MACHINERY'S HANDBOOK

A Reference Book for the Mechanical Engineer, Draftsman, Toolmaker and Machinist

By Erik Oberg and F. D. Jones

Holbrook L. Horton, *Editor*

Paul B. Schubert, *Associate Editor*

Franklin D. Jones, *Editor, 1914 to 1950*

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which failure would occur if the stress were constant. Fatigue properties are determined by subjecting test specimens to stress cycles and counting the number of cycles to failure. From a series of such tests in which maximum stress values are progressively reduced, S-N diagrams can be plotted as illustrated by the accompanying figures. The S-N diagram at (1) shows the behavior of a material for which there is an *endurance limit*, S_{en}. Endurance limit is the stress value at which the number of cycles to failure is infinite. Steels have endurance limits that vary according to hardness, composition, and quality; but many non-ferrous metals do not. The S-N diagram at (2) does not have an endurance limit. For a metal that does not have an endurance limit, it is standard practice to specify fatigue strength as the stress value corresponding to a specific number of stress reversals, usually 100,000,000 or 500,000,000.

Factors of safety. — There is always a risk that the working stress to which a member is subjected will exceed the strength of its material. The purpose of a factor of safety is to minimize this risk.

Factors of safety can be incorporated into design calculations in many ways. For most calculations the following equation is used:

$$s_w = \frac{S_m}{f_s} \qquad (1)$$

f_s is the factor of safety, S_m is the strength of the material in pounds per square inch, and s_w is the allowable working stress, also in pounds per square inch. Since the factor of safety is greater than 1, the allowable working stress will be less than the strength of the material.

In general, S_m is based on yield strength for ductile materials, ultimate strength for brittle materials, and fatigue strength for parts subjected to cyclic stressing. Most strength values are obtained by testing standard specimens at 68°F. in normal atmospheres. If, however, the character of the stress or environment differs significantly from that used in obtaining standard strength data, then special data must be obtained. If special data are not available, standard data must be suitably modified.

General recommendations for values of factors of safety are given in the following table.

f_s	Application
1.25–1.5	For use with highly reliable materials where loading and environmental conditions are not severe, and where weight is an important consideration.
1.5–2	For applications using reliable materials where loading and environmental conditions are not severe.
2–2.5	For use with ordinary materials where loading and environmental conditions are not severe.
2.5–3	For less tried and for brittle materials where loading and environmental conditions are not severe.
3–4	For applications in which material properties are not reliable and where loading and environmental conditions are not severe, or where reliable materials are to be used under difficult loading and environmental conditions.

Working Stress. — Calculated working stresses are the products of calculated nominal stress values and stress concentration factors. Calculated nominal stress values are based on the assumption of idealized stress distributions. Such nominal

stresses may be simple stresses, combined stresses, or cyclic stresses. Depending on the nature of the nominal stress, one of the following equations apply:

(2) $s_w = K\sigma$ (3) $s_w = K\tau$ (4) $s_w = K\sigma'$ (5) $s_w = K\tau'$ (6) $s_w = K\sigma_{cy}$ (7) $s_w = K\tau_{cy}$

where K is a stress concentration factor; σ and τ are, respectively, simple normal (tensile or compressive) and shear stresses; σ' and τ' are combined normal and shear stresses; σ_{cy} and τ_{cy} are cyclic normal and shear stresses.

Where there is uneven stress distribution, as illustrated in the accompanying table of simple stresses for Cases 3, 4 and 6, the maximum stress is the one to which the stress concentration factor is applied in computing working stresses. The location of the maximum stress in each case is discussed under the section "Simple Stresses" and the formulas for these maximum stresses are given in the table of simple stresses on page 358.

Stress concentration factors. — Stress concentration is related to type of material, the nature of the stress, environmental conditions, and the geometry of parts. When stress concentration factors are not available that specifically match all of the foregoing conditions, then the following equation may be used:

$$K = 1 + q(K_t - 1) \qquad (8)$$

K_t is a theoretical stress concentration factor that is a function only of the geometry of a part and the nature of the stress; q is the *index of sensitivity* of the material. If the geometry is such as to provide no theoretical stress concentration, $K_t = 1$.

Curves for evaluating K_t are on pages 354 to 357. For constant stresses in cast iron and in ductile materials, $q = 0$ (hence $K = 1$). For constant stresses in brittle materials such as hardened steel, q may be taken as 0.15; for very brittle materials such as steels that have been quenched but not drawn, q may be taken as 0.25. When stresses are suddenly applied (impact stresses) q ranges from 0.4 to 0.6 for ductile materials; for cast iron it is taken as 0.5, and 1 for brittle materials.

