Appendix C The Factor of Safety as a Design Variable

C.1 INTRODUCTION

The factor of safety is a factor of ignorance. If the stress on a part at a critical location (the applied stress) is known precisely, if the material’s strength (the allowable strength) is also known with precision, and the allowable strength is greater than the applied stress, then the part will not fail. However, in the real world, all of the aspects of the design have some degree of uncertainty, and therefore a fudge factor, a factor of safety, is needed. A factor of safety is one way to account for the uncontrollable noises that were discussed in Chap. 11.

In practice the factor of safety is used in one of three ways. (1) It can be used to reduce the allowable strength, such as the yield or ultimate strength of the material, to a lower level for comparison with the applied stress; (2) it can be used to increase the applied stress for comparison with the allowable strength; or (3) it can be used as a comparison for the ratio of the allowable strength to the applied stress. We apply the third definition here, but all three are based on the simple formula

\[ FS = \frac{S_{al}}{\sigma_{ap}} \]

Here \( S_{al} \) is the allowable strength, \( \sigma_{ap} \) is the applied stress, and FS is the factor of safety. If the material properties are known \textit{precisely} and there is no variation in them—and the same holds for the load and geometry—then the part can be designed with a factor of safety of 1, the applied stress can be equal to the allowable strength, and the resulting design will not fail (just barely). However, not only are these measures not known with precision, they are not constant from sample to sample or use to use. In a statistical sense all these measures have some variance about their mean values (see App. B for the definitions of the mean and variance).
For example, typical material properties, such as ultimate strength, even when measured from the same bar of material, show a distribution of values (a variance) around a nominal mean of about 5 percent. This distribution is due to inconsistencies in the material itself and in the instrumentation used to take the data. If strength figures are taken from handbook values based on different samples and instrumentations, the variance of the values may be 15 percent or higher. Thus, the allowable strength must be characterized as a nominal or mean value with some statistical variation about it.

Even more difficult to establish are the statistics of the applied stress. The exact magnitude of the applied stress is a factor of the loading on the part (the forces and moments on the part), the geometry of the part at the critical location, and the accuracy of the analytic method used to determine the stress at the critical point due to the load.

The accuracy of the comparison of the applied stress to the allowable strength is a function of the accuracy and applicability of the failure theory used. If the stress is steady and the failure mode yielding, then accurate failure theories exist and can be used with little error. However, if the stress state is multiaxial and fluctuating (with a nonzero mean stress), there are no directly applicable failure theories and the error incurred in using the best available theory must be taken into account.

Beyond the above mechanical considerations, the factor of safety is also a function of the desired reliability for the design. As will be shown in Section C.3, the reliability can be directly linked to the factor of safety.

There are two ways to estimate the value of an acceptable factor of safety: the classical rule-of-thumb method (presented in Section C.2) and the probabilistic, or statistical, method of
relating the factor of safety to the desired reliability and to knowledge of the material, loading, and geometric properties (presented in Section C.3).

An additional note on standards. Most established design disciplines and companies have factors of safety used as standards. But often these values are based on lost or outdated material specifications and quality control procedures. At a minimum the following tools will help explore the basis of these standards; at a maximum they can be used to update them.

For example, the Jet Propulsion Laboratory, in its design of the Mars Rover used the factors of safety shown in Table C1.

<table>
<thead>
<tr>
<th></th>
<th>Qualified by testing</th>
<th>Not tested</th>
</tr>
</thead>
<tbody>
<tr>
<td>Metallic FS&lt;sub&gt;ultimate&lt;/sub&gt;</td>
<td>1.4</td>
<td>1.5</td>
</tr>
<tr>
<td>Metallic FS&lt;sub&gt;yield&lt;/sub&gt;</td>
<td>1.25</td>
<td>--</td>
</tr>
<tr>
<td>Composite</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The Factor of Safety for both yield and ultimate and for metallic and ceramic material is given. If the components have not been tested, the required Factor of Safety is much higher. Note that there is no value for composites yielding – they don’t.

C.2 THE CLASSICAL RULE-OF-THUMB FACTOR OF SAFETY

The factor of safety can be quickly estimated on the basis of estimated variations of the five measures previously discussed: material properties, stress, geometry, failure analysis, and desired reliability. The better known the material properties and stress, the tighter the tolerances, the
more accurate and applicable the failure theory, and the lower the required reliability, the closer
the factor of safety should be to 1. The less known about the material, stress, failure analysis, and
graphy and the higher the required reliability, the larger the factor of safety.

