1
General Topics

Design Philosophy

In general, pressure vessels designed in accordance with the ASME Code, Section VIII, Division 1, are designed by rules and do not require a detailed evaluation of all stresses. It is recognized that high localized and secondary bending stresses may exist but are allowed for by use of a higher design margin and design rules for details. It is required, however, that all loadings (the forces applied to a vessel or its structural attachments) must be considered. (See Reference 1, Para. UG-22).

While Section VIII, Division 1 provides formulas for thickness and stress of basic components, it is up to the designer to select appropriate analytical procedures for analyzing other components and to combine the calculated stresses in a manner appropriate with the intended operation of the equipment for an economical and safe design. For the supporting structures, the designer must also abide to the load combinations determined by the applicable building code.

Section VIII, Division 1 establishes allowable stresses by stating in Para. UG-23(c) that the maximum general primary membrane stress must be less than allowable stresses outlined in material sections. Further, it states that the maximum primary membrane stress *plus* primary bending stress may not exceed 1.5 times the allowable stress of the material sections. In other sections of the Code, including both design-by-rules and design-by-analysis methods in Section VIII, Division 2, higher stress levels are permitted if appropriate analyses are made. These higher allowable stresses clearly indicate that different stress levels for different stress categories are acceptable.

Section VIII, Division 2 also has a section outlining 'design-by-rules' requirements (Part 4). Though also classified as design-by-rules, the design margin is lower than in Division 1 and as a result will require more analysis. Part 4 clearly indicates that if rules are not provided for a specific detail, geometry, or loading, then an analysis in accordance with Part 5, or 'design-by-analysis', shall be performed. Most vessels designed to Part 4 will use both the rules in Part 4 as well as specific procedures in Part 5. Both Part 4 and Part 5 contain load combinations for the pressure envelope. The load combinations in Part 4 are based on the allowable stress design (ASD) load combinations from ASCE 7, with non-applicable loads removed.

It is general practice when doing a more detailed stress analysis to apply higher allowable stresses. In effect, the detailed evaluation of stresses permits substituting knowledge of localized stresses and the use of higher allowable stresses in place of the larger design margin used by the Code. This larger design margin really reflected lack of knowledge about actual stresses.

A calculated value of stress means little until it is associated with its location and distribution in the vessel and with the type of loading by which it was produced. Different types of stresses have different degrees of significance.

The designer must be familiar with the various types of loadings and their stresses in order to accurately understand the results of the analysis. The designer must also consider the stress categories to determine the allowable stress limits.

The following sections will provide the fundamental knowledge for determining and understanding the results of an analysis. The topics covered in Chapter 1 form the basis by which the rest of the book is to be used. A section on special problems and considerations is included to alert the designer to more complex problems that exist.

Stress Analysis

Stress analysis is the determination of the relationship between external forces applied to a vessel and the corresponding stresses. The emphasis of this book is not how to do stress analysis in particular, but rather how to analyze vessels and their component parts in an effort to arrive at an economical and safe design—the difference being that we analyze stresses where necessary to determine thickness of material and sizes of members. We are not so concerned with building mathematical models as with providing a step-by-step approach to the design of ASME Code vessels. It is not necessary to find every stress but rather to know the governing stresses and how they relate to the vessel or its respective parts, attachments, and supports.

The starting place for stress analysis is to determine all the design conditions for a given problem and then determine all the related external forces. We must then relate these external forces to the vessel parts which must resist them to find the corresponding stresses. By isolating the causes (loadings), the effects (stresses) can be more accurately determined.

The designer must also be keenly aware of the types of loads and how they relate to the vessel as a whole. Are the effects long or short term? Do they apply to a localized portion of the vessel or are they uniform throughout?

How these stresses are interpreted and combined, what significance they have to the overall safety of the vessel, and what allowable stresses are applied will be determined by three things:

1. The strength/failure theory utilized.
2. The types and categories of loadings.
3. The hazard the stress represents to the vessel.

Membrane Stress Analysis

Pressure vessels commonly have the form of spheres, cylinders, cones, ellipsoids, tori, or composites of these. When the thickness is small in comparison with other dimensions ($R_m/t \geq 10$), vessels are referred to as membranes and the associated stresses resulting from the contained pressure are called membrane stresses. These membrane stresses are average tension or compression stresses. They are assumed to be uniform across the vessel wall and act tangentially to its surface. The membrane or wall is assumed to offer no resistance to bending. When the wall offers resistance to bending, bending stresses occur in addition to membrane stresses.

In a vessel of complicated shape subjected to internal pressure, the simple membrane-stress concepts do not suffice to give an adequate idea of the true stress situation. The types of heads closing the vessel, effects of supports, variations in thickness and cross section, nozzles, external attachments, and overall bending due to weight, wind, and seismic activity all cause varying stress distributions in the vessel. Deviations from a true membrane shape set up bending in the vessel wall and cause the direct loading to vary from point to point. The direct loading is diverted from the more flexible to the more rigid portions of the vessel.

In any pressure vessel subjected to internal or external pressure, stresses are set up in the shell wall. The state of stress is triaxial and the three main defining stresses are:

σ_x = longitudinal/meridional stress

σ_ϕ = circumferential/latitudinal stress

σ_r = radial stress

In addition, there may be bending and shear stresses. The radial stress is a direct stress, which is a result of the pressure acting directly on the wall, and causes a compressive stress equal to the pressure on the surface on which it acts. In thin-walled vessels this stress is so small compared to the circumferential and longitudinal stresses that it is generally ignored. Thus we assume for purposes of analysis that the state of stress is biaxial. This greatly simplifies the method of combining stresses in comparison to triaxial stress states. For thick walled vessels ($R_m/t < 10$), the radial stress cannot be ignored and formulas are quite different from those used in finding membrane stresses in thin shells.

Since ASME Code, Section VIII, Division 1 is basically for designing by rules; a higher design margin and specific rules are used to allow for the high localized and secondary bending stresses at safe levels consistent with experience. This higher design margin can impose a penalty on design but requires much less analysis. The design techniques outlined in this book are a compromise between finding all stresses and utilizing minimum code formulas. This additional knowledge of stresses warrants the use of higher allowable stresses in some cases, while meeting the requirements that all loadings be considered.

