Introduction to Mechanical Engineering Design

Design

To design is either to formulate a plan for the satisfaction of a specified need or to solve a specific problem. If the plan results in the creation of something having a physical reality, then the product must be functional, safe, reliable, competitive, usable, manufacturable, and marketable.

Design is an innovative and highly iterative process. It is also a decision-making process. Decisions sometimes have to be made with too little information, occasionally with just the right amount of information, or with an excess of partially contradictory information. Decisions are sometimes made tentatively, with the right reserved to adjust as more becomes known. The point is that the engineering designer has to be personally comfortable with a decision-making, problem-solving role.

Design is a communication-intensive activity in which both words and pictures are used, and written and oral forms are employed. Engineers have to communicate effectively and work with people of many disciplines. These are important skills, and an engineer's success depends on them.

A designer's personal resources of creativeness, communicative ability, and problemsolving skill are intertwined with the knowledge of technology and first principles.
Engineering tools (such as mathematics, statistics, computers, graphics, and languages)
are combined to produce a plan that, when carried out, produces a product that is ***functional, safe, reliable, competitive, usable, manufacturable, and marketable***, regardless of who builds it or who uses it.

The desing methods

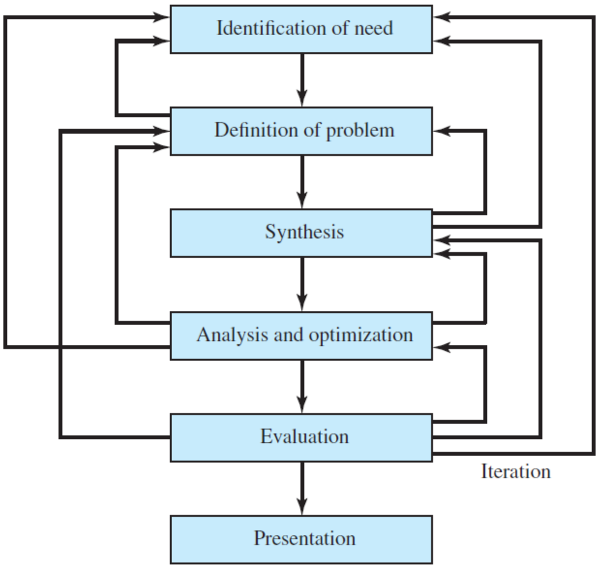
Implicity metod -
Systematical metod -
Standard metod -

Phases and Interactions of the Design Process

What is the design process? How does it begin? Does the engineer simply sit down at a desk with a blank sheet of paper and jot down some ideas? What happens next? What factors influence or control the decisions that have to be made? Finally, how does the design process end? The complete design process, from start to finish, is often outlined as in Fig. 1.1.

Figure 1.1

The phases in design, acknowledging the many feedbacks and iterations.



The process begins with an identification of a need and a decision to do something about it. After many iterations, the process ends with the presentation of the plans for satisfying the need. Depending on the nature of the design task, several design phases may be repeated throughout the life of the product, from inception to termination. In the next several subsections, we shall examine these steps in the design process in detail.

Identification of need generally starts the design process. Recognition of the need and phrasing the need often constitute a highly creative act, because the need may be only a vague discontent, a feeling of uneasiness, or a sensing that something is not right. The need is often not evident at all; recognition can be triggered by a particular adverse circumstance or a set of random circumstances that arises almost simultaneously. For example, the need to do something about a food-packaging machine may be indicated by the noise level, by a variation in package weight, and by slight but perceptible variations in the quality of the packaging or wrap.

There is a distinct difference between the statement of the need and the definition of the problem. The definition of problem is more specific and must include all the specifications for the object that is to be designed. The specifications are the input and output quantities, the characteristics and dimensions of the space the object must occupy, and all the limitations on these quantities. We can regard the object to be designed as something in a black box. In this case we must specify the inputs and outputs of the box, together with their characteristics and limitations. The specifications define the cost, the number to be manufactured, the expected life, the range, the operating temperature, and the reliability. Specified characteristics can include the speeds, feeds, temperature limitations, maximum range, expected variations in the variables, dimensional and weight limitations, etc.

