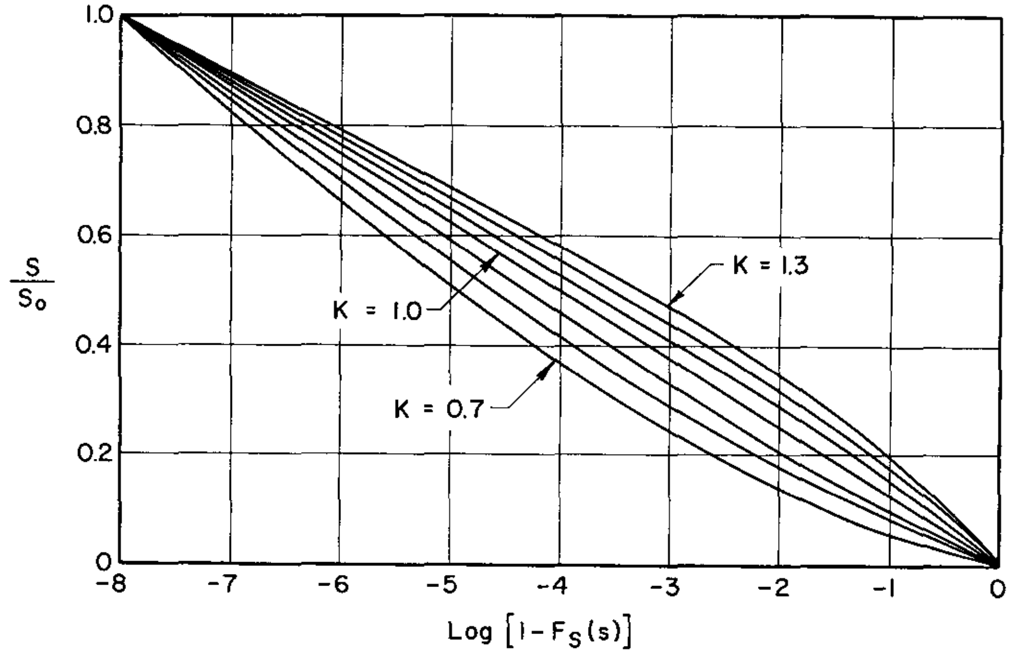


Fig. 7. Ship Loading Histories Modeled by Weibull Distributions

TABLE V

Ship Loading Histories Compared With Weibull Distributions

Type of Ship	Name of Ship	Notes	Weibull Load Shape, k	Stress Change at Probability of Exceedance $= 10^8, S_{10^8}$ (ksi)
Dry Cargo	Wolverine State	1,5	1.2	16.5
	California State	1,5	1.0	18.0
	Mormacscan	1,5,7	1.3	12.0
	Mormacscan	1,5,8	1.0	10.0
Large Tankers	Idemitsu Maru	2,5	1.0	12.3
	R. G. Follis	2,5	0.8	30.0
	Esso Malaysia	2,5	0.8	21.8
	Universe Ireland	2,3,5,	0.7	18.7
Bulk Carrier	Fotini L.	2,5	0.9	29.5
SL-7 Containerships	See Note 9	4,6,9	1.2	34.1

Notes:

1) Data from ref. (14)
2) Data from ref. (20)
3) Data from ref. (21)
4) Data from ref. (15)
5) Load history is for wave-induced loading with dynamic effects filtered.
6) Load history is for wave-induced loading with dynamic effect included.
7) Load history based on North Atlantic voyages.
8) Load history based on South American voyages.
9) Load history based on data collected from eight SL-7 containerships.