In conclusion, "membrane stress analysis" is not completely accurate but allows certain simplifying assumptions to be made while maintaining a fair degree of accuracy. The main simplifying assumptions are that the stress is biaxial and that the stresses are uniform across the shell wall. For thin-walled vessels these assumptions have proven themselves to be reliable. No vessel meets the criteria of being a true membrane, but we can use this tool with a reasonable degree of accuracy.

Stress/Failure Theories

As stated previously, stresses are meaningless until compared to some stress/failure theory. The significance of a given stress must be related to its location in the vessel and the failure mode being evaluated. Historically, various "theories" have been derived to combine and measure stresses against the potential failure mode. A number of stress theories, also called "yield criteria", are available for describing the effects of multi-axial stresses. For purposes of this book, as these failure theories apply to pressure vessels, three theories will be discussed. They are the "maximum principal stress theory", the "maximum shear stress theory", and the "distortion energy theory".

Maximum Principal Stress Theory

This theory is the oldest, most widely used and simplest to apply. Both ASME Code, Section VIII, Division 1, and Section I use the maximum principal stress theory as a basis for design. This theory simply asserts that yielding occurs when the largest principal stress equals the yield strength. Stresses in the other directions are disregarded for this criteria. Only the maximum principal stress must be determined to apply this criterion. This theory is used for biaxial states of stress assumed in a thin-walled pressure vessel. As will be shown later it is unconservative in some instances of biaxial stress states. While the maximum principal stress theory more accurately predicts failure in brittle materials, it is not always accurate for ductile materials. Ductile materials often fail along lines 45° to the applied force by shearing, long before the tensile or compressive stresses are a maximum.

Where $\sigma 1 > \sigma 2$, this theory states that yielding will occur when

$\sigma_1 = F_y$

This theory is illustrated graphically for the four states of biaxial stress shown in Figure 1-1.

It can be seen that uniaxial tension or compression lies on the two axes. Inside the box (outer boundaries) is the elastic range of the material. Yielding is predicted for stress combinations by the outer line.

Maximum Shear Stress Theory

This theory asserts that yielding occurs when the largest difference of shear stress equals the shear yield strength. According to this theory, yielding will start at a point when the maximum shear stress at that point reaches one-half of the uniaxial yield strength, F_y. Thus for a biaxial state of stress where $\sigma_1 > \sigma_2$, the maximum shear stress will be $(\sigma_1 - \sigma_2)/2$.

This theory states that yielding will occur when

$$\frac{\sigma_1 - \sigma_2}{2} = \frac{F_y}{2}$$

This theory closely approximates experimental results in ductile materials and is also easy to use. This theory

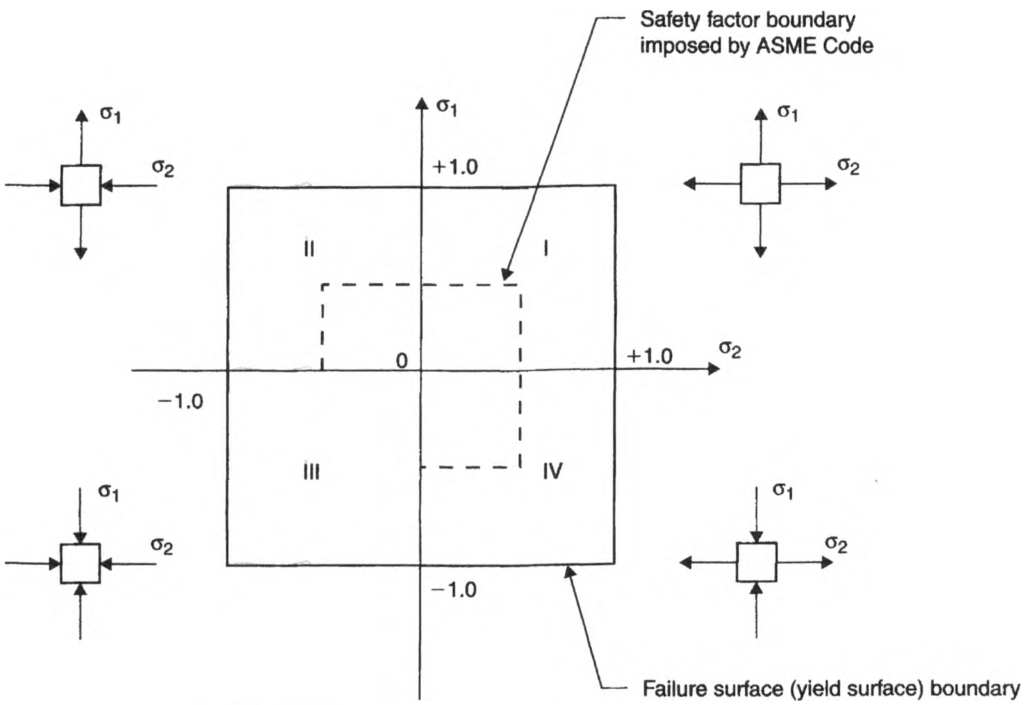


Figure 1-1. Graph of maximum stress theory. Quadrant I: biaxial tension; Quadrant II: tension; Quadrant III: biaxial compression; Quadrant IV: compression.

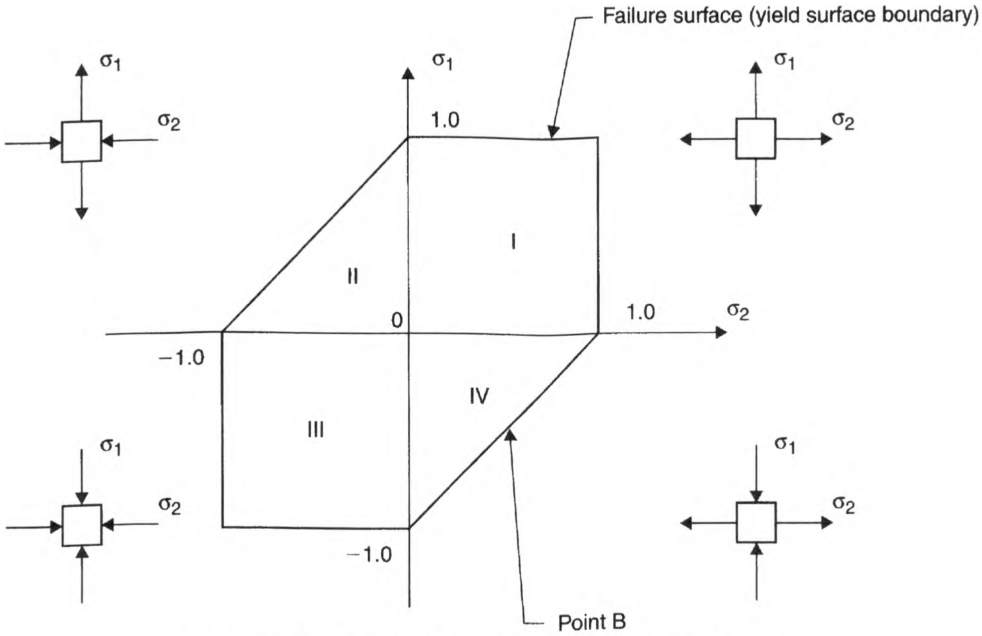


Figure 1-2. Graph of maximum shear stress theory.