Simple stresses. — Simple stresses are produced by constant conditions of loading on elements that can be represented as beams, rods, or bars. The table on page 358 summarizes information pertaining to the calculation of simple stresses. Following is an explanation of the symbols used in simple stress formulae:

σ = simple normal (tensile or compressive) stress in pounds per square inch
τ = simple shear stress in pounds per square inch
F = external force in pounds
V = shearing force in pounds
M = bending moment in inch pounds
T = torsional moment in inch pounds
A = cross sectional area in square inches
Z = section modulus in inches[3]
Z_p = polar section modulus in inches[3]
I = moment of inertia in inches[4]
J = polar moment of inertia in inches[4]
a = area of the web of wide flange and I beams in square inches
y = perpendicular distance from axis through center of gravity of cross sectional area to stressed fiber in inches
c = radial distance from center of gravity to stressed fiber in inches

For direct tension and direct compression loading, Cases 1 and 2 in the Table of Simple Stresses on page 358, the force F must act along a line through the center of

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Mechanical Engineering Design

Example of a cybernetic anthropomorphous machine (CAM), a symbiosis of man and machine. This electro-hydraulic servomanipulator has 20 degrees of freedom, 10 in each slave arm. The slave follows the movements of the master smoothly and continuously while a fraction of the forces encountered by the slave are fed back to the master so that he experiences the sensations of doing the task himself. (COURTESY OF GENERAL ELECTRIC COMPANY.)

JOSEPH EDWARD SHIGLEY

Professor of Mechanical Engineering
University of Michigan

McGraw-Hill Book Company
new york san francisco toronto london

2. The nature and magnitude of the applied load. How was the load obtained in the first place? By test? Or by estimation of some kind? Is the load a static one, a fatigue load, or a combination of the two? Has the possibility of impact been considered? Have all the factors influencing the loading been considered? Many designers, for example, do an excellent job of engineering a new design and then fail to consider the loads which are applied to the equipment during shipment or assembly.

3. Discontinuities and stress concentrations. Sometimes discontinuities which cause stress concentration are introduced after the drawings have been released by the design engineering department. These discontinuities may occur in production, assembly, or inspection, for a variety of reasons. Inspectors, for example, may not appreciate the reasons for maintaining a large fillet radius.

Each time a factor of safety is chosen it is necessary for the engineer to carefully evaluate all the unknowns relating to strength and all those relating to stress. Even in the design of a single part the designer may employ several different factors of safety because of the changes in these uncertainties for each set of calculations. For example, it may be necessary to check a part for the possibility of a fatigue failure as well as for a static failure. One should not expect that the same factor of safety will be used for both these calculations; the reason for this is that the degree of uncertainty for the two types of failure may be quite different.

Selection of n. Now that the reader knows the meaning of factor of safety, he is ready to ask the question, How do I know when I have selected a suitable value? This is indeed a difficult question to answer.

Experience, more than anything else, teaches one what factors to choose. One learns what an ample factor of safety is by having designed an item of equipment that never fails. Similarly, one learns what an inadequate factor of safety is by having designed something which does fail or which fails sometimes. Industries and companies build up a background of experience in this manner, which they can extrapolate for new designs.

Vidosic[1] states that commonly used factors of safety in basic design vary from 1.25 to 4, depending on the uncertainties involved. He applies these to the yield strength for ductile materials, to the ultimate strength for brittle materials, and to the fatigue strength for parts subjected to fatigue loads.

Lipson[2] states that the stresses and strengths should be carefully and thoroughly determined, and then the factor of safety can be chosen from the range

$$1.3 \leq n \leq 2.0$$

He furthermore states that, if an $n > 2$ seems desirable, the problem has not been investigated in sufficient detail.

[1] *Op. cit.*
[2] *Op. cit.*, p. 190

In general, it would seem that if the *minimum* properties, not the average, of the material were determined by many tests, and if there is no deterioration in strength during the lifetime of the part, a margin of 10 to 15 per cent should be sufficient allowance for the uncertainties in strength. And if the stresses are accurately determined and if the nature of the part is such that it cannot be overloaded, or if the overload is known and accounted for, an allowance of 15 to 20 per cent should be sufficient to account for the uncertainties in stress. These add up to a margin of safety of about 30 per cent, or a minimum factor of safety of 1.30.

When danger to human life is involved, these recommended values should be increased. But the designer also has a special responsibility to build into his design extra safeguards to prevent failure. The principles of "fail-safe" can frequently be used; alternatively, one can employ redundant members in the structures to take the load if another fails.