There are many implied specifications that result either from the designer's particular environment or from the nature of the problem itself. The manufacturing processes that are available, together with the facilities of a certain plant, constitute restrictions on a designer's freedom, and hence are a part of the implied specifications. It may be that a small plant, for instance, does not own cold-working machinery. Knowing this, the designer might select other metal-processing methods that can be performed in the plant. The labor skills available and the competitive situation also constitute implied constraints. Anything that limits the designer's freedom of choice is a constraint. Many materials and sizes are listed in supplier's catalogs, for instance, but these are not all easily available and shortages frequently occur. Furthermore, inventory economics requires that a manufacturer stock a minimum number of materials and sizes.

The synthesis of a scheme connecting possible system elements is sometimes called the invention of the concept or concept design. This is the first and most important step in the synthesis task. Various schemes must be proposed, investigated, and quantified in terms of established metrics. As the fleshing out of the scheme progresses, analyses must be performed to assess whether the system performance is satisfactory or better, and, if satisfactory, just how well it will perform. System schemes that do not survive analysis are revised, improved, or discarded. Those with potential are optimized to determine the best performance of which the scheme is capable. Competing schemes are compared so that the path leading to the most competitive product can be chosen. Figure 1.1 shows that synthesis and analysis and optimization are intimately and iteratively related.

We have noted, and we emphasize, that design is an iterative process in which we proceed through several steps, evaluate the results, and then return to an earlier phase of the procedure. Thus, we may synthesize several components of a system, analyze and optimize them, and return to synthesis to see what effect this has on the remaining parts of the system. For example, the design of a system to transmit power requires attention to the design and selection of individual components (e.g., gears, bearings, shaft). However, as is often the case in design, these components are not independent. In order to design the shaft for stress and deflection, it is necessary to know the applied forces. If the forces are transmitted through gears, it is necessary to know the gear specifications in order to determine the forces that will be transmitted to the shaft. But stock gears come with certain bore sizes, requiring knowledge of the necessary shaft diameter. Clearly, rough estimates will need to be made in order to proceed through the process, refining and iterating until a final design is obtained that is satisfactory for each individual component as well as for the overall design specifications. Throughout the text we will elaborate on this process for the case study of a power transmission design.

Both analysis and optimization require that we construct or devise abstract models of the system that will admit some form of mathematical analysis. We call these models mathematical models. In creating them it is our hope that we can find one that will simulate the real physical system very well. As indicated in Fig. 1.1, evaluation is a significant phase of the total design process. Evaluation is the final proof of a successful design and usually involves the testing of a prototype in the laboratory.

Here we wish to discover if the design really satisfies the needs. *Is it reliable? Will it compete successfully with similar products? Is it economical to manufacture and to use? Is it easily maintained and adjusted? Can a profit be made from its sale or use? How likely is it to result in product-liability lawsuits? And is insurance easily and cheaply obtained? Is it likely that recalls will be needed to replace defective parts or systems?* The project designer or design team will need to address a myriad of engineering and non-engineering questions.

Communicating the design to others is the final, vital presentation step in the design process. Undoubtedly, many great designs, inventions, and creative works have been lost to posterity simply because the originators were unable or unwilling to properly explain their accomplishments to others. Presentation is a selling job. The engineer, when presenting a new solution to administrative, management, or supervisory persons, is attempting to sell or to prove to them that their solution is a better one. Unless this can be done successfully, the time and effort spent on obtaining the solution have been largely wasted. When designers sell a new idea, they also sell themselves. If they are repeatedly successful in selling ideas, designs, and new solutions to management, they begin to receive salary increases and promotions; in fact, this is how anyone succeeds in his or her profession.