also applies to triaxial states of stress. In a triaxial stress state, this theory predicts that yielding will occur whenever one-half the algebraic difference between the maximum and minimum stress is equal to one-half the yield stress. Where $\sigma_1 > \sigma_2 > \sigma_3$, the maximum shear stress is $(\sigma_1 - \sigma_3)/2$.

Yielding will begin when

$$\frac{\sigma_1 - \sigma_3}{2} = \frac{F_y}{2}$$

This theory is illustrated graphically for the four states of biaxial stress in Figure 1-2.

Both the pre-2007 ASME Section VIII, Division 2 and Part 4 of the new Section VIII, Division 2 utilize the maximum shear stress criterion for determining the primary thicknesses of a shell under internal pressure.

Distortion Energy Theory

This theory asserts that the total strain energy is composed of two parts; the strain energy required for hydrostatic strain and the strain energy required for distortion. In this theory, it is assumed that yielding will begin when the distortion component is equal to the uniaxial yield strength, F_y. Where $\sigma_1 > \sigma_2 > \sigma_3$, yielding will occur when

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

This theory is illustrated graphically for the four states of biaxial stress shown in Figure 1-3. This theory correlates even better with ductile test specimens than the maximum shear stress theory.

The new ASME Section VIII, Division 2, Part 5 utilizes the distortion energy theory to establish the equivalent stress in an elastic analysis where in the pre-2007 edition this was done with the maximum shear stress theory.

Comparison of the Three Theories

Figure 1-4 is an overlay of Figures 1-1, 1-2, and 1-3 and will illustrate the major differences between the three theories. For the case of biaxial stress state, all three theories are in agreement where their bounded areas graphically overlap. The bounded area by each theory indicates the elastic range of which there is no yielding, however, it is important to note that in quadrants II and IV that the maximum principal stress theory provides unconservative results. For example, consider point B at the midpoint of the line in Figure 1-2. It shows $\sigma_2 = (-)\sigma_1$; therefore the shear stress is equal to $(\sigma_2 - (-\sigma_1))/2$, which equals $(\sigma_2 + \sigma_1)/2$ or one-half the stress which would

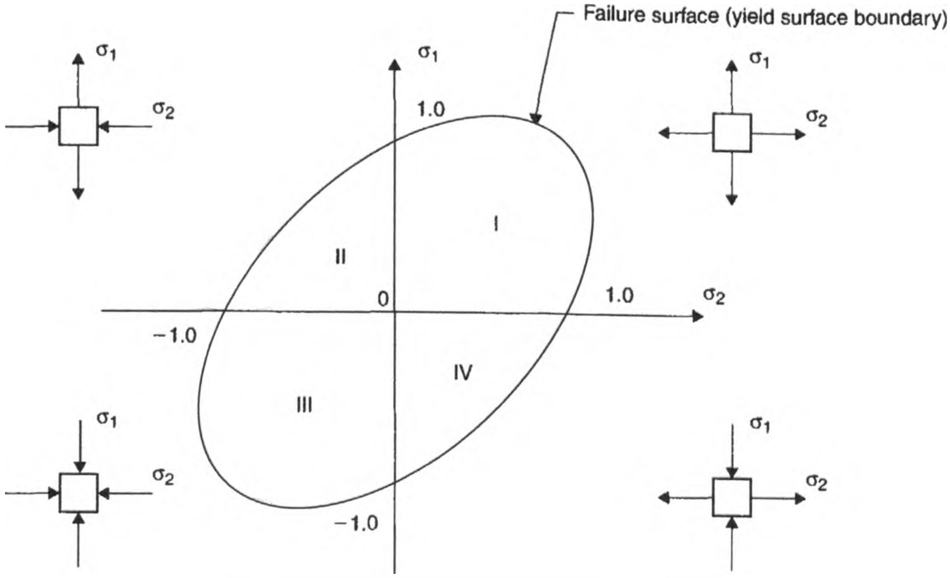


Figure 1-3. Graph of distortion energy theory.

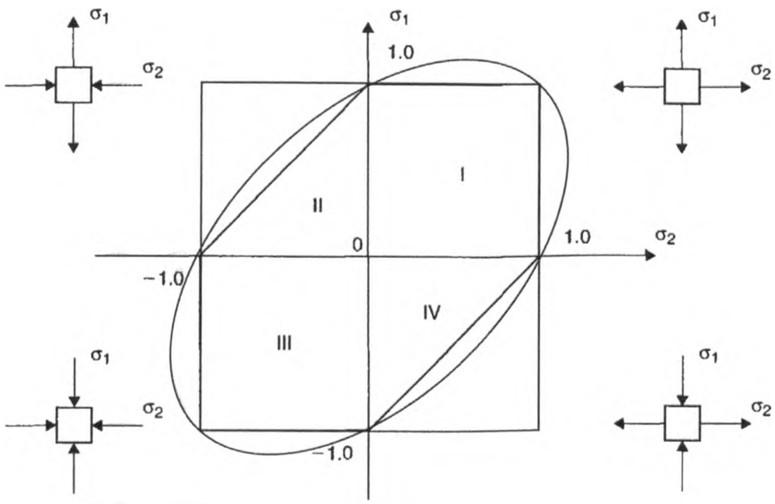


Figure 1-4. Comparison of the three theories.

cause yielding as predicted by the maximum principal stress theory.

