

	<b><i>FUNDAMENTALS OF METAL FATIGUE ANALYSIS</i></b>

Julie A. Bannantine, Ph.D.

*University of Illinois*

Jess J. Comer, Ph.D.

*South Dakota School of Mines*

James L. Handrock, Ph.D.

*University of Illinois*

*Currently at Sandia National Labs.*



Prentice Hall  
Englewood Cliffs, New Jersey 07632

# STRESS-LIFE

## 1.1 INTRODUCTION

The stress-life,  $S-N$ , method was the first approach used in an attempt to understand and quantify metal fatigue. It was the standard fatigue design method for almost 100 years. The  $S-N$  approach is still widely used in design applications where the applied stress is primarily within the elastic range of the material and the resultant lives (cycles to failure) are long, such as power transmission shafts. The stress-life method does not work well in low-cycle applications, where the applied strains have a significant plastic component. In this range a strain-based approach (Chapter 2) is more appropriate. The dividing line between low and high cycle fatigue depends on the material being considered, but usually falls between  $10^4$  and  $10^5$  cycles.

## 1.2 $S-N$ DIAGRAM

The basis of the stress-life method is the Wöhler or  $S-N$  diagram, which is a plot of alternating stress,  $S$ , versus cycles to failure,  $N$ . The most common procedure for generating the  $S-N$  data is the rotating bending test. One example is the R. R. Moore test, which uses four-point loading to apply a constant moment to a rotating (1750 rpm) cylindrical hourglass-shaped specimen. This loading produces a fully reversed uniaxial state of stress. The specimen is mirror polished with a typical diameter in the test section of 0.25 to 0.3 in. The stress level at the surface of the specimen is calculated using the elastic beam equation ( $S = Mc/I$ ) even if the resulting value exceeds the yield strength of the material.

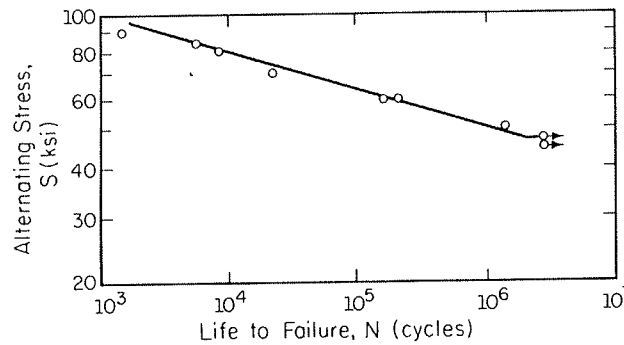


Figure 1.1  $S-N$  curve for 1045 steel.

One of the major drawbacks of the stress-life approach is that it ignores true stress-strain behavior and treats all strains as elastic. This may be significant since the initiation of fatigue cracks is caused by plastic deformation (i.e., to-and-fro slip). The simplifying assumptions of the  $S-N$  approach are valid only if the plastic strains are small. At long lives most steels have only a small component of cyclic strain which is plastic (in some cases it is effectively too small to measure) and the  $S-N$  approach is valid.

$S-N$  test data are usually presented on a log-log plot, with the actual  $S-N$  line representing the mean of the data. Certain materials, primarily body-centered cubic (BCC) steels, have an endurance or fatigue limit,  $S_e$ , which is a stress level below which the material has an "infinite" life (see Fig. 1.1). For engineering purposes, this "infinite" life is usually considered to be 1 million cycles. The endurance limit is due to interstitial elements, such as carbon or nitrogen in iron, which pin dislocations. This prevents the slip mechanism that leads to the formation of microcracks. Care must be taken when using the endurance limit since it can disappear due to:

1. Periodic overloads (which unpin dislocations)
2. Corrosive environments (due to fatigue corrosion interaction)
3. High temperatures (which mobilize dislocations)

It should be pointed out that the effect of periodic overloads mentioned above relates to smooth specimens. Notched components may have completely different behavior, due to the residual stresses set up by overloads. This is discussed more fully in Section 1.4.4.

Most nonferrous alloys have no endurance limit and the  $S-N$  line has a continuous slope (Fig. 1.2). A pseudo-endurance limit or fatigue strength for these materials is taken as the stress value corresponding to a life of  $5 \times 10^8$  cycles.

There are certain general empirical relationships between the fatigue properties of steel and the less expensively obtained monotonic tension and hardness properties. When the  $S-N$  curves for several steel alloys are plotted in

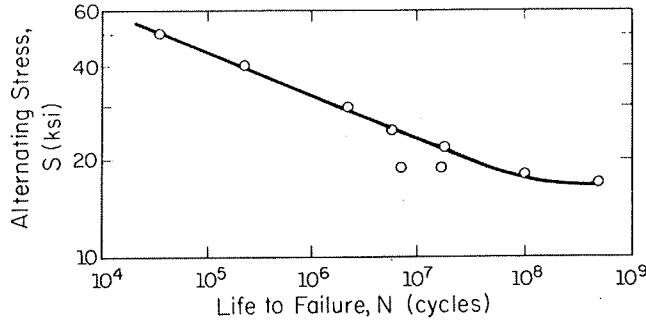


Figure 1.2 S-N curve for 2024-T4 aluminum. (Data from Ref. 1.)

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non-dimensional form using the ultimate strength, they tend to follow the same curve (Fig. 1.3).

The ratio of endurance limit to ultimate strength for a given material is the fatigue ratio (Fig. 1.4). Most steels with an ultimate strength below 200 ksi have a fatigue ratio of 0.5. It should be noted that this ratio can range from 0.35 to 0.6. Steels with an ultimate strength over 200 ksi often have carbide inclusions formed during the tempering of martensite. These nonmetallic inclusions serve as crack initiation points, which effectively reduce the endurance limit.