Principles of desing

Design Considerations

Sometimes the strength required of an element in a system is an important factor in the determination of the geometry and the dimensions of the element. In such a situation we say that strength is an important design consideration. When we use the expression design consideration, we are referring to some characteristic that influences the design of the element or, perhaps, the entire system. Usually quite a number of such characteristics must be considered and prioritized in a given design situation. Many of the important ones are as follows (not necessarily in order of importance):

Functionality, Strength/stress, Distortion/deflection/stiffness, Wear, Corrosion, Safety, Reliability, Manufacturability, Utility, Cost, Friction, Weight, Life, Noise, Styling, Shape, Size, Control, Thermal properties, Surface, Lubrication, Marketability, Maintenance, Volume, Liability, Remanufacturing/resource recovery

Some of these characteristics have to do directly with the dimensions, the material, the processing, and the joining of the elements of the system. Several characteristics may be interrelated, which affects the configuration of the total system.

Uncertainty

Uncertainties in machinery design abound. Examples of uncertainties concerning stress and strength include:

- Composition of material and the effect of variation on properties.
- Variations in properties from place to place within a bar of stock.
- Effect of processing locally, or nearby, on properties.
- Effect of nearby assemblies such as weldments and shrink fits on stress conditions.
- Effect of thermomechanical treatment on properties.
- Intensity and distribution of loading.
- Validity of mathematical models used to represent reality.
- Intensity of stress concentrations.
- Influence of time on strength and geometry.
- Effect of corrosion.
- Effect of wear.
- Uncertainty as to the length of any list of uncertainties.

Engineers must accommodate uncertainty. Uncertainty always accompanies change. Material properties, load variability, fabrication fidelity, and validity of mathematical models are among concerns to designers.

There are mathematical methods to address uncertainties. The primary techniques are the deterministic and stochastic methods. The deterministic method establishes a design factor based on the absolute uncertainties of a loss-of-function parameter and a maximum allowable parameter. Here the parameter can be load, stress, deflection, etc. Thus, the design factor n_d is defined as

$$n_d = \frac{\text{loss-of-function parameter}}{\text{maximum allowable parameter}} \tag{1}$$

If the parameter is load (as would be the case for column buckling), then the maximum allowable load can be found from

$$\text{Maximum allowable load} = \frac{\text{loss-of-function load}}{n_d} \tag{2}$$

Design Factor and Factor of Safety

A general approach to the allowable load versus loss-of-function load problem is the deterministic design factor method, and sometimes called the classical method of design. The fundamental equation is Eq. (1) where nd is called the design factor. All loss-of - function modes must be analyzed, and the mode leading to the smallest design factor governs. After the design is completed, the actual design factor may change as a result of changes such as rounding up to a standard size for a cross section or using off-theshelf components with higher ratings instead of employing what is calculated by using the design factor. The factor is then referred to as the factor of safety, n. The factor of safety has the same definition as the design factor, but it generally differs numerically.

Since stress may not vary linearly with load, using load as the lossof- function parameter may not be acceptable. It is more common then to express the design factor in terms of a stress and a relevant strength. Thus Eq. (1) can be rewritten as The stress and strength terms in Eq. (3) must be of the same type and units. Also, the stress and strength must apply to the same critical location in the part.

$$n_d = \frac{\text{loss-of-function strength}}{\text{allowable stress}} = \frac{S}{\sigma (\text{or } \tau)} \tag{3}$$

Limit states for desing of machine parts

Limit states are most frequenty used for desining of machine parts. The state when machine part or construction loses its ability to fulfill its function, is caed the limit state. in technical practise are often use these limit states:

A. **Static Strength**
B. **Fatigue Strength**
C. **Surface Fatigue Strength**

Static Strength

Ideally, in designing any machine element, the engineer should have available the results of a great many strength tests of the particular material chosen. These tests should be made on specimens having the same heat treatment, surface finish, and size as the element the engineer proposes to design; and the tests should be made under exactly the same loading conditions as the part will experience in service. This means that if the part is to experience a bending load, it should be tested with a bending load. If it is to be subjected to combined bending and torsion, it should be tested under combined bending and torsion. If it is made of heat-treated AISI 1040 steel drawn at 500°C with a ground finish, the specimens tested should be of the same material prepared in the same manner. Such tests will provide very useful and precise information. Whenever such data are available for design purposes, the engineer can be assured of doing the best possible job of engineering.

