THE UNIVERSITY OF QUEENSLAND
DEPARTMENT OF MECHANICAL ENGINEERING

DESIGN ENGINEERING NOTES

DESIGN AGAINST MECHANICAL FAILURE

MODES OF FAILURE

One of the major decisions confronting the designer is the selection of appropriate failure prevention criteria. This is largely influenced by the mode of failure of the machine or structural member. The focus of these notes is on mechanical modes as opposed to economic or other modes of failure.

The common modes of mechanical failure are:

1) Yielding

(a) Maximum stress exceeds the yield strength of the material, causing it to deform plastically.

(b) Creep deformation, whereby the member deforms under a constant load, usually at an elevated temperature.

2) Fracture

a) Due to static loads

b) Due to cyclic loads (fatigue failure – high cycle or low cycle)

c) Due to impact loads (e.g. brittle fracture)

3) Excessive Elastic Deformation (either deflection or slope)

4) Wear

5) Buckling (local or global)

6) Corrosion fatigue and caustic embrittlement

The mode of failure in turn is influenced by such factors as the type and duration of the load, the shape of the part (effect of stress concentration), and the nature of the material (ductile or brittle).

There are a number of conditions that cause a normally ductile material to act like a brittle material. They are:

- Repeated or fatigue loading
- Impact, particularly at low temperature
- Creep
- Triaxial state of stress
- Severe quenching without tempering
- Strain hardening accompanying yielding.

WARNING

The designer's decision in choosing the most likely mode or modes of failure is crucial. If it is wrong then all subsequent analysis, no matter how elegant or sophisticated, is meaningless and misleading.

UNKNOWNS AND UNCERTAINTIES IN DESIGN CALCULATIONS

Simple example

Consider the design problem of transmitting a load, P, over a short distance by means of a tension member of constant cross-sectional area.

The designer has to choose a suitable material in terms of environment, availability and cost and then determine the required cross-sectional area of the member.

Assume that the material chosen is ordinary structural steel.

The first and very critical decision to be made is ‘how will a straight piece of uniform cross-section structural steel under a tensile load fail, if subjected to an increasing load?’ or ‘how will such a bar fail under a constant load if the cross-sectional area is gradually reduced?’

What we are trying to establish is the mode or modes of failure under these circumstances.

If the load is increased slowly a piece of structural steel will stretch elastically (to all intents and purposes) until the yield point is reached and then stretch plastically. If the load continues to increase, the bar will eventually fracture after about a 30% strain. The designer may decide that permanent set can not be tolerated so that yield in tension is the mode of failure, or that only complete separation is important so that final fracture is the relevant mode of failure and the ultimate tensile strength is of interest.

For other types of loading, failure may occur in a variety of ways. In many cases the part may fail due to a combination of effects so that the actual mode of failure is very complex.

For this case, assume that the load is static and that the mode of failure is yield. The problem then is to predict when this condition will be reached so that the part can be sized adequately.

Using the concept of stress we can write -

$$\sigma = \frac{P}{A}$$

where σ = tensile stress (MPa)

P = applied load (N)
A = original cross-sectional area (m^2)

[It should be emphasized that stress is only a mathematical concept and is not a description of the actual loading state across the bar - it is based on original area whereas as soon as any load is applied the actual area is decreased (μ = 0.3

for steel). It also assumes that the load is applied perfectly along the axis of the member (no bending) and that the member is straight, isotropic, homogeneous and so on.]

Hence we may define the failure condition for the bar as -

$\sigma = S_Y$ where S_Y indicates a yield condition.

So the bar may be sized, given a load P, by –

$$A = \frac{P}{S_y}$$

This apparently straightforward design relation seems to imply that a unique and accurate answer to the problem can be found. This is not really so.

The relationship derives from a mathematical model that is not an accurate representation of the real situation. In addition, any estimation of values for the factors on the right hand side must take into account the many uncertainties involved. So if we wish to make sure that components designed on this basis will be strong enough we have to estimate and allow for these uncertainties.