Using a rule of thumb relating hardness and ultimate strength [ $S_u(\text{ksi}) \approx 0.5 \times \text{BHN}$ ], the following relationships for steel can be given:

Endurance limit related to hardness:

$$S_e(\text{ksi}) \approx 0.25 \times \text{BHN} \quad \text{for BHN} \leq 400 \quad \text{if BHN} > 400, S_e \approx 100 \text{ ksi} \quad (1.1)$$

where BHN is the Brinell hardness number.

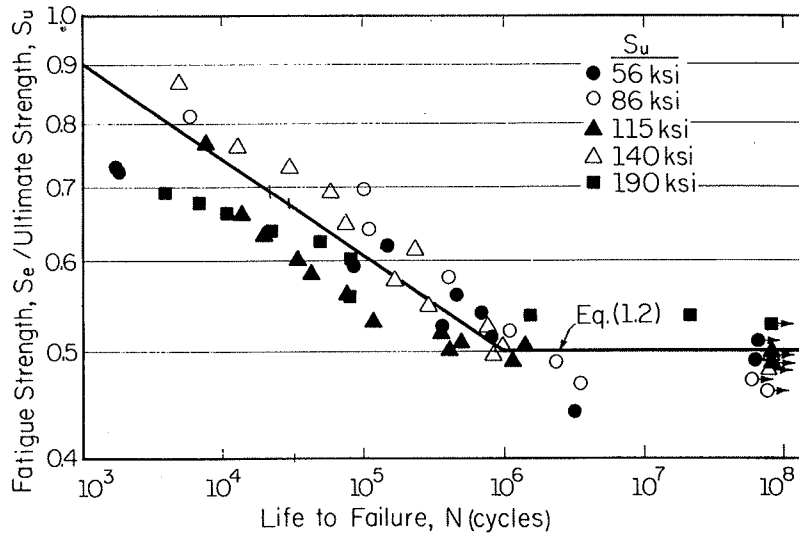
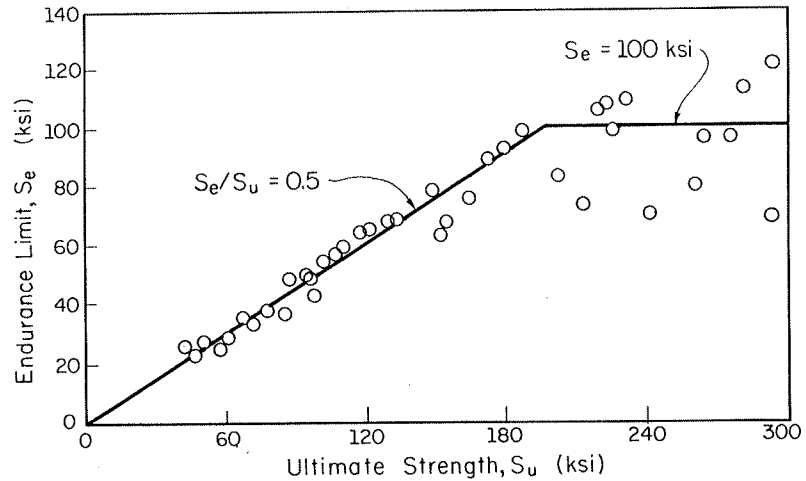


Figure 1.3 S-N curves for several wrought steels, plotted in ratio form ( $S_e/S_u$ ).



**Figure 1.4** Relation between rotating bending endurance limit and tensile strength of wrought steels.

Endurance limit related to ultimate strength:

$$S_e \approx 0.5 \times S_u \quad \text{for } S_u \leq 200 \text{ ksi} \quad \text{if } S_u > 200 \text{ ksi, } S_e \approx 100 \text{ ksi} \quad (1.2)$$

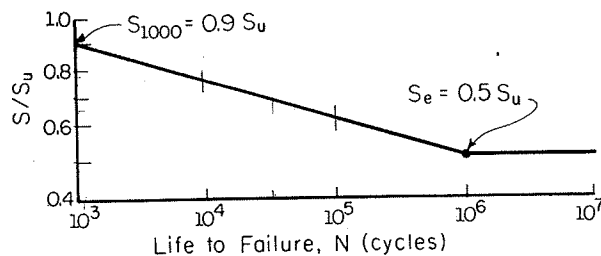
The alternating stress level corresponding to a life of 1000 cycles,  $S_{1000}$ , can be estimated as 0.9 times the ultimate strength. The line connecting this point and the endurance limit is the estimate used for the  $S$ - $N$  design line (Fig. 1.5) if no actual fatigue data are available for the material.

In place of the graphical approach shown above, a power relationship can be used to estimate the  $S$ - $N$  curve for steel:

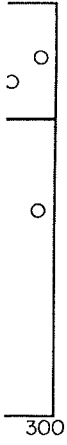
$$S = 10^C N^b \quad (\text{for } 10^3 < N < 10^6) \quad (1.3)$$

where the exponents,  $C$  and  $b$ , of the  $S$ - $N$  curve are determined using the two defined points shown in Fig. 1.5:

$$b = -\frac{1}{3} \log_{10} \frac{S_{1000}}{S_e} \quad C = \log_{10} \frac{(S_{1000})^2}{S_e} \quad (1.4)$$



**Figure 1.5** Generalized  $S$ - $N$  curve for wrought steels on log-log plot.



The equation giving life in terms of an alternating stress is

$$N = 10^{-C/b} S^{1/b} \quad (\text{for } 10^3 < N < 10^6) \tag{1.5}$$

Note that when the estimates for  $S_{1000}$  and  $S_e$  are made,

$$S_{1000} \approx 0.9S_u \quad \text{and} \quad S_e \approx 0.5S_u \tag{1.6}$$

The  $S-N$  curve is defined as

$$S = 1.62S_u N^{-0.085} \tag{1.7}$$

Similar empirical relationships for materials other than steel are not as clearly defined.