The simplest way to present this technique is to associate a value greater than 1 with each
of the measures and define the factor of safety as the product of these five values:

\[
FS = FS_{\text{material}} \cdot FS_{\text{stress}} \cdot FS_{\text{geometry}} \cdot FS_{\text{failure analysis}} \cdot FS_{\text{reliability}}
\]

Details on how to estimate these five values are given next. These values have been developed
by breaking down the rules given in textbooks and handbooks into the five measures and cross
checking the values with those from the statistical method described in the next section.

**Estimating the Contribution for the Material**

\( FS_{\text{material}} = 1.0 \)  
If the properties for the material are well known, if they have been
experimentally obtained from tests on a specimen known to be identical to
the component being designed and from tests representing the loading to
be applied.

\( FS_{\text{material}} = 1.1 \)  
If the material properties are known from a handbook or are
manufacturer’s values.

\( FS_{\text{material}} = 1.2–1.4 \)  
If the material properties are not well known.

**Estimating the Contribution for the Load Stress**

\( FS_{\text{stress}} = 1.0–1.1 \)  
If the load is well defined as static or fluctuating, if there are no
anticipated overloads or shock loads, and if an accurate method of
analyzing the stress has been used
FS_{stress} = 1.2–1.3 If the nature of the load is defined in an average manner, with overloads of 20–50 percent, and the stress analysis method may result in errors less than 50 percent

FS_{stress} = 1.4–1.7 If the load is not well known or the stress analysis method is of doubtful accuracy

*Estimating the Contribution for Geometry (Unit-to-Unit)*

FS_{geometry} = 1.0 If the manufacturing tolerances are tight and held well.

FS_{geometry} = 1.0 If the manufacturing tolerances are average.

FS_{geometry} = 1.1–1.2 If the dimensions are not closely held.

*Estimating the Contribution for Failure Analysis*

FS_{failure theory} = 1.0–1.1 If the failure analysis to be used is derived for the state of stress, as for uniaxial or multiaxial static stresses, or fully reversed uniaxial fatigue stresses.

FS_{failure theory} = 1.2 If the failure analysis to be used is a simple extension of the above theories, such as for multiaxial, fully reversed fatigue stresses or uniaxial nonzero mean fatigue stresses.

FS_{failure theory} = 1.3–1.5 If the failure analysis is not well developed, as with cumulative damage or multiaxial nonzero mean fatigue stresses.

*Estimating the Contribution for Reliability*
If the reliability for the part need not be high, for instance, less than 90 percent.

If the reliability is an average of 92–98 percent.

If the reliability must be high, say, greater than 99 percent.

These values are, at best, estimates based on a verbalization of the factors affecting the design combined and on experience with how these factors affect the design. The stress on a part is fairly insensitive to tolerance variances unless they are abnormally large. This insensitivity will be more evident in the development of the statistical factor of safety.

C.3 THE STATISTICAL, RELIABILITY-BASED, FACTOR OF SAFETY

C.3.1 Introduction

As can be appreciated, the classical approach to establishing factors of safety is not very precise and the tendency is to use it very conservatively. This results in large factors of safety and overdesigned components. Consider now the approach based on statistical measures of the material properties, of the stress developed in the component, of the applicability of the failure theory, and of the reliability required. This technique gives the designer a better feel of just how conservative, or nonconservative, he or she is being.

With this technique, all measures are assumed to have normal distributions (details on normal distributions appear in App. B). This assumption is a reasonable one, though not as accurate as the Weibull distribution for representing material-fatigue properties. What makes the normal distribution an acceptable representation of all the measures is the simple fact that, for most of them, not enough data are available to warrant anything more sophisticated. In addition,
the normal distribution is easy to understand and work with. In upcoming sections, each measure is discussed in terms of the two factors needed to characterize a normal distribution—the mean and the standard deviation (or variance).