TABLE VI

Random Load Factors

m	Random Load Factors, ξ						
	k=0.7	k=0.8	k=0.9	k=1.0	k=1.1	k=1.2	k=1.3
2.0	28.63	20.93	16.17	13.02	10.83	9.24	8.05
2.5	23.12	17.49	13.86	11.39	9.63	8.33	7.33
3.0	19.23	14.96	12.12	10.14	8.69	7.60	6.75
3.5	16.35	13.04	10.77	9.14	7.93	7.00	6.27
4.0	14.15	11.53	9.68	8.32	7.30	6.50	5.86
4.5	12.41	10.31	8.79	7.64	6.76	6.07	5.52
5.0	11.01	9.31	8.04	7.07	6.31	5.71	5.21
5.5	9.87	8.48	7.41	6.58	5.92	5.39	4.95
6.0	8.91	7.77	6.87	6.15	5.58	5.10	4.71
6.5	8.11	7.16	6.40	5.78	5.27	4.85	4.50
7.0	7.42	6.64	5.99	5.45	5.00	4.63	4.31
7.5	6.83	6.18	5.62	5.16	4.76	4.43	4.14
8.0	6.31	5.78	5.30	4.89	4.54	4.24	3.98
8.5	5.86	5.42	5.01	4.66	4.35	4.08	3.84
9.0	5.46	5.10	4.75	4.44	4.17	3.92	3.71
9.5	5.11	4.81	4.52	4.25	4.00	3.78	3.59
10.0	4.79	4.55	4.30	4.07	3.85	3.65	3.48

Values are based on a life of 10^8 cycles. For any other life N, the values in this table would be multiplied by:

$$\frac{(\ln N)^{1/k}}{(18.42)^{1/k}}$$

The term S_N is the mean constant-cycle stress range from Table I that would be expected to produce a fatigue failure in the selected detail at N cycles. However, the value so determined is the mean value (50 percent level of reliability) and must be modified by a reliability factor (factor of safety) to obtain an appropriate maximum design stress value.

Reliability Factor (Factor of Safety)

The basic relationships for the fatigue reliability design criteria developed herein have been presented by Ang (17) in his development of reliability analysis for design. In this analysis it is recognized that the fatigue life of a structural detail is a random variable and assumed that its distribution can be represented by an approximation to the Weibull distribution (18). On this basis, the reliability function (the probability of no failure through a life n) can be given by:

$$L(n) = \exp\left\{-\left[\frac{n}{N}\,\Gamma(1+\Omega_N^{1.08})\right]^{\Omega_N^{-1.08}}\right\} \quad (9)$$

where:

N = mean life to fatigue failure

Ω_N = total uncertainty in fatigue life

The total uncertainty is a function of the uncertainty in the mean fatigue strength (Table I), the uncertainty in the stress analysis, effects of fabrication, workmanship, corrosion, etc. The uncertainty in the mean fatigue strength (standard deviation divided by the mean) varies considerably from one detail to the next but averages about 0.50 (from 0.2 to 0.8 depending upon the detail, the amount of data, etc.) for the details in Table I. Relatively little is known concerning the other factors. Nevertheless, until better values can be established it is recommended that an estimated value of total uncertainty of 0.80 be used.

To utilize the reliability function in design, the mean life N necessary to produce a useful life n with a reliability of L(n) is obtained from from the following:

$$N = n\gamma_L \tag{10}$$

where γ_L is the scatter factor and is given by equation 11 (Fig. 8).

$$\gamma_L = \frac{\Gamma(1 + \Omega_N^{1.08})}{(p_F)^{\Omega_N^{1.08}}} \tag{11}$$

where:
(p_F) = the probability of failure and equal to $[1 - L(n)]$.

In Fig. 8 it can be seen that a 99 percent level of reliability (one percent probability of failure), and an uncertainty in life $\Omega_N = 0.50$, would require a design for a life N equal to about 8 times the expected useful life n.

