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ADP013493

TITLE: A Comparison of Fatigue Design Methods

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A COMPARISON OF FATIGUE DESIGN METHODS

R. J. Scavuzzo

Professor Emeritus
The University of Akron
Akron, OH 44325-0301

Abstract: There are two basic fatigue-testing methods: fatigue using a maximum cyclic force, the stress-life method, and fatigue using a maximum cyclic strain, the strain-life method. Wőhler first tests to establish a S-N diagram were based on a rotating beam with a constant maximum force loading. Subsequently, the R. R. Moore test that used four-point loading with a constant load was developed and became one of the standard tests. Most machine design textbooks teach fatigue design methods based on these basic tests. Because of research and development activities in the 1960s, cyclic strain methods were developed Low-cycle fatigue analyses based on these new methods have been found to be more accurate in this low-cycle regime. Of course endurance limits of both methods do not change; differences are in predicting low-cycle fatigue life.

This paper presents a short comparison of these two fatigue design methods.

Key Words: Fatigue, Fatigue Analysis, Load Fatigue, Strain Fatigue, Low-cycle Fatigue

Introduction: In the 1950s and 1960s, cyclic thermal stress in nuclear reactors became an object of research. Very high elastic thermal stresses are often calculated in nuclear reactor components because of high temperature gradients. It was recognized that these stresses were fundamentally different from constant load stresses. A small amount of yielding decreased thermal stresses significantly. As a result, elaborate test programs were conducted to thermally cycle components to develop high thermal stresses and initiate fatigue failures. Thermal stresses are like residual stresses; a small amount of yielding relieves these stresses. As thinking matured in this area, researchers realized that the same results could be obtained by mechanical strain cycling specimens rather than trying to simulate the cyclic thermal conditions. Tests could be run in a much shorter time with much less cost and control on the temperature of the specimen and the actual cyclic strains were much more accurate. Thus, data were based on cyclic mechanical strain tests in lieu of cyclic thermal strain tests. Manson [1] and Coffin [2] contributed significantly in these areas. Design procedures for the design of nuclear pressure vessels were developed in the 1960s based on these data. The ASME Boiler and Pressure Vessel Code [3] presents these methods and has expanded the procedures to other pressure vessels besides nuclear pressure vessels. B. F. Langer [4], while at the Bettis Atomic Power Laboratory run by Westinghouse Electric Corporation, contributed significantly to these code procedures.

The endurance limit is the same developed by either test method. Differences in the two methods occur in the low-cycle regime. The cyclic strain method is much more accurate in this area. As a result, fatigue design methods in the nuclear industry as well as the aerospace field and others make use of these cyclic strain procedures. The fact that stress-life design methods, based on constant load data, are usually taught in undergraduate mechanical engineering programs adds to confusion in this area.

A number of more recent textbooks on fatigue cover both load cycling and strain cycling fatigue. Bannantine, Comer and Handrock, all former students of Professor JoDean Morrow at the University of Illinois, cover both methods very well in their text [5]. The first chapter is devoted to the "stress-life" method and the second to the "strain-life" method. Chapter 6 compares these methods. The text by Fuchs and Stephens [6], which is more scientific in approach, is an in depth presentation of many aspects of the strain cyclic method. Collins [7] also presents strain cyclic design methods and is an excellent contribution. However, the third edition of Shigley's mechanical engineering design textbook [8] and Juvinall's textbook [9] only cover the stress-life method. The most recent edition of Shigley's textbook does include some aspects of the strain-life cyclic method.

In this paper, the stress-life method is presented based on References [5,6,7]. The strain-life cyclic design method is taken from the ASME Code [3] that is based on the work B. F. Langer [4] and others. This short review paper cannot treat the subject thoroughly and the reader is referred to References [5,6,7] for additional insight. Fracture mechanics concepts are used to predict fatigue crack growth and finial fracture [5-7]. Crack initiation can be related to the cyclic Von Mises stress [9]. These topics are not considered.

The endurance limit is required in the cycles versus life graph, the S-N curve, of both methods and is reviewed first. Then, the stress-life and strain-life methods are treated.