The factor of safety is defined as the ratio of the allowable strength, $S_{al}$, to the applied stress, $\sigma_{ap}$. The allowable strength is a measure of the material properties; the applied stress is a measure of the stress (as a function of both the applied load and the stress analysis technique used to find the stress), the geometry, and the failure theory used. Since both of these measures are distributions, the factor of safety is better defined as the ratio of their mean values: $FS = \frac{\bar{S}_{al}}{\bar{\sigma}_{ap}}$.

Figure C.1 shows the distribution about the mean for both the applied stress and the allowable strength. There is an area of overlap between these two curves no matter how large the factor of safety is and no matter how far apart the mean values are. This area of overlap is where the allowable strength has a probability of being smaller than the applied stress; the area of overlap is thus the region of potential failure. Keeping in mind that areas under normal distribution curves represent probabilities, we see that this area of overlap then is the probability of failure (PF). The reliability, the probability of no failure, is simply $1 - PF$. Thus, by considering the statistical nature of these curves, the factor of safety is directly related to the reliability.

Figure C.1 Distribution of applied stress and allowable strength.

To develop this relationship more formally, we define a new variable, $z = S_{al} - \sigma_{ap}$, the difference between the allowable strength and the applied stress. If $z > 0$, then the part will not
fail. But failure will occur at $z \leq 0$. The distribution of $z$ is also normal (the difference between two normal distributions is also normal), as shown in Fig. C.2. The mean value of $z$ is simply $\bar{z} = \bar{S}_a - \bar{\sigma}_a$. If the allowable strength and the applied stress are considered as independent variables (which is the case), then the standard deviation of $z$ is

$$\rho_z = \sqrt{\rho_{\bar{S}_a}^2 + \rho_{\bar{\sigma}_a}^2}$$

Figure C.2 Distribution of $z$.

Normalizing any value of $z$ by subtracting the mean and dividing by the standard deviation, we can define the variable $t_z$ as

$$t_z = \frac{(\bar{S}_a - \bar{\sigma}_a) - (\bar{S}_a - \bar{\sigma}_a)}{\sqrt{\rho_{\bar{S}_a}^2 + \rho_{\bar{\sigma}_a}^2}}$$

The variable $t_z$ has a mean value of 0 and a standard deviation of 1. Since failure will occur when the applied stress is greater than the allowable stress, a critical point to consider is when $z = 0$, $S_{al} = \sigma_{ap}$. So, for $z = 0$,

$$t_{z=0} = \frac{-(\bar{S}_a - \bar{\sigma}_a)}{\sqrt{\rho_{\bar{S}_a}^2 + \rho_{\bar{\sigma}_a}^2}}$$

Thus, any value of $t$ that is calculated to be less than $t_{z=0}$ represents a failure situation. The probability of a failure then is $Pr(t < t_{z=0})$, which, assuming the normal distribution, can be found directly from a normal distribution table. If the distributions of the applied stress and the allowable strength are known, $t_{z=0}$ can be found from the above equation and the probability of failure found from normal distribution tables. Finally, the reliability is 1 minus the probability of failure; thus $R = 1 - Pr(t_z \leq t_{z=0})$. To make using normal distribution tables (App. B) easier by
utilizing the symmetry of the distribution, we can drop the minus sign on the above equation and consider values of $t_z > t_{z=0}$ to represent failure. Some values showing the relation of $t_{z=0}$ to reliability are given in Table C.2.

To reduce the equations to a usable form in which the factor of safety is the independent variable, we rewrite the previous equation, dividing by the mean value of the applied stress and using the definition of the factor of safety:

$$t_{z=0} = \frac{FS - 1}{\sqrt{FS^2(\rho_{al}/\bar{S}_a)^2 + (\rho_{ap}/\bar{\sigma}_{ap})^2}}$$

With $t_{z=0}$ directly dependent on the reliability, there are four variables related by this equation: the reliability, the factor of safety, and the coefficients of variation (standard deviation divided by the mean) for the allowable and applied stresses. In the development here the unknown will be the factor of safety. Thus the final form of the statistical factor of safety equation is

$$FS = 1 + t_{z=0} \sqrt{(\rho_{al}/\bar{S}_a)^2 + (\rho_{ap}/\bar{\sigma}_{ap})^2 - t_{z=0}^2(\rho_{ap}/\bar{\sigma}_{ap})^2(\rho_{al}/\bar{S}_a)^2 \over 1 - t_{z=0}^2(\rho_{al}/\bar{S}_a)^2} \quad \text{[C.1]}$$