Under constant-cycle stress range, from equation (1a), the required design

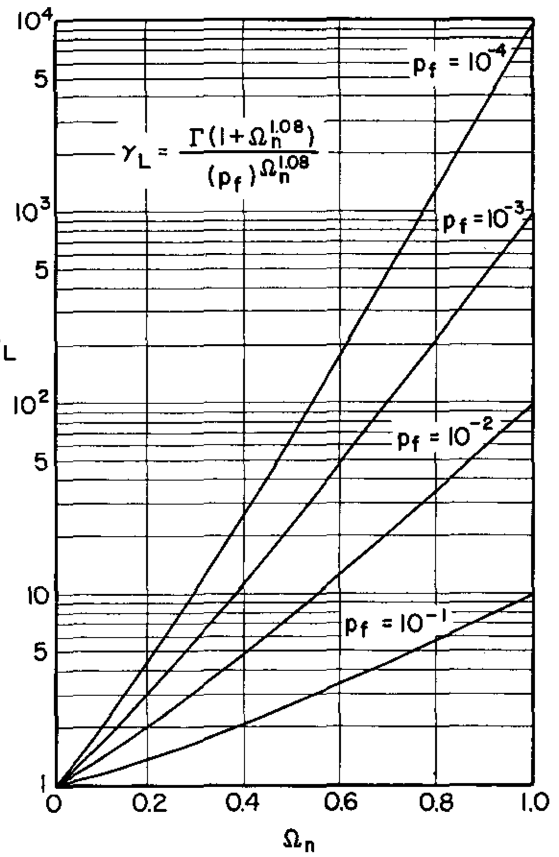


Fig. 8. Relationship Between the Scatter Factor and the Uncertainty in Fatigue Life and Various Probabilities of Failure (17)

stress for a specified level of reliability will be given by:

$$S_D = (\frac{C}{N})^{1/m} \tag{12}$$

or

$$S_D = (\frac{C}{n})^{1/m} \cdot (\frac{1}{\gamma_L})^{1/m} \tag{12a}$$

where:
S_D = constant cycle design stress range for a useful life n and a specified reliability L(n).

Designating the last term of equation (12a) as the 'Reliability Factor', R_F,

$$R_F = (\frac{1}{\gamma_L})^{1/m} = \left\{\frac{(p_F)^{\Omega_N^{1.08}}}{\Gamma(1+\Omega_N^{1.08})}\right\}^{1/m} \tag{13}$$

and the allowable constant-cycle design stress becomes,

$$S_D = S_N \cdot R_F \tag{14}$$

The value of R_F for three levels of reliability, various S-N curve slopes, and uncertainties Ω_N equal to 0.4, 0.6 or 0.8 can be obtained from Table VII.

FATIGUE DESIGN

With the relationships presented above a simple fatigue verification or design procedure that takes into account the principal fatigue parameters can now be provided. Table VIII shows the six steps of the procedure to be followed to verify the adequacy of a given ship structure detail in fatigue.

In step 1 the expected loading history for the ship detail must be established (the shape factor for the Weibull distribution selected). Some guidance can be obtained from Table V, a summary of the limited data now available. As more complete data are obtained concerning ship loading histories, the values in Table V can be expanded, improved and updated to provide the designer with the best possible guidance.

In the second step the ship details at which the fatigue resistance should be checked must be identified. The critical locations in the ship assemblies, such as shown in Fig. 3, can be used to identify the critical details in terms of the numerous details shown in Fig. 2. (An extensive summary of details in the twelve families of Table II are presented in the Reference 22 Report).

The third step is to obtain, for the detail, the fatigue strength and slope of the S-N curve (this comes from Table I). The fourth step is to obtain the random load factor from Table VI, and the fifth step is to select the

TABLE VIII

Design Procedure

1.	Ship Loading Distribution	Choose a loading shape parameter k, of the Weibull distribution.
2.	Ship Detail Catalogs	Identify the number designation of the critical details. (Figs. 2 and 3)
3.	Design Table 1	Find: 1) SN curve slope, m, of detail 2) Mean fatigue stress-range, for detail.
4.	Design Table 6	Find random load factor, ξ, based on k-value and m-value.
5.	Design Table 6	Find reliability factor, R_f, based on m-value and Ω_N-value for desired level of reliability.
6.	Design Equation $(S_D)_{all} = S_N \cdot \xi \cdot R_F$	Compute allowable stress-range $(S_D)_{all}$, for probability of exceedance of 10^{-8}.

appropriate reliability factor from Table VII. The maximum allowable fatigue stress range, $(S_D)_{all}$[4], is then obtained from the following equation

$$(S_D) = S_N \cdot \xi \cdot R_F \tag{15}$$

This relationship, using Table VI, is based on a desired life of 10^8 cycles. For any other life the values, as noted on Table VI, would need to be modified.