Before continuing, certain points about the  $S-N$  curve should be emphasized:

1. The empirical relationships outlined in this section are only estimates. Depending on the level of certainty required in the fatigue analysis, actual test data may be necessary.
2. The most useful concept of the  $S-N$  method is the endurance limit, which is used in infinite-life or "safe stress" designs.
3. In general, the  $S-N$  approach should not be used to estimate lives below 1000 cycles.

Regarding point 3, although there are several methods used to estimate the  $S-N$  curve in the range 1 to 1000 cycles they are not recommended. These methods use some percentage of ultimate strength,  $S_u$ , or true fracture stress,  $\sigma_f$ , as the estimate for alternating stress at either 1 or  $\frac{1}{4}$  cycle. One of the main problems in using this approach is that most materials have an  $S-N$  curve which is very flat in the low cycle region. This is due to the large plastic strains caused by high load levels. When doing low cycle analysis a strain-based approach is more appropriate.

### 1.3 MEAN STRESS EFFECTS

The following relationships and definitions are used when discussing mean and alternating stresses (Fig. 1.6):

$$\Delta\sigma = \sigma_{\max} - \sigma_{\min} = \text{stress range}$$

$$\sigma_a = \frac{\sigma_{\max} - \sigma_{\min}}{2} = \text{stress amplitude}$$

$$\sigma_m = \frac{\sigma_{\max} + \sigma_{\min}}{2} = \text{mean stress}$$

$$R = \frac{\sigma_{\min}}{\sigma_{\max}} = \text{stress ratio} \quad A = \frac{\sigma_a}{\sigma_m} = \text{amplitude ratio}$$

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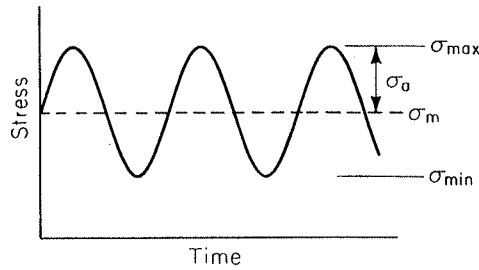


Figure 1.6 Terminology for alternating stress.

The  $R$  and  $A$  values corresponding to several common loading situations are:

$$\text{Fully reversed: } R = -1 \quad A = \infty$$

$$\text{Zero to max: } R = 0 \quad A = 1$$

$$\text{Zero to min: } R = \infty \quad A = -1$$

The results of a fatigue test using a nonzero mean stress are plotted on a Haigh diagram (alternating stress versus mean stress) with lines of constant life drawn through the data points (Fig. 1.7). This diagram is sometimes incorrectly called the modified Goodman diagram. The data can also be plotted on a master diagram (Fig. 1.8) which has an extra set of axes for maximum and minimum stress.

Since the tests required to generate a Haigh diagram can be expensive, several empirical relationships have been developed to generate the line defining the infinite-life design region. These methods use various curves to connect the endurance limit on the alternating stress axis to either the yield strength,  $S_y$ , ultimate strength,  $S_u$ , or true fracture stress,  $\sigma_f$ , on the mean stress axis.

The following relationships are commonly used and are shown on Fig. 1.9:

$$\text{Soderberg (USA, 1930): } \frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_y} = 1 \quad (1.8)$$

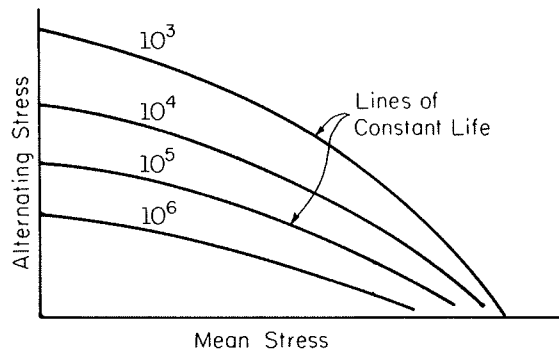


Figure 1.7 Haigh diagram.