The cost of gathering such extensive data prior to design is justified if failure of the part may endanger human life or if the part is manufactured in sufficiently large quantities. Refrigerators and other appliances, for example, have very good reliabilities because the parts are made in such large quantities that they can be thoroughly tested in advance of manufacture. The cost of making these tests is very low when it is divided by the total number of parts manufactured. You can now appreciate the following four design categories:

1. Failure of the part would endanger human life, or the part is made in extremely large quantities; consequently, an elaborate testing program is justified during design.
2. The part is made in large enough quantities that a moderate series of tests is feasible.
3. The part is made in such small quantities that testing is not justified at all; or the design must be completed so rapidly that there is not enough time for testing.
4. The part has already been designed, manufactured, and tested and found to be unsatisfactory. Analysis is required to understand why the part is unsatisfactory and what to do to improve it.

See Exercise 1

Stress Concentration

Stress concentration is a highly localized effect. In some instances it may be due to a surface scratch. If the material is ductile and the load static, the design load may cause yielding in the critical location in the notch. This yielding can involve strain strengthening of the material and an increase in yield strength at the small critical notch location. Since the loads are static and the material is ductile, that part can carry the loads satisfactorily with no general yielding. In these cases the designer sets the geometric (theoretical) stress-concentration factor K_t to unity. The rationale can be expressed as follows. The worst-case scenario is that of an idealized non–strain-strengthening material shown in Fig. 1.2. The stress-strain curve rises linearly to the yield strength Sy, then proceeds at constant stress, which is equal to Sy. If the material is ductile, with a yield point of 40 kpsi, and the theoretical stress-concentration factor (SCF) K_t is 2,

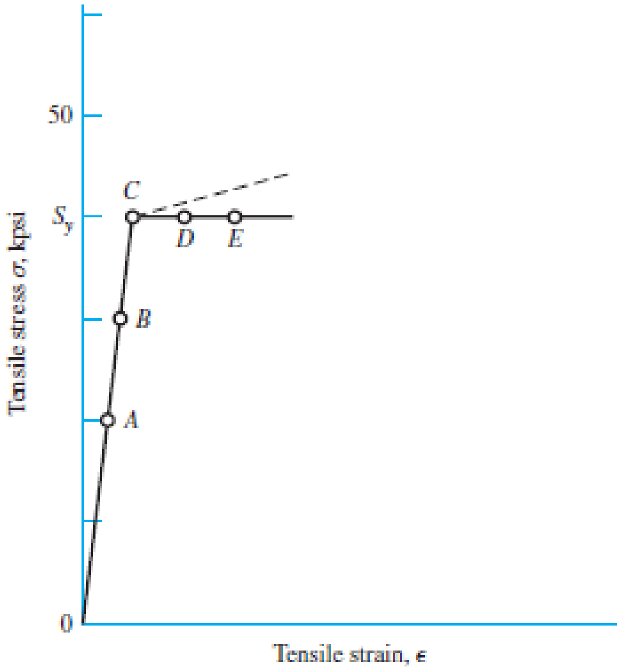
• A load of 20 kip induces a nominal tensile stress of 20 kpsi in the shank as depicted at point A in Fig. 5–6. At the critical location in the fillet the stress is 40 kpsi, and the SCF is $K = \sigma_{max}/\sigma_{nom} = 40/20 = 2$.

• A load of 30 kip induces a nominal tensile stress of 30 kpsi in the shank at point B. The fillet stress is still 40 kpsi (point D), and the SCF $K = \sigma_{max}/\sigma_{nom} = S_y/\sigma = 40/30 = 1.33$.

• At a load of 40 kip the induced tensile stress (point C) is 40 kpsi in the shank. At the critical location in the fillet, the stress (at point E) is 40 kpsi. The SCF $K = \sigma_{max}/\sigma_{nom} = S_y/\sigma = 40/40 = 1$.