A design example for a beam bracket with four structural details is presented in Fig. 9. In this instance, the Weibull shape factor for the loading history was taken as 1.0, the total uncertainty in fatigue life as 0.80 and the desired level of reliability was assumed to be equal to 90 percent. The resulting maximum allowable fatigue stress range at the deck-bulkhead intersection (detail 39B) is found to be 31.8 ksi. This maximum stress-range provides only for fatigue; in addition, the maximum stress must not exceed the nominal permissible design stress provided by the basic design rules (19). For detail number 7, at the toe of the stiffener weld in the web and flange, the maximum allowable fatigue stress range is 39.7 ksi for a 20 year life. Similar calculations can readily be made for details number 37 and 38.

[4]This maximum allowable stress range at the point in question, is the maximum peak-to-trough stress range expected under the most severe sea state and during the entire life of the structure.

SUMMARY AND CONCLUSIONS:

This paper summarizes a simple design procedure that has been developed to provide for a fatigue strength verification in ship design. The criteria provides for:

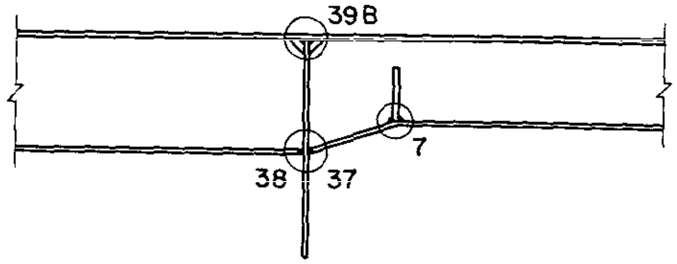
(a) A large variety in ship structure details.

(b) The basic fatigue resistance of the numerous welded details.

(c) A distribution function that can be used to represent the long life loading (10^8 cycles-20 years) for various types of ships.

Fig. 9. Design Example

Ship Assembly 1A1 – Beam Bracket (Fig. 3a)



Detail No. (See Fig. 2)	$(S)_{10^8}$ (See Table 1)	m
7	7.41	4.11
39B	5.9 Est.	4.0 Est.
37	5.0 Est.	3.7 Est.
38	4.97	3.71

Reliability – 90%
Coef. of Var. – 0.80
Weibull Dist. k = 1.0

Considering Deck Plate – Detail 39B

$(S)_{10^8}$ = 5.9 ksi Est. m = 4.0 Est. (Table 1)
Reliability Factor = 0.648 (Table 7)
Random Load Factor = 8.32 (Table 6)
Max. Allowable Stress Range = 5.9 x 0.648 x 8.32 = 31.8 ksi

Considering Stiffener Detail – Detail 7

$(S)_{10^8}$ = 7.41 ksi m = 4.11
R_F = 0.655 ξ = 8.17
Max. Allowable Stress Range = 7.41 x 0.655 x 8.17 = 39.7 ksi

TABLE VII

Reliability Factors

m	Reliability Factors, R_F								
	R = 90%			R = 95%			R = 99%		
	Ω_N			Ω_N			Ω_N		
	0.40	0.60	0.80	0.40	0.60	0.80	0.40	0.60	0.80
2.0	.691	.546	.420	.608	.447	.320	.451	.281	.170
2.5	.744	.616	.500	.671	.525	.402	.528	.362	.242
3.0	.782	.668	.561	.717	.585	.468	.588	.429	.307
3.5	.810	.708	.609	.752	.631	.521	.634	.484	.363
4.0	.831	.739	.648	.780	.669	.566	.671	.530	.412
4.5	.849	.764	.680	.801	.699	.603	.702	.569	.455
5.0	.863	.785	.707	.819	.725	.634	.727	.602	.492
5.5	.874	.802	.730	.834	.746	.661	.748	.630	.525
6.0	.884	.817	.749	.847	.765	.684	.767	.655	.554
6.5	.893	.830	.766	.858	.781	.705	.782	.677	.580
7.0	.910	.841	.781	.867	.795	.722	.796	.696	.603
7.5	.906	.851	.794	.876	.807	.738	.808	.713	.623
8.0	.912	.860	.805	.883	.818	.752	.819	.728	.642
8.5	.917	.867	.815	.889	.827	.765	.829	.742	.659
9.0	.921	.874	.825	.895	.836	.776	.838	.754	.675

(d) A random loading factor that accounts for the randomness of the loading during the life of the structure.

(e) A reliability factor (Factor of Safety) that accounts for the many uncertainties that exist.