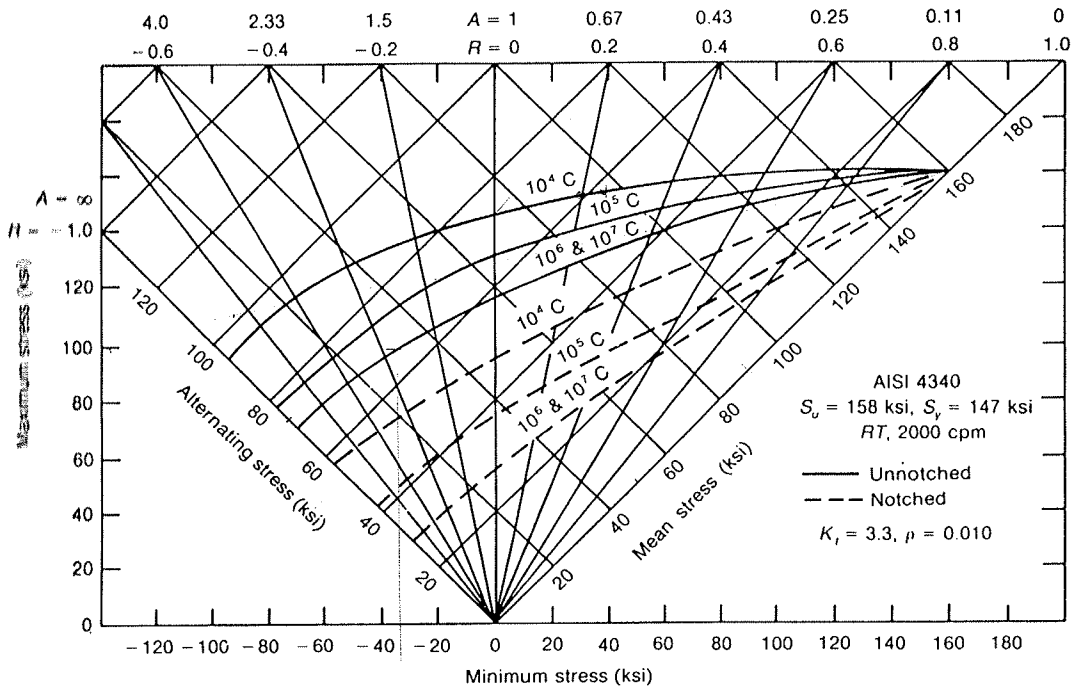


Figure 1.8 Master diagram for AISI 4340 steel. (From Ref. 2.)

Goodman (England, 1899):  $\frac{\sigma_a}{S_e} + \frac{\sigma_m}{S_u} = 1$  (1.9)

Gerber (Germany, 1874):  $\frac{\sigma_a}{S_e} + \left(\frac{\sigma_m}{S_u}\right)^2 = 1$  (1.10)

Morrow (USA, 1960s):  $\frac{\sigma_a}{S_e} + \frac{\sigma_m}{\sigma_f} = 1$  (1.11)

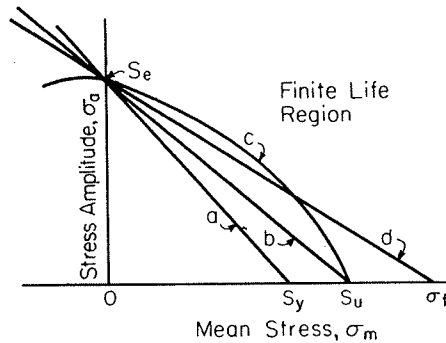


Figure 1.9 Comparison of mean stress equations (a. Soderberg, b. Goodman, c. Gerber, d. Morrow).

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The following generalizations can be made when discussing cases of tensile mean stress:

1. The Soderberg method is very conservative and seldom used.
2. Actual test data tend to fall between the Goodman and Gerber curves.
3. For hard steels (i.e., brittle), where the ultimate strength approaches the true fracture stress, the Morrow and Goodman lines are essentially the same. For ductile steels ( $\sigma_f > S_u$ ) the Morrow line predicts less sensitivity to mean stress.
4. For most fatigue design situations,  $R < 1$  (i.e., small mean stress in relation to alternating stress), there is little difference in the theories.
5. In the range where the theories show a large difference (i.e.,  $R$  values approaching 1), there is little experimental data. In this region the yield criterion may set design limits.

For finite-life calculations the endurance limit in any of the equations can be replaced with a fully reversed alternating stress level corresponding to that finite-life value.

#### Example 1.1

A component undergoes a cyclic stress with a maximum value of 110 ksi and a minimum value of 10 ksi. The component is made from a steel with an ultimate strength,  $S_u$ , of 150 ksi, an endurance limit,  $S_e$ , of 60 ksi, and a fully reversed stress at 1000 cycles,  $S_{1000}$ , of 110 ksi. Using the Goodman relationship, determine the life of the component.

**Solution** Determine the stress amplitude and mean stress.

$$\sigma_a = \frac{\sigma_{\max} - \sigma_{\min}}{2} = \frac{110 - 10}{2} = 50 \text{ ksi}$$

$$\sigma_m = \frac{\sigma_{\max} + \sigma_{\min}}{2} = \frac{110 + 10}{2} = 60 \text{ ksi}$$

Generate a Haigh diagram with constant life lines at  $10^6$  and  $10^3$  cycles. These lines are constructed by connecting the endurance limit,  $S_e$ , and  $S_{1000}$  values on the alternating stress axis to the ultimate strength,  $S_u$ , on the mean stress axis (see Fig. E1.1).

When the stress conditions for the component ( $\sigma_a = 50$  ksi,  $\sigma_m = 60$  ksi) are plotted on the Haigh diagram, the point falls between the  $10^3$  and  $10^6$  life lines. This indicates that the component will have a finite life, but the life is greater than 1000 cycles. Next, a line is drawn through the point representing the stress conditions and the ultimate strength,  $S_u$ , on the mean stress axis. This represents a constant life line at a life equal to the life of the component. This line intersects the fully reversed alternating stress axis at a value of 83 ksi. Note that this value could also be obtained by solving a modification of Eq. (1.9):

$$\frac{\sigma_a}{S_n} + \frac{\sigma_m}{S_u} = 1$$

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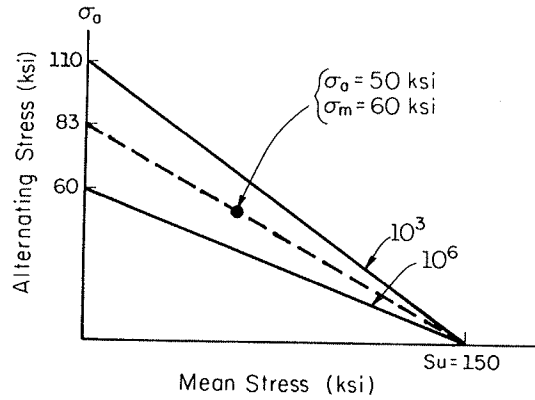