For simple analysis upon which the thickness formulas for ASME Code, Section I or Section VIII, Division 1, are based, it makes little difference whether the maximum principal stress theory or maximum shear stress theory is used. For example, according to the maximum principal stress theory, for a cylinder only under internal pressure the controlling stress governing the thickness of a cylinder is σ_ϕ, the circumferential stress, since it is the largest of the three principal stresses. According to the maximum shear stress theory, the controlling stress would be one-half the algebraic difference between the maximum and minimum stress (this was defined as the stress intensity in the pre-2007 Section VIII, Division 2), and according to the distortion energy theory the equivalent stress is as shown:

- The maximum principal stress is the circumferential stress, σ_ϕ
  $\sigma_\phi = \sigma_1 = PR/t$
- The middle principal stress is the longitudinal stress, σ_r
  $\sigma_x = \sigma_2 = PR/2t$

- The minimum principal stress is the radial stress, σ_r
  $\sigma_r = \sigma_3 = (-)P$

Therefore, the stress used in the maximum principal stress theory is

$$\sigma_\phi = \frac{PR}{t},$$

the calculated stress used in the maximum shear stress theory (stress intensity) is

$$\sigma_\phi - \sigma_r = \frac{PR}{t} + P,$$

and for information, the equivalent stress used in the distortion energy theory is

$$\sigma_e = \frac{1}{\sqrt{2}}\left[(\sigma_\phi - \sigma_x)^2 + (\sigma_x - \sigma_r)^2 + (\sigma_r - \sigma_\phi)^2\right]^{0.5}$$

Note that in the derivation of the shell thickness calculation in the pre-2007 Section VIII, Division 2, $\sigma_r = (-)P/2$ was used as a representation of the average stress through the thickness. Also, for information, if σ_r is assumed to be zero for calculation purposes, $\sigma_e \approx 0.866$ PR/t for the distortion energy theory.

For a cylinder where P = 300 psi, R = 30 in., and t = .5 in., the three theories would compare as follows:

- *Maximum principal stress theory*
  $\sigma_\phi = \sigma_1 = PR/t = 300(30)/.5 = 18{,}000$ psi
- *Maximum shear stress theory*
  $\sigma_\phi - \sigma_r = PR/t + P = 300(30)/.5 + 300 = 18{,}300$ psi
- *Distortion energy theory*
  $\sigma_e = 15{,}850$ psi

Three points are obvious from a comparison of the maximum principal stress theory and the maximum shear stress theory:

1. For thin-walled pressure vessels, the two theories yield approximately the same results.
2. For thin-walled pressure vessels the radial stress is so small in comparison to the other principal stresses that it can be ignored and a state of biaxial stress is assumed to exist.
3. For thick-walled vessels ($R_m/t < 10$), the radial stress becomes significant in defining the ultimate failure of the vessel. The maximum principal stress theory is unconservative for designing these vessels. For this reason, this book has limited most of its application to thin-walled vessels where a biaxial state of stress is assumed to exist.

The results of the equivalent stress calculation from the distortion energy theory indicate that reduced shell thickness values may be obtained if a more rigorous analysis is performed. This is the basis for part of the design-by-analysis section in Section VIII, Division 2.

Failures in Pressure Vessels

Vessel failures can be grouped into four major categories, which describe *why* a vessel failure occurs. Failures can also be grouped into types of failures, which describe how the failure occurs. Each failure has a why and how to its history. It may have failed *through* corrosion fatigue *because* the wrong material was selected. The designer must be as familiar with categories and types of failure as with categories and types of stress and loadings. Ultimately they are all related.

Categories of Failures

1. *Material*—Improper selection of material; defects in material.
2. *Design*—Incorrect design data; inaccurate or incorrect design methods; inadequate shop testing.
3. *Fabrication*—Poor quality control; improper or insufficient fabrication procedures including welding; heat treatment or forming methods.
4. *Service*—Change of service condition by the user; inexperienced operations or maintenance personnel; upset conditions. Some types of service which require special attention both for selection of material, design details, and fabrication methods are as follows:
   a. Lethal
   b. Fatigue (cyclic)
   c. Brittle (low temperature)
   d. High temperature
   e. High shock or vibration
   f. Vessel contents
      - Hydrogen
      - Ammonia

- Compressed air
- Caustic
- Chlorides

Types of Failure Modes

1. *Elastic deformation*—Elastic instability or elastic buckling, must be evaluated by considering vessel geometry, stiffness as well as properties of materials.
2. *Excessive plastic deformation*—The primary stress limits as outlined in ASME Section VIII, Division 2, are intended to prevent excessive plastic deformation.
3. *Brittle fracture*—Can occur at low or intermediate temperatures. Brittle fractures have occurred in vessels made of low carbon steel in the 40°–50°F range during hydrotest where minor flaws exist. This is addressed greatly in material toughness.
4. *Stress rupture*—Italicized values in Section II, Part D indicate that allowable stress values are governed by time-dependent properties, e.g. stress rupture and creep rate.
5. *Plastic instability*—Incremental collapse; incremental collapse is cyclic strain accumulation or cumulative cyclic deformation. Cumulative damage leads to instability of vessel by plastic deformation. The primary plus secondary limits are intended to preclude any ratcheting and validate the use of elastic analysis.
6. *High strain*—Low cycle fatigue is strain-governed and occurs mainly in lower-strength/high-ductile materials. The peak stresses are used to evaluate this condition.
7. *Stress corrosion*—It is well known that chlorides cause stress corrosion cracking in stainless steels; likewise caustic service can cause stress corrosion cracking in carbon steels. Material selection is critical in these services.
8. *Corrosion fatigue*—Occurs when corrosive and fatigue effects occur simultaneously. Corrosion can reduce fatigue life by pitting the surface and propagating cracks. Material selection and fatigue properties are the major considerations.

Creep rupture from long term loading, creep-fatigue, and creep-buckling should also be addressed and procedures to do so within the Code have been around for some time now. In the past, at least some portions of the Code were limited to temperatures with the intent to avoid any time-dependent effects on the material, meaning that maximum permitted temperatures for materials were kept below the creep range.

In dealing with these various modes of failure, the designer must have an understanding and a picture of the state of stress in the various parts. It is against these failure modes that the designer must compare and interpret stress and strain values. Setting allowable stresses is not enough, as in the case of elastic instability one must consider geometry, stiffness, and the properties of the material. Material selection is a major consideration when related to the type of service. Design details and fabrication methods are as important as "allowable stress" in design of vessels for cyclic service. The designer and all those persons who ultimately affect the design must have a clear picture of the conditions under which the vessel will operate.