<table>
<thead>
<tr>
<th>$R$</th>
<th>$t_{z=0}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.50</td>
<td>0.00</td>
</tr>
<tr>
<td>0.90</td>
<td>1.28</td>
</tr>
<tr>
<td>0.95</td>
<td>1.64</td>
</tr>
<tr>
<td>0.99</td>
<td>2.33</td>
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</table>
Before proceeding with details into the development of the applied stress and allowable strength coefficients of variation, let us look at an example of the use of the above equations. Say that the allowable strength coefficient of variation (see Sec. C.3.2) is 0.08 (the standard deviation is 8 percent of the mean value), the applied stress coefficient (see Sec. C.3.3) is 0.20 and the desired reliability is 95 percent. Using Table C.2, a 95 percent reliability gives $t_{z=0} = 1.64$. Thus, using Eq. [C.1], the design factor of safety can be computed to be 1.37. If the reliability is increased to 99 percent, the design factor of safety increases to 1.55. These design factor of safety values are not dependent on the actual values of the material properties or the stresses in the material but only on their statistics and the reliability and applicability of the failure theory. This is a very important point.

C.3.2 The Allowable Strength Coefficient of Variation

Measured material properties, such as the yield strength, the ultimate strength, the endurance strength, and the modulus of elasticity, all have distributions about their means. This is evident in Fig. B.1, which shows the result of static tests on 913 different samples of 1035 steel as hot-rolled, round bars, 1 to 9 in. diameter. Although not perfectly normal, as shown by the fit of the data on the normal distribution paper, an approximation to the straight line is not bad. Not all kinds of data fit this well. Typically, fatigue data and data on ceramic material tend not to be as evenly distributed and are better represented by a skewed distribution, such as the Weibull distribution. (Unfortunately, the four factors needed to represent the Weibull distribution have only been determined for a limited number of materials.) However, the adequacy and simplicity of the normal distribution make it the best choice for representing the material properties here. From Fig. B.2, the mean ultimate strength is 86 kpsi and its standard deviation is 4 kpsi. Note
that the standard deviation is 4.6 percent of the mean, so the coefficient of variation for 1033 hot-rolled steel is 0.046. Unfortunately, it is not always simple to find the standard deviation for the material properties. In looking up the ultimate stress for 1035 HR in standard design books, the following values were found: 72, 85, 72, 82, and 67 kpsi. From this limited sample, the mean value is 75.6 kpsi with a standard deviation of 6.08 kpsi, resulting in a coefficient of variation of 0.80 (6.08/75.6). If the heat treatment on the material or the exact composition of the material are unknown, the deviation can be much higher.

The allowable stress may be based on the yield, ultimate, or endurance strengths or some combination of them, depending on the failure criteria used. In the formulation above the allowable stress appears only as a ratio of standard deviation to average value. For most materials this ratio, irrespective of which allowable stress is considered, is in the range of 0.05–0.15. It is recommended that the statistics for the strength that best represent the nature of the failure be used. For example, in cases of nonzero-mean fluctuating stresses, the allowable stress coefficient for the endurance limit should be used if the mean is small relative to the amplitude, and the coefficient for the ultimate should be used if the mean is large relative to the amplitude. Any complexity beyond this is not warranted.

C.3.3 The Applied Stress Coefficient of Variation

The applied stress coefficient of variation is somewhat more difficult to develop. The statistics of the geometry and the load obviously affect the statistics of the applied stress, as does the accuracy of the method used to find the stress. Additionally, a measure of the accuracy of the failure analysis method to be used will also be reflected in the statistics of the applied stress.
(This measure could be taken into account elsewhere, but it is convenient to consider it as a correction on the applied stress.)

To see how these various factors are combined to form the applied stress coefficient of variation, consider the following example. (The statistics for the stress analysis technique and the failure theory accuracy will be included later in the example.) Consider a round, axially loaded uniform bar. In this bar the average maximum stress is given by the ratio of the average maximum force divided by the average area:

\[ \sigma_{ap} = \frac{F}{\pi r^2} \]

The standard deviation of the stress is a function of the independent statistics of the geometry and the load. Using standard, normal-distribution relations,

\[ \frac{\rho_{ap}}{\sigma_{ap}} = \sqrt{4 \left( \frac{\rho_r}{I} \right)^2 + \left( \frac{\rho_F}{F} \right)^2} \]

Thus, the applied-stress coefficient of variation is written in terms of the coefficients of variation for the geometry \( (\rho_r/I) \) and the loading \( (\rho_F/F) \). In general, the same form can be derived for any loading and shape. No matter whether it is normal or shear, the stress will have the form of force/area, and area always has units of length squared.