Figure E1.1 Haigh diagram.

where  $S_n$  is the fully reversed stress level corresponding to the same life as that obtained with the stress conditions  $\sigma_a$  and  $\sigma_m$ . For this problem,

$$\frac{50}{S_n} + \frac{60}{150} = 1$$

$$S_n = 83 \text{ ksi}$$

The value for  $S_n$  can now be entered on the  $S-N$  diagram (Fig. E1.2) to determine the life of the component,  $N_f$ . (Recall that the  $S-N$  diagram represents fully reversed loading). When a value of 83 ksi is entered on the  $S-N$  diagram for the material used for the component, the resulting life to failure,  $N_f$ , is  $2.4 \times 10^4$  cycles.

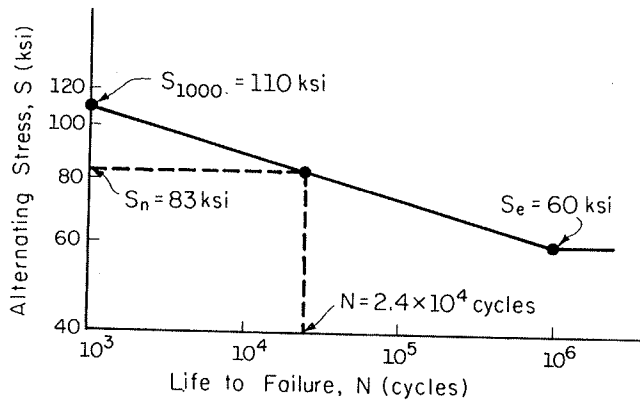
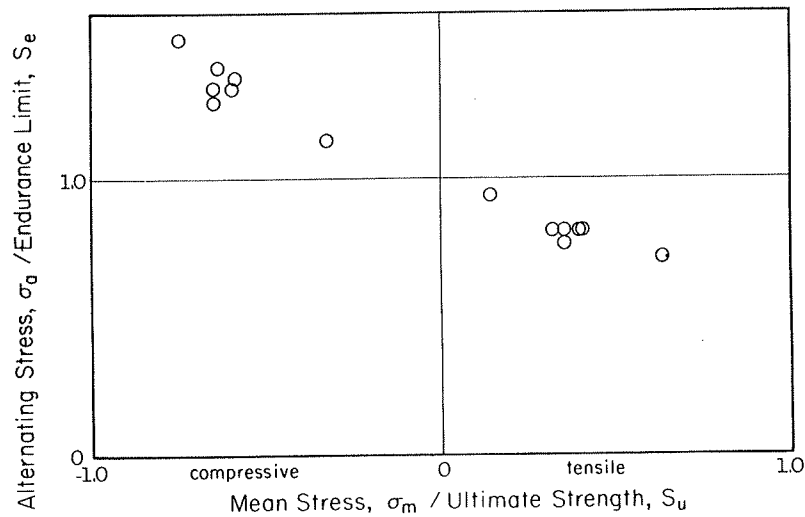


Figure E1.2  $S-N$  diagram.

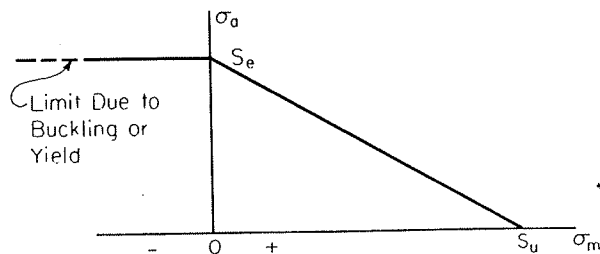
This problem could be redone using the Gerber [Eq. (1.10)], Soderberg [Eq. (1.8)], and Morrow [Eq. (1.11)] mean stress equations. Each technique would provide a slightly different life estimate.



**Figure 1.10** Compressive and tensile mean stress effect for smooth specimen. (Data from Ref. 3.)

As seen in Fig. 1.9, the three linear models predict that compressive mean stresses are very beneficial and allow for very large alternating stresses. Experimental results from smooth specimens do indeed indicate that a compressive mean stress is beneficial and increases the life at a given alternating stress (Fig. 1.10). There is difficulty in relating this behavior to notched components. The problems arise when trying to predict the residual stresses generated near the notch root. When extrapolating the Haigh diagram into the range of compressive mean stress, a conservative estimate for notched components is that a compressive mean stress has no effect (Fig. 1.11). At very large compressive mean stresses the design envelope will be set by yield or buckling limits.

Test results from torsion tests of unnotched specimens indicate that a mean shear stress has no effect on life when added to an alternating shear stress. This trend does not appear to hold for notched torsional components. The effect of a mean shear stress on an alternating normal stress is discussed in Chapter 7.



**Figure 1.11** Estimate of Haigh diagram for notched components using Goodman line.

### 1.4 MODIFYING FACTORS

The results of an R. R. Moore test are from the special case of a mirror-polished 0.25 in diameter specimen loaded under fully reversed bending. To denote this, the endurance limit found using the R. R. Moore test is often given a prime,  $S'_e$ . The endurance limit needed for design situations,  $S_e$ , must take into account differences in size, surface finish, and so on.