Endurance Limit: The endurance limit, S_e, is a constant alternating stress below which failure will not occur. Steels have an endurance limit; most nonferrous alloys do not have an endurance limit. The endurance limit of steels can be approximated by the fact that a mirror polished laboratory specimen with a 0.3 inch diameter has a value of about ½ of the ultimate strength, S_u.

$$S_e = \tfrac{1}{2} S_u \qquad (1)$$

Also, since the ultimate strength is related to the Brinell Hardness Number (BHN), the endurance limit can also be approximated with this hardness measurement.

$$S_u \cong 500 \text{ BHN} \qquad (2)$$

and

$$S_e \cong 250 \text{ BHN} \qquad (3)$$

This relationship is depicted graphically in Fig. 1. Note that after the endurance limit

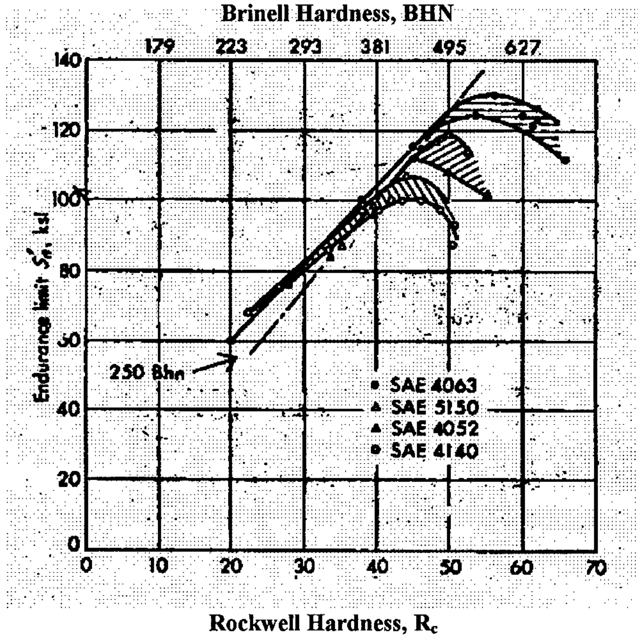


Fig. 1 Relationship between the endurance limit and hardness [9].

reaches 100 ksi the relationship between hardness and the endurance limit is lost for all alloys. The endurance limit can be increased or decreased by the following factors:

1. Surface finish
2. Size Effect
3. Temperature Effect
4. Environment
5. Surface Treatment

Surface Finish: The standard test specimen has a polished finished. Any other surface finish will decrease the endurance limit. For example, a fine ground or commercially polished surface will reduce the endurance limit from 10% to 28% and depends on hardness or ultimate strength. Below an ultimate strength of 140 ksi (280 BHN), the reduction is 10%; for higher hardnesses the decrease reaches 28% of the laboratory specimen. Other surface effects are as follows: machined surface 20% to 50%, hot-rolled from 28% to 78% and as-forged from 45% to 86%. Surface stress concentrations from scratches, pits machining marks, changes in surface strength as well as tensile residual stresses cause these reductions from these various surface finishes.

Size Effect: The diameter of the standard laboratory specimen is 0.3". As the fatigued specimen increases to about 2", the high cycle fatigue strength is deceased. There are two main reasons for this reduction: probability of a defect is increased with size and stress gradients from bending or torsion are decreased allowing more volume to be highly stressed.

Of course, tensile stresses have no gradient and a reduction of 10 % is recommended by Juvinall [9]. A recommended empirical equation [5] is as follows:

$$\text{Csize} = \begin{cases} 1.0 & \text{if } d \leq 0.3\text{in.} \\ 0.869d^{-0.097} & \textit{if } 0.3 \leq d \leq 10.0\,in. \end{cases} \tag{4}$$

Temperature Effects: The ASME Boiler and Pressure Vessel Code does not alter design S-N Curves below 320^0 C (600^0 F). However, other metals such as aluminum decreases significantly above 200^0 C (400^0 F) or less. The fatigue strength of Titanium decreases above room temperature. Thermoplastics are so sensitive to a temperature increase that heating associated with cyclic stresses can significantly reduce fatigue strength. Thus, the cyclic rate must be specified in data and taken into account in design. Metals are not sensitive to the cyclic rate unless rates are in the acoustic range,

Environment: A corrosive environment can reduce the endurance limit to a fraction of its value in air. Water can reduce the endurance limit of carbon and low alloy steel by more than a factor of three. For example, steel SAE 1050 steel with an ultimate strength of 120 ksi, the endurance limit is reduced to 20 ksi in water from about 60 ksi in air [5]. In adverse environments, alloy steels must be used if high endurance limits are required.