In many applications the magnitude of load forces and moments are well known through either experience or measurement. Essentially, two types of loads are considered here: static loads and fatigue, or fluctuating, loads. Regardless of which type of loading is considered, the exact magnitude of the forces and moments may have to be estimated. The determination of the statistical factor of safety takes into account the confidence in this estimation. This approach is much like that used in project planning (PERT) and requires the designer to make three estimates
of the load: an optimistic estimate \( o \); a most likely estimate \( m \); and a pessimistic estimate \( p \).

From these three the mean \( \bar{m} \), standard deviation \( \rho \), and coefficient of variation can be found:

\[
\bar{m} = \frac{1}{5} (o + 4m + p)
\]
\[
\rho = \frac{1}{5} (p - o)
\]
\[
\frac{\rho}{\bar{m}} = \frac{p - o}{o + 4m + p}
\]

These equations are based on a beta distribution function rather than a normal distribution. However, if the most likely estimate is the mean load, and the optimistic and pessimistic estimates are the mean ±3 standard deviations, then the beta distribution reduces to the normal distribution. The beauty of this is that an estimate of the important statistics can be made even if the distribution of the estimates is not symmetrical. For example, suppose the maximum load on a bracket is quoted as a force of 25,000 N. This may just be the most likely estimate. There is a possibility that the maximum load may be as low as 15,000 N or, because of light shock loading, the force may be as high as 50,000 N. Thus, from the formulas above, the expected value is 27,500 N, the standard deviation is 5833 N, and the coefficient of variation is 0.21. If the optimistic load had been 0, no load at all, the expected load would be 25,000 N and the standard deviation 8333 N. In this case the pessimistic and optimistic estimates are ±3 standard deviations from the expected or mean value. The coefficient of variation is 0.33, reflecting the wider range of estimates. Note again that the load coefficient of variation is independent of the absolute value of the load itself and only gives information on its distribution.

The hardest factor to take into account in failure analysis is the effect of shock loads. In the example just given the potential maximum load was double the nominal value. Without dynamic modeling there is no way to find the effect of shock loads on the state of stress. These choices are suggested:
• If the load is smoothly applied and released, use a ratio of optimistic: most likely: pessimistic of 1:1:2 or 1:2:4.

• If the load gives moderated shocks, use a ratio of optimistic: most likely: pessimistic of 1:1:4 or 1:4:16. Examples of moderate shock applications are blowers, cranes, reels, and calenders.

• If the load gives heavy shocks, use a ratio of optimistic: most likely: pessimistic of 1:1:10 or 1:10:100. Examples of heavy shock applications are crushers, reciprocating machinery, and mixers.

The geometry of the part is important in that, in combination with the load, the geometry determines the applied stress. Normally, the geometry is given as nominal dimensions with a bilateral tolerance (3.084 ± 0.010 inches). The nominal is the mean value, and the tolerance is usually considered to be three times the standard deviation. This implies that, assuming a normal distribution, 99.74 percent of all the samples will be within the limits of the tolerance. It is assumed that there is one dimension that is most critical to the stress, and the coefficient of variation for this dimension is used in the analysis. For the example just considered, the coefficient of variation is 0.0011 (0.010/(3 • 3.084)), which is an order of magnitude smaller than that for the load. This is typical for most tolerances and loadings.

Using the discussed examples, the applied stress coefficient of variation is

\[ \rho_{ap} = \sqrt{4(0.0011)^2 + 0.21^2} \approx 0.21 \]

Note the lack of sensitivity to the tolerance.
The above does not take into account the accuracy of the stress analysis technique used to find the stress state from the loading and geometry nor the adequacy of the failure analysis method. To include these factors, the allowable strength needs to be compared with the calculated applied stress corrected for the stress analysis and the failure analysis accuracy. Thus,

\[ \sigma_{ap} = \sigma_{calc} \times N_{sa} \times N_{fa} \]

where \( N_{sa} \) is a correction multiplier for the accuracy of the stress analysis technique and \( N_{fa} \) is a correction multiplier for the failure analysis accuracy.