For many years the emphasis of most fatigue testing was to gain an empirical understanding of the effects of various factors on the baseline  $S-N$  curves for ferrous alloys in the intermediate to long life ranges. The variables investigated include:

1. Size
2. Type of loading
3. Surface finish
4. Surface treatments
5. Temperature
6. Environment

The results of these tests have been quantified as modification factors which are applied to the baseline  $S-N$  data.

$$S_e = S'_e C_{size} C_{load} C_{surf.fin} \dots \tag{1.12}$$

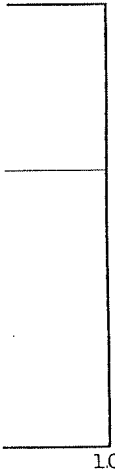
This modified endurance limit tends to be conservative.

The modification factors are usually specified for the endurance limit, and the correction for the remainder of the  $S-N$  curve is not as clearly defined. The general trend is for these modification factors to have less effect at short lives. At the extreme limit of monotonic loading they all approach a value of 1. A conservative estimate is to use the modification factors on the entire  $S-N$  curve.

It is very important to remember that these modification factors are empirical models of a phenomenon and may give limited insight into the underlying physical processes. Great care must be taken when extrapolating these empirical modification factors beyond the range of data used to generate them.

#### 1.4.1 Size Effects

The fatigue failure of a material is dependent on the interaction of a large stress with a critical flaw. In essence, fatigue is controlled by the weakest link of the material, with the probability of a weak link increasing with material volume. This differs from bulk material properties such as yield strength and modulus of elasticity. This phenomenon is evident in the fatigue test results of a material using specimens of varying diameters (see Table 1.1). The size effect has been correlated with the thin layer of surface material subjected to 95% or more of the



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TABLE 1.1 Influence of Size on Endurance Limit

Diameter (in)	Endurance Limit (ksi)
0.3	33.0
1.5	27.6
6.75	17.3

Source: J. H. Faupel and F. E. Fisher, *Engineering Design*, John Wiley and Sons, New York, 1981. Reprinted with permission.

maximum surface stress. A large component will have a less steep stress gradient and hence a larger volume of material subjected to this high stress (Fig. 1.12). Consequently, there will be a greater probability of initiating a fatigue crack in large components. This concept is backed up by test results which show a less pronounced size effect for axial loading, where there is no gradient, than for bending or torsion. The idea of a highly stressed volume is important when considering stress gradients due to notches (see Chapter 4).

There are many empirical fits to the size effect data. A fairly conservative one is [5], in English units,

$$C_{\text{size}} = \begin{cases} 1.0 & \text{if } d \leq 0.3 \text{ in.} \\ 0.869d^{-0.097} & \text{if } 0.3 \text{ in.} \leq d \leq 10 \text{ in.} \end{cases} \quad (1.13)$$

and in SI units,

$$C_{\text{size}} = \begin{cases} 1.0 & \text{if } d \leq 8 \text{ mm} \\ 1.189d^{-0.097} & \text{if } 8 \text{ mm} \leq d \leq 250 \text{ mm} \end{cases} \quad (1.14)$$

where  $d$  is the diameter of the component. Some other points to consider when dealing with the size effect are:

1. The effect is seen mainly at very long lives.
2. The effect is small in diameters up to 2.0 in. even in bending or torsion.
3. Due to the processing problems inherent in large components, there is a greater chance of having residual stresses and various metallurgical variables, which may adversely affect fatigue strength.

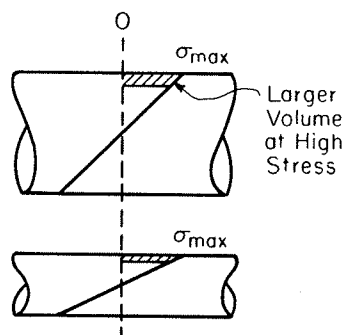


Figure 1.12 Stress gradient in large and small specimens.

The idea of critical volume can also be used to find a size modification factor for noncircular sections (see Ref. 5, p. 294).

**1.4.2 Loading Effects**

When relating the fatigue data from rotating bending and axial loading for a similar specimen, the volume idea discussed in the preceding section can be used. Since the axial specimen has no gradient, it has a greater volume of material subjected to the high stress. The ratio of endurance limits for a material found using axial and rotating bending tests ranges from 0.6 to 0.9. These test data may include some error due to eccentricity in axial loading. A conservative estimate is

$$S_e(\text{axial}) \approx 0.70S_e(\text{bending}) \tag{1.15}$$

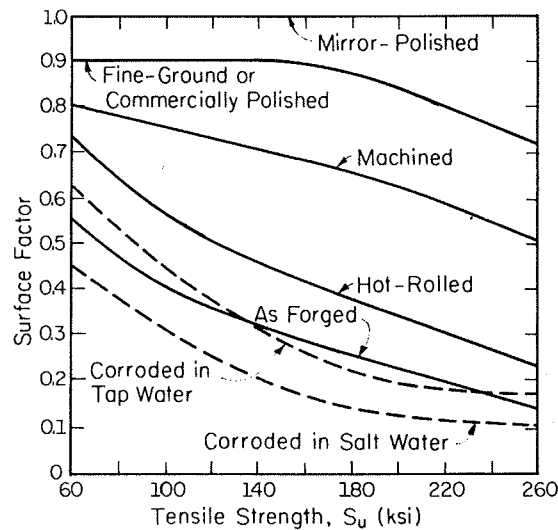
The ratio of endurance limits found using torsion and rotating bending tests ranges from 0.5 to 0.6. A theoretical value of 0.577 has been explained using the von Mises failure criterion. This relationship is discussed more thoroughly in Chapter 7. A reasonable estimate is