Surface Treatments: Surface treatments that increase the strength of the surface or develop compressive residual stresses on the surface or cause both effects can improve high cycle fatigue strength measurably. Helpful surface treatments are shot peening, roll hardening, nitriding or carburizing the surface can, at times, double the endurance limit over the untreated value. On the other hand, decarburization from forging, hot rolling, etc. can decrease the limit by over a factor of two. Grinding and other machining operations develop tensile residual stresses that decrease strength [5,8,9]. Electro plating of steel with hard metal such as chromium or nickel also develops tensile residual and can reduce strengths by over 50% [10].

Stress Concentrations: Stress concentration effects both methods are based on the fatigue strength reduction factor, K_f, defined as follows:

$$K_f = \frac{\textit{Unnotched Fatigue Strength}}{\textit{Notched Fatigue Strength}} \tag{5}$$

This factor is related to the theoretical stress concentration factor, K_t, and the notch sensitivity, q. The notch sensitivity is a function of material, hardness and notch radius or the volume of material affected by the stress concentration. Thus as the notch radius becomes larger, the sensitivity becomes larger. The two design methods differ in one aspect: in the stress-life q is a function or cycles [7,9] whereas in the strain-life method q does not vary with cycles and, therefore, K_f, does not vary with cycles.

$$q = \frac{K_f - 1}{K_t - 1} \tag{6}$$

The notch sensitivity is considered at times to be a material property. Thuw, given q and K_t the fatigue reduction factor can be calculated.

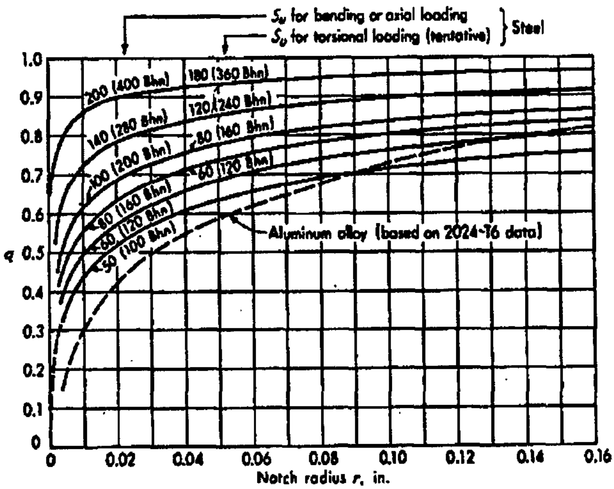


Fig. 2 Notch Sensitivity as a function of hardness, notch radius and

S-N Curve: The cycles to failure curve (S-N Curve) of the stress life method is approximated by using the fact that the cycles to failure at 1,000 cycles is about 90% of the ultimate strength. Using this point and the endurance limit, a straight line on log-log approximates the S-N curve as shown on Fig. 3 where S in the alternating stress..

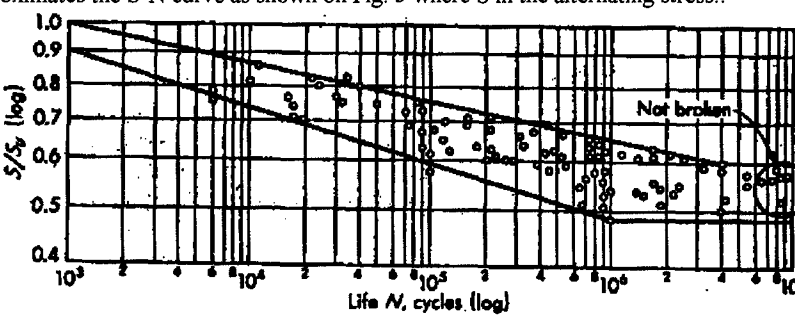


Fig. 3 Stress-life S-N Curve showing Data [8].

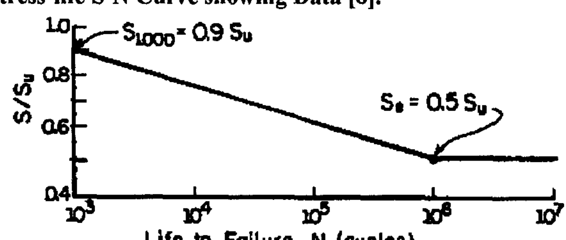
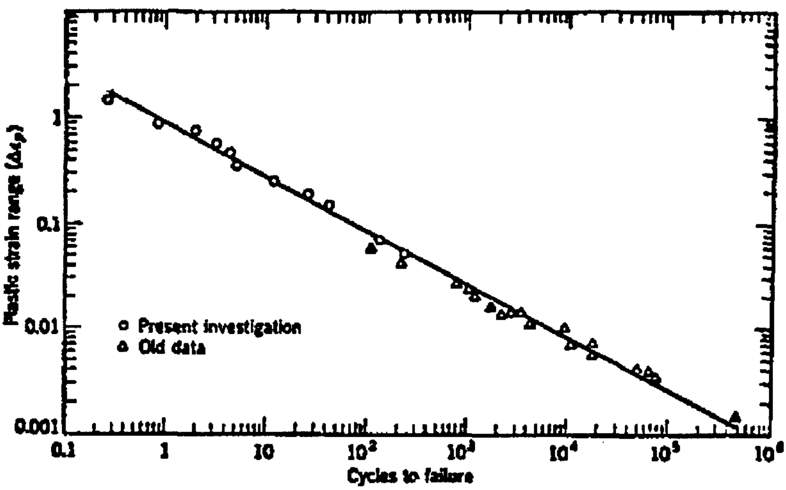