The uncertainties come from a number of sources:

A: Properties of Materials (e.g. S_Y in our simple example)

1) An uncertainty exists as to the exact properties of the chosen material. Published details on the properties of a material from a variety of reliable sources show numerous discrepancies. Also, there is always some variation in a material produced in different batches as well as doubts about whether yield point refers to high or low yield (0.2% offset or 0.5% offset) and so on.

2) Size effect. Frequently standard test data are acquired using 0.5 inches or 10mm diameter specimens but design parts may require larger thicknesses. The properties of such larger parts will be different from the test data.

3) Effect of manufacturing operations. Machining, cold working, grinding, heat treating, plating operations etc. all affect properties to a greater or lesser degree.

4) Effect of assembly operations, i.e. riveting, bolting, shrink or press fitting may affect the strength.

5) Effect of time. Properties may vary with the age of the material.

For fairly well known materials, if minimum (not average) properties have been determined from many tests, and there is no significant deterioration in strength during the lifetime of the part, then a margin of 10 to 15% should be sufficient allowance for uncertainties in material properties. For less well-known materials the allowance should be increased.

B: Estimation of Load (P in the example)

Loads should always be estimated as precisely as possible. They should not be scaled up by a so-called ‘safety factor’(see Design Factor later). The loads used in design should be your best estimate of the actual loads we would expect to measure in the component, part or structure if it were instrumented during service.

In some applications the loads are well known, whereas in others loads are not accurately known. Estimation of loads under dynamic conditions (and these are the conditions normally encountered in Mechanical Engineering) usually proves more difficult. Shock factors are often used to estimate loads under dynamic conditions (shock factors are quite different to safety factors).

The allowance for this uncertainty must depend on a careful appreciation of the individual problem.

C: Stress Calculations

1) Mathematical Models of Component Behavior

The various theories or models which are used in stress-strain calculations are only attempts to describe reality. The engineering designer must not treat them as being universally true and infallible. Some may be fairly accurate descriptions of what really happens (or appears to happen) while others may only be used as a rough guide in predicting behavior. They are normally applicable over a restricted range and include many assumptions. It must be ensured that the use of a particular model in any situation is warranted.

2) Theories of Failure

Most of the information available about material properties has been derived from simple tension or compression tests. In general, however, the designer is confronted with a much more complicated situation where the part is subjected to combined stress. The behavior of the material under the more complicated stress state must be predicted using theories of failure and simple tension test data (e.g. Maximum Shear Stress Theory). Once again these are only attempts to describe reality.

A common allowance for such uncertainties in cases where the designer has a good deal of confidence in the stress calculations is about 10 to 15%.

D: Consequences of Failure

The designer must compromise between the viability of the product in terms of cost, weight and size and the consequences of failure. Failure of a component may result in effects ranging from minor irritation, through some inconvenience, to loss of production from minor injury or breakdown, up to severe injury or death or damage to national prestige.

The designer has to make the decision as to the consequences of failure and then make the necessary allowances in the design of the product. This is perhaps one of the most difficult design decisions. Where failure simply involves loss of production it may be reasonably easy to balance costs of design and manufacture with costs of lost production.

Insurance companies have scales that award certain amounts as compensation for injury, disability and fatality but this sort of costing does not include human quality of life.

The allowances made to account for the consequences of failure must be based on a very careful study of each case.

E: Variations in Stress and Strength

It is obvious that there could be a good deal of variation in material properties for a batch of apparently identical components. Equally, each of these components is likely to experience a unique range of loads during a lifetime. If every component is to function satisfactorily, then, for every component, the strength of the material must exceed or at least equal the stresses on the component in service.

Figure 1 illustrates the idealized distribution of strength of a group of ‘identical’ components (right hand Bell curve) and the idealized range of stresses each might experience in service (left hand Bell curve).