If the two corrections are assumed to have normal distributions, they can be represented as coefficients of variation. With the product of normally distributed independent standard deviations being the square root of the sum of the squares (see App. B), we have for the applied stress coefficient of variation

\[ \frac{\rho_{ap}}{\sigma_{ap}} = \sqrt{\frac{\rho_r}{F} \rho_r^2 + \frac{\rho_F}{F} \rho_F^2 + \left( \frac{\rho_{sa}}{N_{sa}} \right)^2 + \left( \frac{\rho_{fa}}{N_{fa}} \right)^2} \]

[C.2]

This is the same as before, with the addition of the coefficient of variation \( s \) for the stress analysis method and for the failure theory.

The coefficient of variation for the stress analysis method can be estimated using the same technique as for estimating the statistics on the loading—namely, estimate an optimistic, pessimistic, and most likely value for the stress, based on the most likely load. Again, consider a load of 25,000 N (the most likely estimate of the maximum load). Assume that at the critical point the normal stress caused by this load is 40.9 kpsi (282.0 Mpa), with a stress concentration factor of 3.55. The most likely normal stress is the product of the load and the stress concentration factor, 145 kpsi. However, confidence in the method used to find the nominal
stress and the stress concentration factor is not high. In fact, the maximum stress may really be as high as 160 kpsi or as low as 140 kpsi. With these two values as the pessimistic and the optimistic estimates, the coefficient of variation is calculated at 0.023. For strain gauge data or other measured results, the stress analysis method coefficient of variation will be very small and can, like the geometry statistics, be ignored.

The adequacy of the failure analysis technique, as discussed in the development of the classical factor of safety method, has a marked effect on the design factor of safety. On the basis of experience and the limited data in the references, the coefficient of variations recommended for the different types of loadings are

- Static failure theories: 0.02
- Fully reversed uniaxial infinite life fatigue failure theory: 0.02
- Fully reversed uniaxial finite life fatigue failure theory: 0.05
- Nonzero mean uniaxial fatigue failure theory: 0.10
- Fully reversed multiaxial fatigue failure theory: 0.20
- Nonzero mean multiaxial fatigue failure theory: 0.25
- Cumulative damage load history: 0.50

These values imply that for well-defined failure analysis techniques, where the failure mode is identical to that found with the allowable strength material test, the standard deviation is small, namely, 2 percent of the mean. When the failure theory is comparing a dissimilar applied stress state to an allowable strength, the margin for error increases. The rule used in cumulative
damage failure estimation can be off by as much as a factor of 2 and is therefore used with high uncertainty.

C.3.4 Steps for Finding the Reliability-Based Factor of Safety

We can summarize the method discussed in the previous two sections as an eight-step procedure:

**Step 1: Select reliability.** From Table C.2, find the value of $t_{z=0}$ for the desired reliability.

**Step 2: Find the allowable strength coefficient of variation.** This can be found experimentally or by following the following rules of thumb. If the material properties are well known, use a coefficient of 0.05; if the material properties are not well known, use a coefficient of 0.01–0.15.

**Step 3: Find the critical dimension coefficient of variation.** This value is generally small and can be ignored except when the variation in the critical dimensions are large because of manufacturing, environmental, or aging effects.

**Step 4: Find the load coefficient of variation.** This is an estimate of how well the maximum loading is known. It can be estimated using the PERT method given in Sec. C.3.3.

**Step 5: Find the accuracy of the stress analysis coefficient of variation.** Even though the variation of the load was taken into account in step 4, knowledge about the effect of the load on the structure is a separate issue. The stress due to a well-known load may be hard to determine because of complex geometry. Conversely, the stress caused by a poorly known load on a simple structure is no more poorly known than the load itself. This measure then takes into account how well the stress can be found for a known load.
Step 6: Find the failure analysis technique coefficient of variation. Guidance for this is given near the end of Sec. C.3.2.

Step 7: Calculate the applied stress coefficient of variation. This is found using Eq. [C.2].

Step 8: Calculate the factor of safety. This is found using Eq. [C.1].

C.4 SOURCES