Figure 1.2

An idealized stress-strain curve. The dashed line depicts a strain-strengthening material.



For materials that strain-strengthen, the critical location in the notch has a higher Sy. The shank area is at a stress level a little below 40 kpsi, is carrying load, and is very near its failure-by-general-yielding condition. This is the reason designers do not apply K_t in static loading of a ductile material loaded elastically, instead setting $K_t = 1$. When using this rule for ductile materials with static loads, be careful to assure yourself that the material is not susceptible to brittle fracture in the environment of use. The usual definition of geometric (theoretical) stressconcentration factor for normal stress K_t and shear stress K_{ts} is given by Eq. as

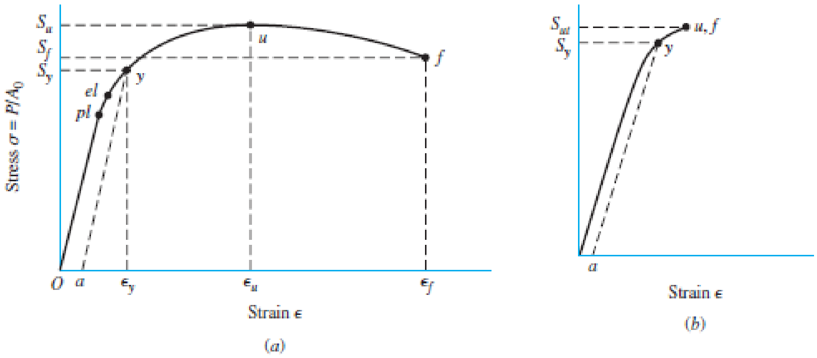
$$\sigma_{max} = K_t \sigma_{nom} \qquad (a)$$

$$\tau_{max} = K_{ts} \tau_{nom} \qquad (b)$$

Since your attention is on the stress-concentration factor, and the definition of σ_{nom} or τ_{nom} is given in the graph caption or from a computer program, be sure the value of nominal stress is appropriate for the section carrying the load. As shown in Fig. 1.3b, brittle materials do not exhibit a plastic range. The stress-concentration factor given by Eq. (a) or (b) could raise the stress to a level to cause fracture to initiate at the stress raiser, and initiate a catastrophic failure of the member.

Figure 1.3

Stress-strain diagram obtained from the standard tensile test (*a*) Ductile material; (*b*) brittle material. *pl* marks the proportional limit; *el*, the elastic limit; *y*, the offset-yield strength as defined by offset strain *a*; *u*, the maximum or ultimate strength; and *f*, the fracture strength.



An exception to this rule is a brittle material that inherently contains microdiscontinuity stress concentration, worse than the macrodiscontinuity that the designer has in mind. Sand molding introduces sand particles, air, and water vapor bubbles. The grain structure of cast iron contains graphite flakes (with little strength), which are literally cracks introduced during the solidification process. When a tensile test on a cast iron is performed, the strength reported in the literature includes this stress concentration. In such cases K_t or K_{ts} need not be applied. An important source of stress-concentration factors is R. E. Peterson, who compiled them from his own work and that of others. Peterson developed the style of presentation in which the stress-concentration factor K_t is multiplied by the nominal stress snom to estimate the magnitude of the largest stress in the locality.

Distortion-Energy Theory for Ductile Materials

The distortion-energy theory predicts that yielding occurs when the distortion strain energy per unit volume reaches or exceeds the distortion strain energy per unit volume for yield in simple tension or compression of the same material. The distortion-energy (DE) theory originated from the observation that ductile materials stressed hydrostatically (equal principal stresses) exhibited yield strengths greatly in excess of the values given by the simple tension test. Therefore it was postulated that yielding was not a simple tensile or compressive phenomenon at all, but, rather, that it was related somehow to the angular distortion of the stressed element. To develop the theory, note, in Fig.1.4 a, the unit volume subjected to any threedimensional stress state designated by the stresses σ_1, σ_2, and σ_3. The stress state shown in Fig. 1.4b is one of hydrostatic normal stresses due to the stresses σ_{av} acting in each of the same principal directions as in Fig. 1.4a.