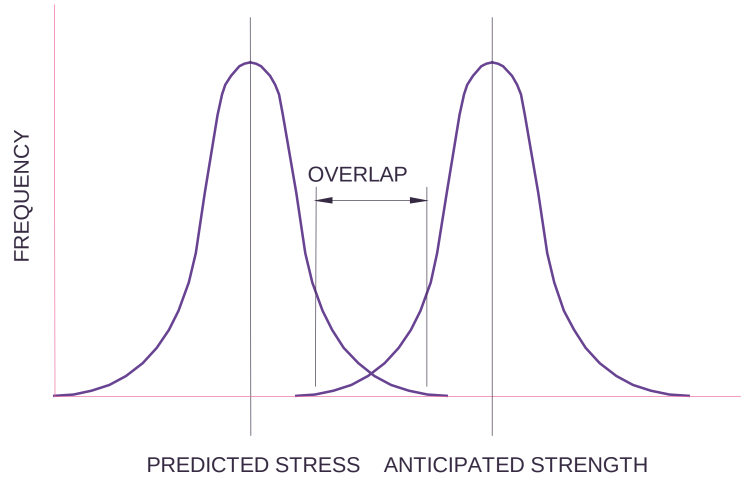

Figure 1

The stress actually on the material and the strength of the material will exhibit variations. In the majority of cases the strength exceeds the stress and the part is satisfactory. However, in some few cases (in the overlap region) the stress may exceed the strength and failure would result.

In order to reduce the likelihood of failure the overlap could possibly be avoided by -

(a) separating the distributions by increasing the difference between the 'average' material stress and the 'average' material strength (Figure 2).

Figure 2

This means that while failure may be almost completely avoided a great number of components are very much stronger than they need to be and, of course, very much more expensive. In fact, such products may not be viable.

(b) maintaining the difference between the averages but reducing the spread of each distribution (Figure 3).

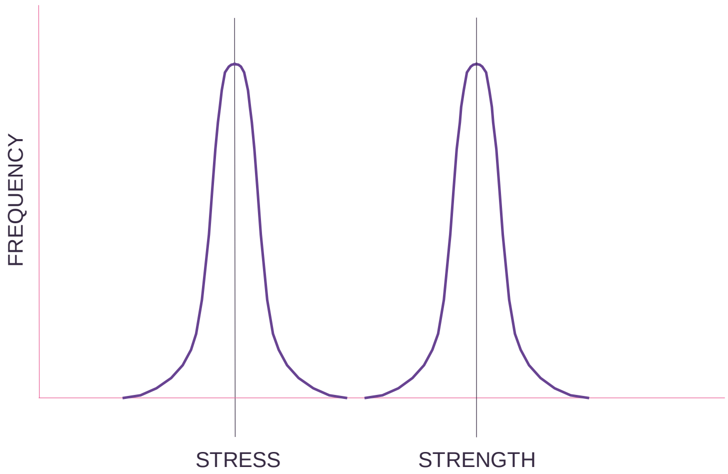

Figure 3

This means that materials must be manufactured and tested under tight controls, more time and effort must be expended in the estimation of loads and the calculation of stresses and parts must be manufactured and installed under close supervision. Once again, a much more expensive component results.

DESIGN FACTORS

We deal with these unknowns and uncertainties by introducing a design factor, d.f. The term design factor is used rather than 'safety factor'. Safety factor might imply that the design is safer if we apply a large enough factor. Ironically, the opposite might be true. This factor should rightfully be called a 'factor of ignorance' since it expresses our lack of complete knowledge as to how each component we design and make will be loaded and how it will behave in service.

We reflect this by ensuring that the following inequality is maintained –

$$\text{stress}(\sigma) \quad \leq \quad \frac{\text{strength}(S)}{\text{d.f.}}$$

Note: Stress, denoted by the symbol σ, is a function of the load(s) acting and the geometry of the part. Strength, denoted by the symbol S, is a property of the part based on a particular mode of failure. Strength depends on the part geometry, the loading conditions and the material behavior. The yield strength, S_Y (not σ_Y), is found for a particular shape test specimen under prescribed loading conditions (i.e. a tensile test).