Loadings

Loadings or forces are the "causes" of stresses in pressure vessels. These forces and moments must be isolated both to determine *where* they apply to the vessel and *when* they apply to a vessel. Categories of loadings define where these forces are applied. Loadings may be applied over a large portion (general area) of the vessel or over a local area of the vessel. Remember both *general* and *local* loads can produce membrane and bending stresses. These stresses are additive and define the overall state of stress in the vessel or component. Stresses from local loads must be added to stresses from general loadings. These combined stresses are then compared to an allowable stress.

Consider a pressurized, vertical vessel bending due to wind, which has an inward radial force applied locally. The effects of the pressure loading are longitudinal and circumferential tension. The effects of the wind loading are longitudinal tension on the windward side and longitudinal compression on the leeward side. The effects of the local inward radial load are some local membrane stresses and local bending stresses. The local stresses would be both circumferential and longitudinal, tension on the inside surface of the vessel, and compressive on the outside. Of course the steel at any given point only sees a certain level of stress of the combined effect. It is the designer's job to combine the stresses from the various loadings to arrive at the worst probable combination of stresses utilizing both knowledge of the operation of the equipment and applicable load combinations, determine what section and part within the Code and corresponding failure theory to use, and compare the results to an acceptable stress level to obtain an economical and safe design.

This hypothetical problem serves to illustrate how categories and types of loadings are related to the stresses they produce. The stresses which are required for equilibrium of the vessel are primary stresses. The stresses due to pressure and wind are primary general membrane stresses since even if yielding occurred, redistribution of stresses would not be possible. These stresses should be limited to the Code allowable stress values, where increases for occasional loading may be allowed for certain sections of the Code.

On the other hand, the stresses from the inward radial load could be either a primary stress or secondary stress. It is a primary stress if it is produced from an unrelenting load or a secondary stress if produced by a relenting load. A general primary membrane stress will not redistribute upon yielding, whereas a primary local membrane stress will, and for a secondary stress the load will relax once slight deformation occurs.

This should make it obvious that the type and location of loading will determine the category of stress. This will be expanded upon later, but basically each combination of stresses (stress categories) will have different allowables, i.e.:

- Primary stress (P_m): $P_m < 1.0$ SE
- Primary local membrane (P_L): $P_L < 1.5$ SE
- Primary local membrane + primary bending ($P_L + P_b$): $P_L + P_b < 1.5$ SE
- Primary local membrane + primary bending + secondary ($P_L + P_b + Q$):
  $P_L + P_b + Q < 3$SE (or $2S_y$)

Whether a loading is steady, more or less continuous, or nonsteady, variable, or temporary will also have an effect on what level of stress will be acceptable. If in our hypothetical problem the loading had been pressure plus seismic plus local load, we would have a different case. Due to the relatively short duration of seismic loading, a higher "temporary" allowable stress would be acceptable. The vessel isn't expected to operate in an earthquake all the time though building codes are written such that vessels do not collapse in the event of an earthquake.

For *normal loads*, the vessel must support these loads more or less continuously during its useful life. As a result, the stresses produced from these loads must be maintained to an acceptable level.

For *occasional loads*, the vessel may experience some or all of these loadings at various times but not all at once and not more or less continuously. Therefore a temporarily higher stress is acceptable.

For *general loads* that apply more or less uniformly across an entire section, the corresponding stresses must be lower, since the entire portion of the vessel must support that loading.

For *local loads*, the corresponding stresses are confined to a small portion of the vessel and normally fall off rapidly in distance from the applied load. As discussed previously, pressurizing a vessel causes bending in certain components. But it doesn't cause the entire vessel to bend. The results are typically not as significant (except in cyclic service) as those caused by general loadings. Therefore a slightly higher allowable stress would be in order.

Loadings can be outlined as follows:

A. *Categories of loadings*
   1. *General loads*—Applied more or less continuously across a vessel section.
      a. Pressure loads—Internal or external pressure (design, operating, hydrotest, and hydrostatic head of liquid).
      b. Moment loads—Due to wind, seismic, erection, transportation.
      c. Compressive/tensile loads—Due to dead weight, installed equipment, ladders, platforms, piping, and vessel contents.
   2. Local loads—Due to reactions from supports, internals, attached piping, attached equipment, i.e., platforms, mixers, etc.
      a. Radial load—Inward or outward.
      b. Shear load—Longitudinal or circumferential.
      c. Torsional load.
      d. Moment load—Longitudinal or circumferential.
      e. Thermal load.

B. *Types of loadings*
1. *Normal*—Long-term duration, continuous.
   a. Internal/external pressure.
   b. Dead weight.
   c. Vessel contents.
   d. Loadings due to attached piping and equipment.
   e. Loadings to and from vessel supports.
   f. Thermal loads.
   g. Wind loads.
2. *Occasional loads*—Short-term duration; variable.
   a. Shop and field hydrotests.
   b. Earthquake.
   c. Erection.
   d. Transportation.
   e. Upset, emergency.
   f. Start up, shut down.

Table 1-1
Design load combinations

Load Combination		Thickness	Temperature	General Primary Membrane Allowable Stress	Description
1	P + Ps + D	Corroded	Design	SE	Pressure and weight
2	P + Ps + D + L	Corroded	Design	SE	Pressure, weight, and live loading
3	P + Ps + D + S	Corroded	Design	SE	Pressure, weight, and snow loading
4	P + Ps + D + 0.75L + 0.75S	Corroded	Design	SE	Pressure, weight, partial live load, and partial snow load
5	P + Ps + D + (W or 0.7E)	Corroded (4) (5)	Design	SE	Pressure, weight, and wind load or seismic load (see description of E)
6	P + Ps + D + 0.75(W or 0.7E) + 0.75L + 0.75S	Corroded (4) (5)	Design	SE	Pressure, weight, partial wind load or seismic load, partial live load, and partial snow load
7	0.6D + (W or 0.7E)	Corroded (4) (5)	Design	SE	Partial weight, and wind or seismic

P = design pressure
Ps = static head
D = dead load (e.g. weight of vessel, supports, internals, external appurtenances)
L = live load (e.g. appurtenance live loading)
E = earthquake load (strength level load)
W = wind load (e.g. wind load on vessel and appurtenances)
S = snow load (e.g. snow load on platforms)

1. For each load combination, the determination and evaluation of P = 0 (and Ps = 0) should be considered (e.g. hot shutdown).
2. For each load combination, the determination and evaluation of P = external pressure condition should be considered.
3. The effects of wind and seismic loading are not assumed to act concurrently.
4. For load combinations with wind loads, the possibility that a full wind load may occur while the vessel is empty should be considered.
5. For load combinations with seismic loads, it is typically conservative to use the uncorroded vessel weight in design load calculations and the corroded shell in stress calculations.