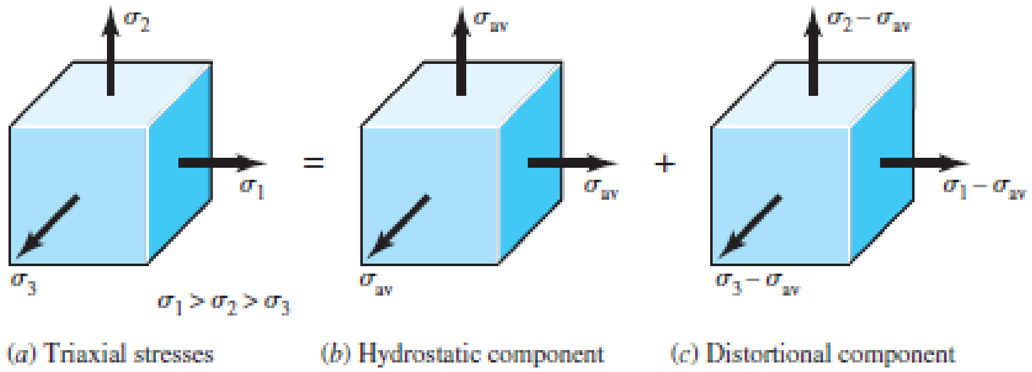


Figure 1.4

(*a*) Element with triaxial stresses; this element undergoes both volume change and angular distortion. (*b*) Element under hydrostatic normal stresses undergoes only volume change. (*c*) Element has angular distortion without volume change.

The formula for σ_{av} is simply

$$\sigma_{av} = (\sigma_1 + \sigma_2 + \sigma_3)/3 \qquad \text{(a)}$$

Thus the element in Fig. 1.4b undergoes pure volume change, that is, no angular distortion. If we regard σ_{av} as a component of σ_1, σ_2, and σ_3, then this component can be subtracted from them, resulting in the stress state shown in Fig. 1.4c. This element is subjected to pure angular distortion, that is, no volume change.

Can be thought of as a single, equivalent, or effective stress for the entire general state of stress given by σ_1, σ_2, and σ_3. This effective stress is usually called the von Mises stress, σ', named after Dr. R. von Mises, who contributed to the theory. Can be written where the von Mises stress is

$$\sigma' = \left[\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}\right]^{1/2}$$

Using xyz components of three-dimensional stress, the von Mises stress can be written as and for plane stress

$$\sigma' = (\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xy}^2)^{1/2}$$

The distortion-energy theory is also called:
• The von Mises or von Mises–Hencky theory
• The shear-energy theory
• The octahedral-shear-stress theory
Understanding octahedral shear stress will shed some light on why the MSS is conservative.
Consider an isolated element in which the normal stresses on each surface are equal to the hydrostatic stress σ_{av}.

Maximum-Normal-Stress Theory for Brittle Materials

The maximum-normal-stress (MNS) theory states that failure occurs whenever one of the three principal stresses equals or exceeds the strength. Again we arrange the principal stresses for a general stress state in the ordered form $\sigma_1 \geq \sigma_2 \geq \sigma_3$. This theory then predicts that failure occurs whenever

$$\sigma_1 \geq S_{ut} \quad \text{or} \quad \sigma_3 \leq -S_{uc}$$

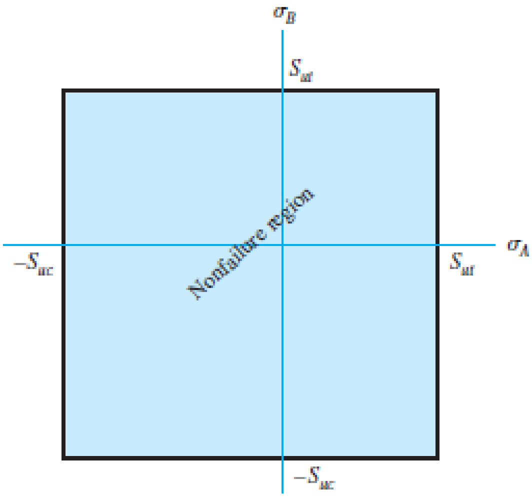
where S_{ut} and S_{uc} are the ultimate tensile and compressive strengths, respectively, given as positive quantities. For plane stress, with the principal stresses given by Eq., with $\sigma_A \geq \sigma_B$, Eq. can be written as

$$\sigma_A \geq S_{ut} \quad \text{or} \quad \sigma_B \leq -S_{uc}$$

which is plotted in Fig. 1.5.