The values of maximum allowable fatigue stress range obtained in the design example provides an excellent calibration of the procedure and is considered very reasonable, based on the history of such details in the ships at sea. Additional evaluations now should be made of those details at which fatigue failures have developed to further evaluate and calibrate the procedure. After the procedure has been further verified, it should be possible also to use the procedure to develop relative fatigue ratings for the many details used for ship structures.

The opinions herein are those of the author and not necessarily those of the Advisory Committee of the Ship Structure Committee under whose guidance the investigation on which this paper is based was conducted.

REFERENCES

1. Vedeler, G., "To What Extent Do Brittle Fracture and Fatigue Interest Shipbuilders Today," Houdremont Lecture 1962, Sveiseteknikk 1962, No. 3.

2. Glasfeld, R., Jordan, D., Ken, M., Jr., and Zoller, D., "Review of Ship Structure Details," SSC-266, 1977.

3. Jordan, C. R., Cochran, C. S., "In-Service Performance of Structural Details," SSC-272, 1978.

4. Jordan, C. R. and Knight, L. T., "Further Survey of In-Service Performance of Structural Details," SSC-294, 1980.

5. Munse, W. H., Fatigue of Welded Steel Structures, Welding Research Council, New York (1964).

6. Gurney, T. R., Fatigue of Welded Structures, Cambridge U. Press, England (1968).

7. B.W.R.A., "Symposium on the Fatigue of Welded Structures, March 29-April 1, 1960," British Welding J. (March-Sept. 1960).

8. Munse, W. H., Stallmeyer, J. E., and Drew, F. P., "Structural Fatigue and Steel Railroad Bridges," Proc., AREA Seminar (1968).

9. "Symposium on Structural Fatigue," J. of the Structural Div., ASCE (Dec. 1965) 44, No. ST12, 2663-2797.

10. The Welding Institute, "Proceedings of the Conference on Fatigue of Welded Structures," July 6-9, 1970, The Welding Institute, Cambridge, England (1971).

11. Pollard, B. and Cover, R. J., "Fatigue of Steel Weldments," Welding J., American Welding Soc. (Nov. 1972) 51, No. 11, 544s-554s.

12. Munse, W. H., Pierce, R. C., Martin, S. A., Mayhew, L., and Mahini, B., "Basic Material Fatigue Properties for Use in Freight Car Fatigue Analysis," Civil Engineering Department, University of Illinois at Urbana-Champaign, March 1981.

13. Bridge, I. C., "Structural Stress in an Oil Tanker under Service Conditions," Trans., I.N.A., p. 161, 1938.

14. Hoffman, D. and Lewis, E. V., "Analysis and Interpretation of Full-Scale Data on Midship Bending Stresses of Dry Cargo Ships," SSC-196, June 1969.

15. Fain, R. A. and Booth, E. T., "Results of the First Five 'Data Years' of Extreme Stress Scratch Gauge Data Collected Aboard Sea-Land's SL-7's," SSC-286, March 1979.

16. Ang, A. H-S and Munse, W. H., "Practical Reliability Basis for Structural Fatigue," Preprint No. 2494 presented at ASCE National Structural Engineering Conference, April 14-18, 1975.

17. Ang, A. H-S., "A Comprehensive Basis for Reliability Analysis and Design," Proc., U. S. - Japan Joint Seminar on Reliability Approach in Structural Engineering, Tokyo, Japan, May 1974.

18. Freudenthal, A. M., "Prediction of Fatigue Failure," J. Appl. Phys. (Dec. 1960) 31, No. 12, 2196-2198.

19. A.B.S., "Rules for Building and Classing Steel Vessels," American Bureau of Shipping, 1979.

20. Little, R. S., Lewis, E. V. and Bailey, F. C., "A Statistical Study of Wave Induced Bending Moments on Large Oceangoing Tankers and Bulk Carriers," Trans. SNAME, 1971.

21. Lewis, E. V. and Zubaly, R. B., "Dynamic Loadings Due to Waves and Ship Motions," SNAME, 1975.

22. Munse, W. H., Investigation at the University of Illinois - Study of Fatigue Characterization of Fabricated Ship Details, Project SR-1257, Ship Structure Committee.