$$\tau_e(\text{torsion}) \approx 0.577S_e(\text{bending}) \tag{1.16}$$

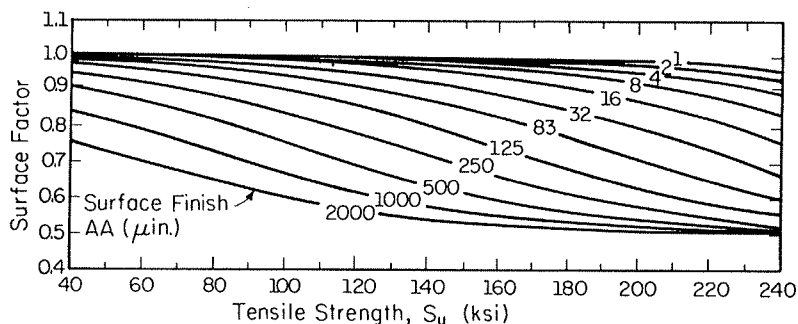
**1.4.3 Surface Finish**

The scratches, pits, and machining marks on the surface of a material add stress concentrations to the ones already present due to component geometry. Uniform fine-grained materials, such as high strength steel, are more adversely affected by a rough surface finish than a coarse-grained material such as cast iron.

The correction factor for surface finish is sometimes presented on graphs that use a qualitative description of surface finish such as "polished" or "machined" (Fig. 1.13). Some of the curves on this plot include effects other than



**Figure 1.13** Surface finish factor: steel parts. (From Ref. 6.)



**Figure 1.14** Surface finish factor versus surface roughness and strength: steel parts. (From Ref. 7.)

surface roughness. For example, the forged and hot-rolled curves include the effects of decarburization.

Other graphs, such as Fig. 1.14, use a quantitative measurement of surface roughness such as  $R_A$ , the root mean square or,  $AA$ , arithmetic average. The surface roughness resulting from various machining operations can be found in machining and manufacturing handbooks.

Some important points about the surface finish effect are:

1. The condition of the surface is more important for higher strength steels.
2. The residual surface stress caused by a machining operation can be important. An example is the residual tensile stress sometimes caused by some grinding operations.
3. At shorter lives, where crack propagation dominates, the condition of surface finish has less effect on the fatigue life.
4. Localized surface irregularities such as stamping marks can serve as very effective stress concentrations and should not be ignored.

#### 1.4.4 Surface Treatment

Since fatigue cracks almost always initiate at a free surface, any surface treatment can have a significant effect on fatigue life. The effect of surface roughness from various forming operations was discussed in the preceding section. Other surface treatments can be categorized as plating, thermal, or mechanical. In all three cases the effect on fatigue life is due primarily to residual stresses.

As a review of residual stress, consider the unnotched beam (Fig. 1.15) subjected to a varying bending moment. The bending moment history is shown in Fig. 1.15d. If the simplifying assumption is made that the material is elastic-perfectly plastic, the history of the stress at the top surface of the beam is as shown in Fig. 1.15e.

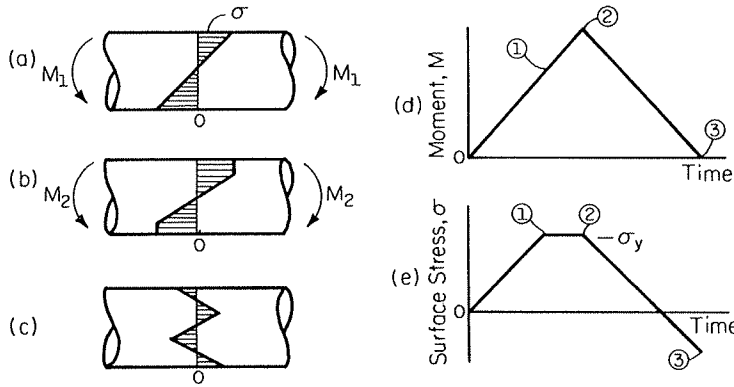
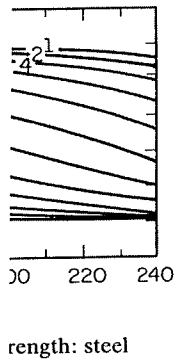


Figure 1.15 Residual stress in unnotched beam in bending.

1. At point 1 the surface of the beam is just at the point of yielding and the stress distribution is linear (Fig. 1.15a).
2. If the moment is increased to point 2, the outer layer of the beam begins to yield (Fig. 1.15b).
3. If the moment is reduced to point 3, the beam will have a residual stress distribution (Fig. 1.15c). When the outer layer of material yielded, it elongated and upon unloading the stresses and strains in the beam must meet compatibility and equilibrium requirements. Although the exact residual stress distribution is difficult to define, the important point is that the outer surface of the beam, which had yielded in tension, is now in residual compression.

Another example of residual stress is the notched member under axial loading, shown in Fig. 1.16. The loading history involves an initial tensile overload followed by fully reversed cyclic loads (Fig. 1.16d).

1. The initial overload (point 1) causes the material at the root of the notch to yield in tension (Fig. 1.16b) and when the load is released (point 2) this material will be in residual compression (Fig. 1.16c).
2. When the cyclic load is applied (points 3 and 4), the stress at the root of the notch will cycle between the limits shown on Fig. 1.16e.

Note that while the load is fully reversed, the stress at the root of the notch (where the fatigue crack will initiate) cycles about a compressive value. The residual stress in the material at the notch root has the same effect as an externally applied compressive mean stress of equal magnitude, and as pointed out in Section 1.3, this will increase life at a given alternating stress level. Remember that this discussion involves the residual mean stress in a notched member, not a mean stress due to an applied load.