The designer should always attempt to obtain the most complete and accurate estimate of the stress and the strength and rely as little as possible on the design factor.

By placing the d.f. as the denominator on the right of the inequality, we are effectively 'derating' the strength of the component or part by the value of the design factor.

Importantly, we are not scaling up the stress (or the loads) by the design factor. Therefore all values of stress in the design calculations will be those we would expect to measure in an actual component.

ESTIMATING THE DESIGN FACTOR

The value of the design factor, d.f., is estimated from all the necessary allowances for uncertainty.

For example:

Allowance for uncertainties in properties	1.15	(15%)
Allowance for uncertainties in load	1.20	(say)
Allowance for uncertainties in stress calculations	1.10	(say)
Allowance for consequences of failure	1.50	(50%)

$$\begin{aligned} \text{d.f.} &= 1.15 \times 1.20 \times 1.10 \times 1.50 \\ &= 2.3 \end{aligned}$$

as

$$\text{stress}(\sigma) \quad \leq \quad \frac{\text{strength}(S)}{\text{d.f.}}$$

the component in the example would be designed on the basis that the required cross-sectional area would be -

$$A = \frac{2.3P}{S_y}$$

This example is based on the use of ‘average’ stress and strength. If the upper limit of stress and the lower limit of strength were used, then the design factor would be much lower.

In addition to those allowances already mentioned, the designer in an individual case may decide that other factors demand some consideration.

For example -

- frequency of maintenance and inspection
- working environment
- quality of manufacture

ESTIMATING DYNAMIC LOADS (SHOCK FACTOR)

So far we have been discussing the load on a component in terms of that load which could normally be expected to be experienced. However, conditions arise where this ‘normal’ or nominal load could be exceeded by a substantial amount.

For example, we may be concerned with the design of a crane for a workshop, perhaps used for loading and unloading a large milling machine. The milling machine may be capable of handling objects up to a maximum of 9 kN and the crane would be expected to carry a normal load of at least as much as this.

If an object on the floor of a workshop is picked up quickly this may result in a temporary increase in load due to inertia effects. Again, if an object is pushed off the milling machine and is attached to the crane by a slack rope the ensuing jolt increases the actual load experienced by the crane.

These dynamic effects are common in mechanical design because we are primarily concerned with moving machinery.

One approach is to scale up a nominal load by a shock factor to estimate the likely actual dynamic load.

$$P_{\text{dynamic shock}} = \text{Shock factor} \times P_{\text{nominal}}$$

This shock factor is determined from the nature of the load and the nature of the power source.

Various texts give some guidance about the load increases to be expected under certain conditions, for example -

SHOCK FACTOR			
Source of Power	Character of Load		
	Uniform Shock	Moderate Shock	Heavy Shock
Electric motor	1.00	1.25	1.75
Multi-cylinder engine	1.25	1.50	1.75
Single-cylinder engine	1.75	2.00	2.25

CLOSURE

- All stress calculations refer to a single, critical location on a component or part. They do not apply to the part as a whole. Be sure to indicate clearly (with a diagram) where that critical location is within the part.
- Strength is based on a mode of failure, e.g. high cycle fatigue.
- Design factors should be estimated and used with full consideration and with caution. Some industries or companies may have developed good practice recommendations for the appropriate value of design factor to be used in particular circumstances. Some Standards contain embedded design factors, e.g. the safe working load of cranes is typically one sixth of the nominal strength, implying a design factor of 6.
- The concept of ‘stress’ and ‘strength’ and the inequality can be extended to modes of failure where the mechanism of failure is not expressed in terms of stress (i.e. MPa). One example is buckling; another is excessive elastic deformation.