Stress

ASME Code, Section VIII, Division 1 vs. Division 2

ASME Code, Section VIII, Division 1 does not explicitly indicate the manner in which stresses should be combined but indicates that engineering judgment must be consistent with the philosophy of Division 1. ASME Code, Section VIII, Division 2, on the other hand, provides specific guidelines and stress categories, how they are combined, and the allowable stresses for each category and combination of categories. Part 5 of Division

2 is design-by-analysis whereas Division 1 and Part 4 of Division 2 are design-by-rules. Although stress analysis as utilized by Part 5 of Division 2 is beyond the scope of this book, the use of stress categories, definitions of stress, and allowable stresses is applicable.

Division 1 and the procedures outlined in this book consider a biaxial state of stress combined in accordance with the maximum principal stress theory. Division 2 considers triaxial stresses evaluated in accordance with the maximum shear stress theory and distortion energy theory. Just as you would not design a nuclear reactor to the rules of Division 1, you would not design an air receiver by the rules of Division 2. Each has its place and application. The following discussion on categories of stress and allowable stresses will utilize information from Division 2, which can be applied in general to all vessels.

Stress Categories

The shell thickness as computed by Code formulas for internal or external pressure alone is often not sufficient to withstand the combined effects of all other loadings. Detailed calculations consider the effects of each loading separately and then must be combined to give the total state of stress in that part.

Types of stress, stress categories, and allowable stresses are based on the type of loading that produced them *and* on the hazard they represent to the structure. Unrelenting loads produce primary stresses. Relenting loads (self-limiting) produce secondary stresses. Primary stresses must be kept lower than secondary stresses. Primary plus secondary stresses are allowed to be higher and so on. Before considering the combination of stress categories, we must first define the various *types* of stress and each *category*.

Types of Stress

There are many names to describe types of stresses. As these stresses apply to pressure vessels, we group all types of stress into three major classes of stresses, and subdivision of each of the groups is arranged according to their effect on the vessel. The following list of stresses describes types of stresses without regard to their effect on the vessel or component. They define a direction of stress or relate to the application of the load.

1. Tensile
2. Compressive
3. Normal
4. Shear
5. Membrane
6. Bending
7. Bearing
8. Axial
9. Discontinuity
10. Principal
11. Thermal
12. Tangential
13. Load controlled
14. Strain controlled
15. Circumferential
16. Longitudinal
17. Radial

Stress Categories

The foregoing list provides the categories and subcategories. It is, however, too general to provide a basis with which to combine stresses or apply allowable stresses. Stress categories are defined by the type of loading which produces them and the hazard they represent to the vessel.

1. *Primary stress*
   a. General membrane stress, P_m
   b. Local membrane stress, P_L
   c. Bending stress, P_b
2. *Secondary stress*
   a. Secondary membrane stress, Q_m
   b. Secondary bending stress, Q_b
3. *Peak stress,* F

Definitions and examples of these stresses follow.

Primary stresses: These stresses are normal or shear stresses which are required to satisfy equilibrium. They are produced by mechanical loads (load controlled) and when exceeding the yield strength can result in failure or gross distortion. The basic characteristic of a primary stress is that it is not self-limiting. Primary stresses are generally due to internal or external pressure or produced by sustained external forces and moments. Thermal stresses from thermal gradients and imposed displacements are never classified as primary stresses.

Primary general stresses are divided into membrane and bending stresses. The need for dividing primary general stress into membrane and bending is that the calculated value of a primary bending stress may be allowed to go to a higher value than that of a primary general membrane stress.

Primary general membrane stress, P_m. This stress is the average primary stress across a solid section, is produced by pressure or mechanical loads, and is remote from discontinuities such as head-shell intersections, cone-cylinder intersections, nozzles, and supports. Examples are:

a. Shells away from discontinuities due to internal pressure.
b. Compressive and tensile axial stresses due to wind.
c. Axial compression due to weight.
d. Nozzles within the limits of reinforcement due to internal pressure.

Primary local membrane stress, P_L. A primary local membrane stress is produced either by design pressure alone or by other mechanical loads. Primary local membrane stresses have some self-limiting characteristics like secondary stresses. Since they are localized, once the yield strength of the material is reached, the load is redistributed to stiffer portions of the vessel. However, since any deformation associated with yielding would be unacceptable, an allowable stress lower than a secondary stress is assigned. The ability of primary local membrane stresses to redistribute after the material yields allows for a higher allowable stress but only in a local area.

The bending stresses associated with a local loading are almost always classified as secondary stresses. Therefore, the membrane stresses from a WRC-107-type analysis must be broken out separately and combined with general primary stresses due to internal pressure, for example.

Examples of local primary membrane stresses exist:

a. Where internal pressure is the origin of stress and at a discontinuity:
   1. On the shell near a nozzle or other opening
   2. Head-shell juncture
   3. Cone-cylinder juncture
   4. Shell-flange juncture
   5. Head-skirt juncture
   6. Shell-stiffening ring juncture
b. Where non-pressure applied loads are the origin of stress and at a discontinuity:
   1. Support lugs
   2. Nozzle external loads
   3. Beam supports
   4. Major attachments

Primary bending stress, P_b. Primary bending stresses are due to pressure or mechanical loads and are through the thickness. There are relatively few areas where primary bending occurs:

a. Center of a flat head or crown of a dished head.
b. Shallow conical head.
c. In the ligaments of closely spaced openings.

Secondary stresses. These stresses are normal or shear stresses which are required to satisfy an imposed strain pattern. The basic characteristic of a secondary stress is that it is self-limiting. As defined earlier, this means that local yielding and minor distortions can satisfy the conditions which caused the stress to occur. Application of a secondary stress cannot cause structural failure of the vessel due to the restraints offered by the body to which the part is attached. Secondary stresses can develop at structural discontinuities but are also used to describe through thickness gradients away from structural discontinuities. Secondary stresses are also produced by sustained loads other than internal or external pressure.

Structural discontinuities that develop secondary stresses should be placed apart by at least 2.5 $\sqrt{R_m t}$. This restriction is to eliminate the additive effects of edge moments and forces.