Fig. 4 Approximate S-N Curve for the Stress-Life Method [5].

The S-N curve presented in Fig. 4 is accurate in the cyclic range near the endurance limit ($>10^5$). In the low cycle range results are not accurate and the strain-life method must be used [5]. Mean stresses are used in the Goodman Equation assuming a nominal mean stress. The fatigue strength reduction factor is applied only to the calculated alternating stress, S_{ca},

$$\frac{S_m}{S_u} + \frac{K_f S_{ca}}{S} = \frac{1}{FS} \tag{7}$$

Strain-Life Method: As implied, this design method is based on strain versus cycles data. Fig. 5 is a plot of the plastic strain range (peak-to-peak measurements and not the alternating value which is ½ the range). The slope the curve is about −½; actual slopes will vary with material [2,6,7] but in the procedure developed by Langer a value of −½ is used.



Plastic strain range versus cycles to failure, 2S aluminum.

Fig. 5 Plastic strain range versus cycles to failure.

Langer developed the following equation with this assumption of the plastic portion of the S-N curve has a negative slope of − ½.

$$\Delta \varepsilon_r^p = \frac{C}{\sqrt{N}} \tag{8}$$

where N is the cycles to failure and C is a constant to be evaluated. By assuming that a tensile test is ¼ of a cycle, the cyclic plastic strain of Eq. (8) can be related to the true strain at fracture. The true strain at fracture can be determined from the Reduction in Area, RA in percent, from a tensile test.

$$\bar{\varepsilon}_f = \ln \frac{100}{100 - RA} \tag{9}$$

Substituting into Eq. (9) yields,

$$C = \frac{1}{2}\ln\frac{100}{100-RA} \tag{10}$$

Eq. (8) becomes

$$\Delta\varepsilon_r^p = \frac{1}{2\sqrt{N}}\ln\frac{100}{100-RA} \tag{11}$$

The alternating component of plastic strain is ½ of this value. Langer assumed that the alternating component of elastic strain is the endurance limit divided by the elastic modulus, E. Thus the total alternating strain is as follows:

$$\Delta\varepsilon_a = \frac{1}{4\sqrt{N}}\ln\frac{100}{100-RA} + \frac{S_e}{E} \tag{12}$$

The approach taken in this analysis was that stresses are calculated by elastic analysis. Stresses that exceed the yield point are included and give an approximation to the strain. Thus Eq. (12) is multiplied by the elastic modulus E to obtain units of stress. Analytically developed values are compared to the design curves. Thus, if Eq. (12) is multiplied by the elastic modulus, E, Langer's equation is developed in units of stress. Present FEA codes allow the calculation of strain ranges directly and can be used in lieu of an elastically calculated stress to evaluate life.

$$S_a = \frac{E}{2\sqrt{N}}\ln\frac{100}{100-RA} + S_e \tag{13}$$

ASME Data: Fig. 6 and Fig. 7 are cyclic strain data for low carbon and low alloy steel, respectively. Langer's equation, Eq. (13), is plotted on the graphs. This procedure uses the concept of the worst-case mean stress that is also shown on these two graphs. It is assumed that the highest value of mean stress that can be obtained is the calculated alternating stress, S_{ca}, taken from the yield point, S_y.

$$K_f S_m = S_y - K_f S_{ca} \tag{14}$$

There are three cases to be considered for application to the Goodman equation:

Case 1 $K_f(S_{ca} + S_m) < S_y$ (Stresses are elastic)

$$\frac{K_f S_m}{S_u} + \frac{K_f S_{ca}}{S} = \frac{1}{FS} \tag{15}$$

Case 2 $K_f(S_{ca} + S_m) > S_y$; ; $K_f S_{ca} < S_y$

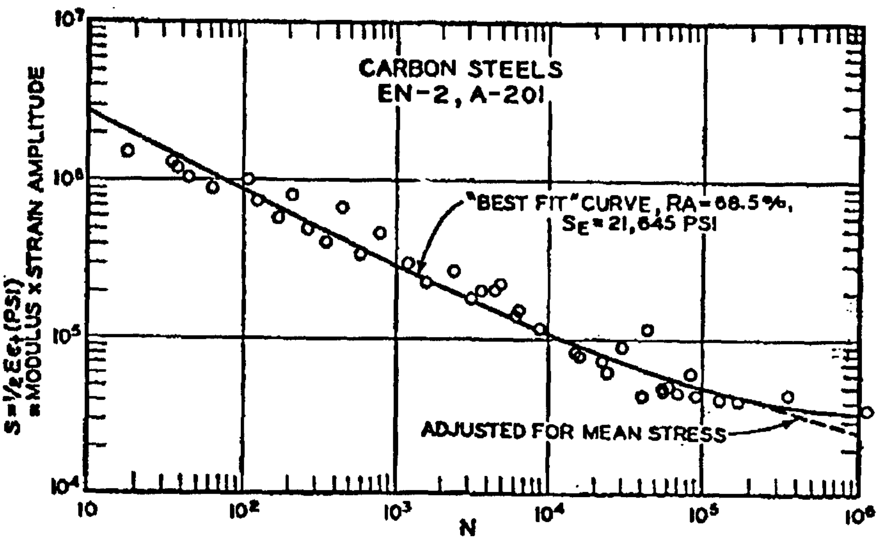


Fig. 6 Alternating Strain Data in Units of Stress for Low Carbon Steel.

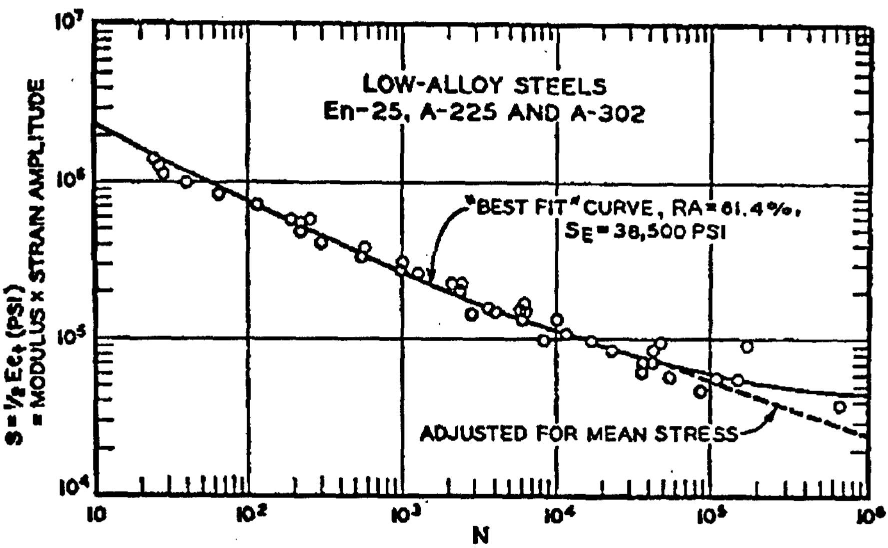


Fig. 7 Alternating Strain Data in Units of Stress for Low-Alloy Steel.

$$\frac{S_y - K_f S_{ca}}{S_u} + \frac{K_f S_{ca}}{S} = \frac{1}{FS} \tag{16}$$

Case 3 $K_f S_{ca} > S_y$ (Cyclic plasticity)

For this case the effective mean stress is zero. The Goodman equation reduces to the following:

$$\frac{K_f S_{ca}}{S} = \frac{1}{FS} \tag{17}$$

Summary: Presented is a short summary of a few differences in the stress-life method versus the strain-life method. First the data in one case is based on the cyclic loads and bending stresses based on the load calculations. In the strain method, cyclic strain data is multiplied by E to have units of stress. However, "stress" values exceed a million psi. Values are strain times the modulus. The effects of stress concentration are treated differently. In one case the usual procedure is to use nominal values of mean stress where in the strain method stress concentrations are included in both the mean and alternating components. In the stress-life method K_f is a function of cycles; in the strain-life method, K_f is constant with cycles. In this short paper many factors had to be omitted and the reader is referred to the referenced material for addition insight.

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