Figure 1.5

Graph of maximum-normal-stress (MNS) theory failure envelope for plane stress states.



As before, the failure criteria equations can be converted to design equations. We can consider two sets of equations where $\sigma_A \geq \sigma_B$ as

$$\sigma_A = S_{ut}/n \quad \text{or} \quad \sigma_B = -S_{uc}/n$$

As will be seen later, the maximum-normal-stress theory is not very good at predicting failure in the fourth quadrant of the σ_A , σ_B plane. Thus, we will not recommend the theory for use. It has been included here mainly for historical reasons.

Important Design Equations

The following equations and their locations are provided as a summary. Note for plane stress: The principal stresses in the following equations that are labeled σ_A and σ_B represent the principal stresses determined from the two-dimensional Eq. (3–13).

Maximum Shear Theory

p. 234
$$\tau_{\max} = \frac{\sigma_1 - \sigma_3}{2} = \frac{S_y}{2n} \tag{5–3}$$

Distortion-Energy Theory

Von Mises stress, p. 237

$$\sigma' = \left[\frac{(\sigma_1 - \sigma_2)^2 + (\sigma_2 - \sigma_3)^2 + (\sigma_3 - \sigma_1)^2}{2}\right]^{1/2} \tag{5–12}$$

p. 237
$$\sigma' = \frac{1}{\sqrt{2}}[(\sigma_x - \sigma_y)^2 + (\sigma_y - \sigma_z)^2 + (\sigma_z - \sigma_x)^2 + 6(\tau_{xy}^2 + \tau_{yz}^2 + \tau_{zx}^2)]^{1/2} \tag{5–14}$$

Plane stress, p. 237

$$\sigma' = (\sigma_A^2 - \sigma_A\sigma_B + \sigma_B^2)^{1/2} \tag{5–13}$$

p. 237
$$\sigma' = (\sigma_x^2 - \sigma_x\sigma_y + \sigma_y^2 + 3\tau_{xy}^2)^{1/2} \tag{5–15}$$

Yield design equation, p. 238

$$\sigma' = \frac{S_y}{n} \tag{5–19}$$

Shear yield strength, p. 239

$$S_{sy} = 0.577\, S_y \tag{5–21}$$

Coulomb-Mohr Theory

p. 243
$$\frac{\sigma_1}{S_t} - \frac{\sigma_3}{S_c} = \frac{1}{n} \tag{5–26}$$

where S_t is tensile yield (ductile) or ultimate tensile (brittle), and S_t is compressive yield (ductile) or ultimate compressive (brittle) strengths.

Modified Mohr (Plane Stress)

$$\sigma_A = \frac{S_{ut}}{n} \qquad \sigma_A \ge \sigma_B \ge 0 \tag{5–32a}$$

$$\sigma_A \ge 0 \ge \sigma_B \quad \text{and} \quad \left|\frac{\sigma_B}{\sigma_A}\right| \le 1$$

p. 250
$$\frac{(S_{uc} - S_{ut})\sigma_A}{S_{uc}S_{ut}} - \frac{\sigma_B}{S_{uc}} = \frac{1}{n} \qquad \sigma_A \ge 0 \ge \sigma_B \quad \text{and} \quad \left|\frac{\sigma_B}{\sigma_A}\right| > 1 \tag{5–32b}$$

$$\sigma_B = -\frac{S_{uc}}{n} \qquad 0 \ge \sigma_A \ge \sigma_B \tag{5–32c}$$