Secondary stresses are divided into two additional groups, membrane and bending, though the Code makes no distinction in nomenclature for Q_m or Q_b. Examples of each are as follows:

Secondary membrane stress, Q_m.

a. Axial thermal gradients in shells, cones, or formed heads.
b. Thermal gradients between the shell and head.
c. Thermal stresses due to differential thermal expansion within a nozzle wall.
d. Pressure stress at an isolated ligament.

Secondary bending stress, Q_b.

a. Axial thermal gradients in shells, cones, or formed heads.
b. Thermal gradients between the shell and head.
c. Head-shell juncture
d. Nozzles outside the limits of reinforcement due to pressure and external loading.
e. Thermal stresses due to differential thermal expansion within a nozzle wall.

Peak stress, F. Peak stresses are the additional stresses due to stress intensification in highly localized areas. They apply to both sustained loads and self-limiting loads. There are no significant distortions associated with peak stresses.

Peak stresses are additive to primary and secondary stresses present at the point of the stress concentration. Peak stresses are only significant in fatigue conditions or brittle materials. Peak stresses are sources of fatigue cracks and apply to normal and shear stresses. Examples are:

a. Stress at the corner of a discontinuity (e.g. fillet weld or corner).
b. Thermal stresses due to differential thermal expansion within a nozzle wall.
c. Thermal stresses in cladding or weld overlay.
d. Stress due to notch effect (stress concentration).

Stress Limits

Once the various stresses of a component are calculated, they must be combined and this final result compared to an allowable stress (Table 1-1). Table 1-1 is derived basically from ASME Code, Section VIII, Division 2, and borrowed for application to Division 1 vessels and determining allowable stresses. It should be used as a guideline only because Division 1 recognizes only two categories of stress—primary membrane stress and primary bending stress. Since the calculations of most secondary and peak stresses are not included in this book, these categories can be considered for reference only. In addition, Division 2 utilizes load combinations, by which short-term loads (such as seismic) are reduced when combined with other loads. It also sets allowable limits of combined stresses for fatigue loading where secondary and peak stresses are major considerations.

Table 1-2
Allowable Stresses for Stress Classifications and Categories

Stress Classification or Category	Allowable Stress
General primary membrane, P_m	SE
Primary membrane stress plus primary bending stress across the thickness, $P_m + P_b$	1.5SE
Local primary membrane, P_L	1.5SE
Local primary membrane plus primary bending, $P_L + P_b$	1.5SE
Secondary membrane plus secondary bending, $Q_m + Q_b$	$3SE < 2F_y$
$P + Q$	$3SE < 2F_y$
$P_m + P_b + Q_m + Q_b$	$3SE < 2F_y$
$P_L + P_b + Q_m + Q_b$	$3SE < 2F_y$
Peak, F	S_a
$P + Q + F$	S_a
$P_m + P_b + Q_m + Q_b + F$	S_a
$P_L + P_b + Q_m + Q_b + F$	S_a

Notes:
F_y = minimum specified yield strength at design temperature
E = joint efficiency
S = allowable stress per ASME Code, Section VIII, Division 1, at design temperature
S_a = alternating stress for any given number of cycles from design fatigue curves
The term 3SE shall be used in lieu of $2F_y$ when the ratio of minimum specified yield strength to ultimate strength exceeds 0.7 or S is governed by time-dependent properties.

Thermal Stresses

Whenever the expansion or contraction that would occur normally as a result of heating or cooling an object is prevented, thermal stresses are developed. Thermal stresses are always caused by some form of mechanical restraint.

Thermal stresses are "secondary stresses" because they are self-limiting. That is, yielding or deformation of the part relaxes the stress (except thermal stress ratcheting). Thermal stresses will not cause failure by rupture in ductile materials except by fatigue over repeated applications. They can, however, cause failure due to excessive deformations.

Mechanical restraints are either internal or external. External restraint occurs when an object or component is supported or contained in a manner that restricts thermal movement. An example of external restraint occurs when piping expands into a vessel nozzle creating a radial load on the vessel shell. Internal restraint occurs when the temperature through an object is not uniform. Stresses from a "thermal gradient" are due to internal restraint. Stress is caused by a thermal gradient whenever the temperature distribution or variation within a member creates a differential expansion such that the natural growth of one fiber is influenced by the different growth requirements of adjacent fibers. The result is distortion or warpage.

A transient thermal gradient occurs during heat-up and cool-down cycles where the thermal gradient is changing with time. Thermal gradients may be logarithmic or linear across a vessel wall. Given a steady heat input inside or outside a tube the heat distribution will be logarithmic if there is a temperature difference between the inside and outside of the tube. This effect is significant for thick-walled vessels. A linear temperature distribution may be assumed if the wall is thin. Stress calculations are much simpler for linear distribution.

Thermal stress ratcheting is progressive incremental inelastic deformation or strain that occurs in a component that is subjected to variations of mechanical and thermal stress. Cyclic strain accumulation ultimately can lead to incremental collapse. Thermal stress ratcheting is the result of a sustained load and a cyclically applied temperature distribution.

The fundamental difference between mechanical stresses and thermal stresses lies in the nature of the loading. Thermal stresses as previously stated are a result of restraint or temperature distribution. The stress pattern must only satisfy the requirements for equilibrium of the internal forces. The result being that yielding will relax the thermal stress. If a part is loaded mechanically beyond its yield strength, the part will continue to yield until it breaks. The external load remains constant, thus the internal stresses cannot relax.

The equations and relationships for thermal stresses become increasingly complex when considering thermal gradients, transient thermal gradients, logarithmic gradients, and partial restraint. The basic equations follow. If the temperature of a unit cube is changed from T_1 to T_2 and the growth of the cube is fully restrained:

where T_1 = initial temperature, °F
T_2 = new temperature, °F
α = mean coefficient of thermal expansion in./in./°F
E = modulus of elasticity, psi
υ = Poisson's ratio = .3 for steel
ΔT = mean temperature difference, °F

Case 1: If the bar is restricted only in one direction but free to expand in the other direction, the resulting uniaxial stress, σ, would be;

$\sigma = -E\,\alpha\,(T_2 - T_1)$
If $T_2 > T_1$, σ is compressive (expansion).
If $T_1 > T_2$, σ is tensile (contraction).