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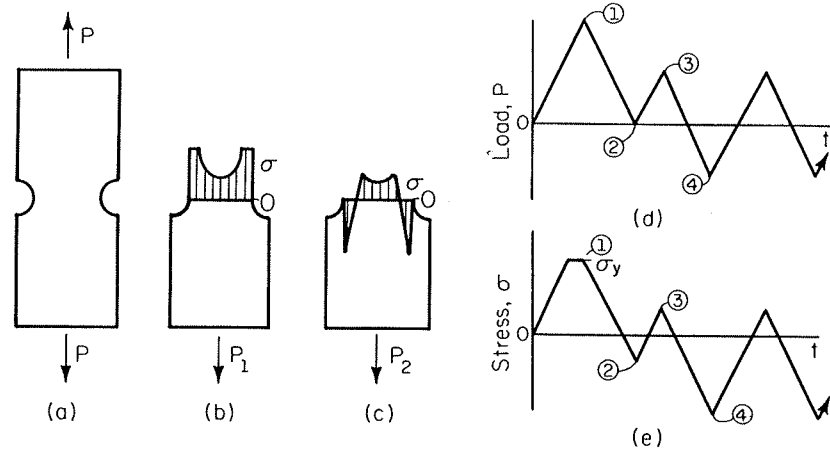


Figure 1.16 Residual stress in notched member under axial loading.

The method just described for producing a residual stress with an initial overload is called *prestressing* or *presetting*. An example of this is given in Table 1.2, which compares the endurance limit values of notched and unnotched plates of 4340 steel ( $S_u = 130$  ksi). A comparison is also made of plates with and without an initial tensile overload. As can be seen, the preload sets up a residual stress which almost negates the effect of the notch.

Presetting is used on such components as coil and leaf springs. It should be noted that the initial overload on a component is favorable for future loading in the same direction as the overload, but unfavorable for loads in the opposite direction. For example, if a coil spring is preloaded in compression, it will have a beneficial effect only for future cyclic loading which is primarily in compression.

Presetting should not be used in cases of fully reversed loading. For example, the cold straightening of an axle shaft can reduce the endurance limit 20 to 50%.

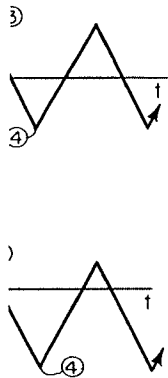
In the following discussion on surface treatments it is important to keep these points in mind:

1. Since fatigue is a surface phenomenon, the residual stress at the surface of the material is critical.

TABLE 1.2 Endurance Limit of Plate with Hole under Axial Loading

	Endurance Limit (ksi)	
	Unnotched	Notched
No preload	58.0	23.0
With preload	56.6	53.7

Source: H. O. Fuchs and R. I. Stephens, *Metal Fatigue in Engineering*, John Wiley and Sons, New York, 1980. Reprinted with permission.



loading.

stress with an initial stress is given in Table 1.1. For unnotched plates of plates with and without residual stress sets up a residual

stress. It should be noted that for future loading in the opposite direction, it will have a net stress in compression. For reversed loading. For endurance limit 20

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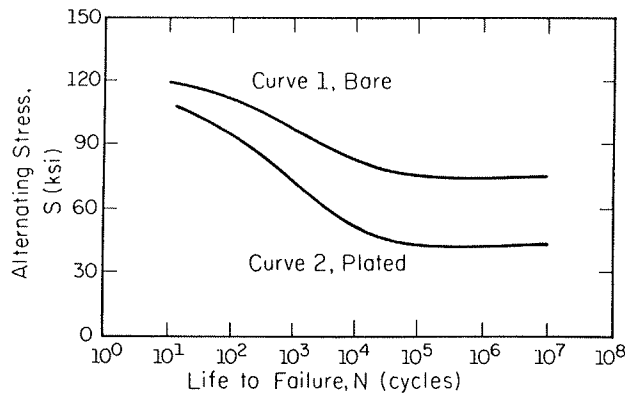


Figure 1.17 Effect of chrome plating on S-N curve of 4140 steel. (From Ref. 1.)

2. Compressive residual stresses are beneficial, and tensile stresses are detrimental to fatigue life.
3. Residual stresses are not always permanent, and various factors, such as high temperatures and overloads, may cause stress relaxation.

**Plating.** Chrome and nickel plating of steels can cause up to a 60% reduction in endurance limits (Figs. 1.17 and 1.18). This is due primarily to the high residual tensile stresses generated by the plating process. The following operations can help alleviate the residual tensile stress problem:

1. Nitride the part before plating.
2. Shot peen the part before or after plating.
3. Bake or anneal the part after plating.

Figure 1.19 shows the effect of shot peening a rotating beam specimen before and after a nickel-plating operation.

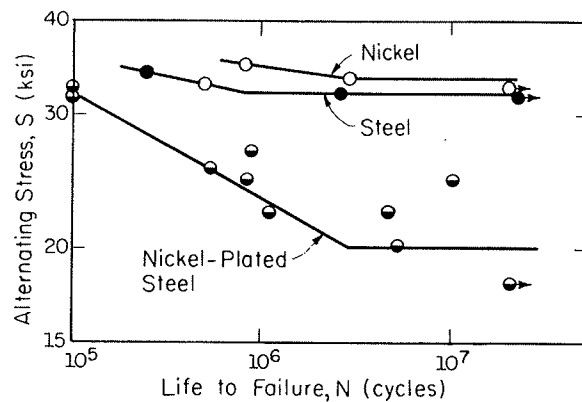


Figure 1.18 Effect of nickel plating on S-N curve of steel ( $S_u = 63$  ksi). (From Ref. 9.)