The range of factors of safety recommended in this book (1.25 to 4.0) differs sharply from many to be found in handbooks on machinery, fabrication, and mechanical engineering, and the reader should be aware of this difference. It is not uncommon, for instance, to see values of n as high as 20 recommended for use. When the fine print is studied, it is often found that these recommendations are based on using an average value of the ultimate strength (not the yield strength or endurance limit), with no corrections for size, surface finish, stress concentration, and the like, and that the stresses are only nominal stresses obtained without taking into consideration such factors as combined stresses, fatigue loads, and overloads.

Design Calculations. All stress equations may be represented by the formula

$$\sigma = Cf(x_1, x_2, \ldots, x_n)F(F_1, F_2, \ldots, F_n) \tag{6-4}$$

where C is a constant, x_n are the dimensions of the part to be designed, and F_n are the forces or loads applied to the part. Equation (6-4) is intended to be very general and can represent any kind of stress or stress component—a normal stress, a shear stress, a von Mises stress, or a stress amplitude, for example. The equations for stress will always be written in the form of Eq. (6-4) in this book. Note that the right-hand side of the equation contains the dimensions and the forces. In design, the forces are usually known, and the dimensions are to be determined. Always set up the equations in the form of Eq. (6-4), and substitute in it the known quantities, leaving the appropriate symbols designating the dimensions to be found. Now, in place of σ substitute S/n on the left-hand side of the equation, giving

$$\frac{S}{n} = Cf(x_1, x_2, \ldots, x_n)F(F_1, F_2, \ldots, F_n) \tag{6-5}$$

Since both the strength S and the factor of safety n have been previously determined, the equation can now be solved for the dimensions.

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MECHANICAL DESIGN AND SYSTEMS HANDBOOK

HAROLD A. ROTHBART, Editor-in-Chief

Dean, College of Science and Engineering
Fairleigh Dickinson University
Teaneck, New Jersey

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If the uncertainties are great enough to cause severe weight, volume, or economic penalties, testing and/or more thorough analyses should be performed rather than relying upon very large factors of safety.

Typical values of design safety factors are:[4]

f.s. = 1.25 to 1.5 for exceptionally reliable materials used under controllable conditions and subjected to loads and stresses that can be determined with certainty. Used almost invariably where low weight is a particularly important consideration.

f.s. = 1.5 to 2 for well-known materials under reasonably constant environmental conditions, subjected to loads and stresses that can be determined readily.

f.s. = 2 to 2.5 for average materials operated in ordinary environments and subjected to loads and stresses that can be determined.

f.s. = 2.5 to 3 for less tried as well as for brittle materials under average conditions of environment, load, and stress.

f.s. = 3 to 4 for untried materials used under average conditions of environment, load, and stress.

f.s. = 3 to 4 should also be employed with better-known materials that are to be used in uncertain environments or subjected to uncertain stresses.

f.s. = 2 for impact of very ductile materials where the small index of sensitivity results in low stress-concentration factors.

f.s. = 1.5 for less ductile materials where a higher sensitivity will provide a larger factor of stress concentration.

f.s. = 1.5 for design at higher temperatures, based on the creep strength of the material that will result in a permissible plastic deformation over a preestablished life period.

NOTE: 1. For repeated loads, the factors of safety established are acceptable but must be applied to the endurance limit rather than the yield strength of the material.

2. For castings, forgings, stampings, and welded components, factors of safety here used do not usually vary appreciably from those presented above.

3. Factors of safety to be used with standard design elements, commercially available, should be those recommended for them by reliable manufacturers and/or by established codes for design of machines.

4. Where higher factors of safety might seem desirable, a more thorough analysis should be undertaken before deciding upon their use.

18.7. TRUE FACTOR OF SAFETY

The true factor of safety, which may be defined in terms of load, stress, deflection, creep, wear, etc., is the ratio of the magnitude of any of the above parameters resulting in damage, to its actual value in service. For example:

$$\text{True factor of safety} = \frac{\text{maximum load part can sustain without damage}}{\text{maximum load part sustains in service}}$$

The true factor of safety is determined after a part is built and tested under service conditions.

18.8. STRESS CONCENTRATION[6,7]

Abrupt increases in local stress due to stress raisers, such as notches, holes, fillets, threads, shoulders, and scratches, are termed stress concentrations.

The theoretical (or elastic) stress-concentration factor is defined as

$$K_t = \frac{\text{maximum stress at section}}{\text{average stress at section based upon net area}}$$

The theoretical stress-concentration factor is a function of geometry only and is determined from photoelastic studies, theory of elasticity, or actual strain measurement. K_t does not consider the mitigating effects of local yielding. Table 18.1 lists values of K_t.

Stress concentration should be considered with respect to its effect upon the strength reduction of the specimen. In statically loaded ductile materials, yielding at the

Dr. Roger L. McCarthy