Case 2: If restraint is in both directions, x and y, then;

$$\sigma_x = \sigma_y = -\alpha\,E\,\Delta T/(1 - \upsilon)$$

Case 3: If restraint is in all three directions, x, y, and z, then;

$$\sigma_x = \sigma_y = \sigma_z = -\alpha\,E\,\Delta T/(1 - 2\upsilon)$$

A linear thermal gradient through the thickness of a vessel wall, due to temperature difference between the outer and inner wall (thin wall) may be shown as follows;

$$\sigma = +/- E\,\alpha\,\Delta T/(2(1 - \upsilon))$$
$$= .715\,E\,\alpha\,\Delta T \text{ (if } \upsilon = .3)$$

This is a bending stress. If the hotter side is on the inside surface, the hot side is in compression since it wants to expand but is restricted, and if the cold side is on the outside surface it is in tension since it wants to contract but is restricted. Note that this is not a function of vessel diameter or thickness. The stress is due to internal restraint.

A sudden temperature change, ΔT, that penetrates a short distance (but not across the entire shell thickness) is as follows:

$$\sigma = +/- E\,\alpha\,\Delta T/((1 - \upsilon))$$
$$= 1.43\,E\,\alpha\,\Delta T \text{ (if } \upsilon = .3)$$

The average temperature between a nozzle attached to a rigid wall, ΔT, has an upper limit of discontinuity stress of the following:

$$\sigma = 1.83\,E\,\alpha\,\Delta T \text{ (if } \upsilon = .3)$$

Discontinuity Stresses

Vessel sections of different thickness, material, diameter, and change in directions would all have different displacements if allowed to expand freely. However, since they are connected in a continuous structure, they must deflect and rotate together. The stresses in the respective parts at or near the juncture are called discontinuity stresses. Discontinuity stresses are necessary to satisfy compatibility of deformation in the region. They are local in extent but can be of very high magnitude. Discontinuity stresses are self-limiting but some stresses require to be classified as local primary membrane stresses to avoid distortion. That is, once the structure has yielded, the forces causing excessive stresses are reduced. In a typical application they will not lead to failure. Discontinuity stresses do become an important factor in fatigue design where cyclic loading is a consideration. Design of the juncture of the two parts is a major consideration in reducing discontinuity stresses.

It is necessary to superimpose the general membrane stresses with the discontinuity stresses. From superposition of these two states of stress, the total stresses are obtained. Generally when combined, a higher allowable stress is permitted. Due to the complexity of determining discontinuity stresses, solutions will not be covered in detail here. The designer should be aware that for designs of high pressure (>1,500 psi), brittle material or cyclic loading, discontinuity stresses may be a major consideration.

There are two major methods for determining discontinuity stresses:

1. *Displacement Method*—Conditions of equilibrium are expressed in terms of displacement.
2. *Force Method*—Conditions of compatibility of displacements are expressed in terms of forces.

Fatigue Analysis for Cyclic Service

Some vessels are subjected to periodic repetitions of mechanical and thermal loads and the resulting stresses during their service life. When a vessel is subject to repeated loading that could cause failure by the development of progressive fracture, the vessel is considered to be in cyclic service. The ASME Code, Section VIII, Division 1, does not specifically provide details for the design of vessels in cyclic service. However ASME Section VIII, Division 2 has detailed procedures for determining if a vessel in cyclic service requires a detailed fatigue analysis or not, and how to conduct the analysis.

Not every vessel in cyclic service is exposed to the number and magnitude of stress cycles that could shorten its design life. The Code recognizes this and has provided a "screening procedure" to determine whether a vessel is exempt from a rigorous fatigue analysis or not. A detailed fatigue analysis is not mandatory when the limits given by the Code are not exceeded.

Failure depends upon the number of repetitions at a given range of stress rather than the total time under load. When the stress level falls below a certain limit, the number of allowable cycles is said to be indefinite. This value of stress is known as the endurance or fatigue limit, and indicates that a very large number of cycles could occur.

Cracks are more likely to occur in areas of high stress. As such, pressure parts are the most susceptible. Since the longitudinal seams are stressed to twice that of the circumferential joints, they would be most suspect. In addition cracks typically begin on the inside of the vessel.

It is recognized that Code formulas for design of details, such as heads, can result in yielding in localized regions. Thus, localized stresses, exceeding the yield strength of the material, may be encountered even though low allowable stresses have been used in the design. These vessels, while safe for relatively static conditions, could develop "progressive fracture" after a large number of repeated loadings due to high localized and secondary bending stresses.

Fatigue failure can be a result of pressure or temperature fluctuations, as well as other loadings. Fatigue failures have occurred in boiler drums due to temperature variations in the shell at the feed water inlet.

Behavior of metal under fatigue conditions varies significantly from normal stress-strain relationships. Damage accumulates during each cycle of loading and develops localized regions of high stress until subsequent repetitions finally cause cracks to begin and propagate. Progressive fractures develop at stress risers and discontinuities even though the average stress in the vessel may be at design levels.

Factors Influencing Design

Vessels in cyclic service require special consideration for both design and fabrication. Design details play a major role in eliminating or reducing regions of stress risers and concentrations. It is not uncommon to have the design life of a vessel cut in half by poor design details. Although it is not possible to eliminate all stress risers and concentrations, some design details to be avoided are as follows;

1. Use integral construction
2. Avoid fillet welds for attachments to pressure boundary
3. Avoid reinforcing pads
4. Avoid threaded connections
5. Avoid partial penetration welds
6. Avoid stud bolt connections
7. Avoid nozzles in knuckle region of heads

In fatigue service the localized stresses at abrupt changes in thickness, abrupt changes in section, such as at a head junction or nozzle opening, misalignment, defects in construction and thermal gradients are the significant stresses. In general it is always beneficial to minimize peak stresses to the lowest level possible. Peak stresses often become the key stresses involving a fatigue analysis. Peak stresses occur at stress concentrations due to;

1. Fillet welds - high stress at corner of welds
2. Changes in thickness
3. Offset plates
4. Change in geometry
5. Welds attaching clips
6. Welds attaching nozzles

In welded regions, the influencing factors are unknown at the design stages, so they must be compensated for in overall safety factors applied to the procedure. The weld influencing factors are as follows;

1. Local surface notches such as weld bead roughness, weld ripples, undercut, local shrinkage grooves, local root concavity and welding start/stop craters.
2. Variation of the material properties in the various weld zones.
3. Residual stresses
4. Internal defects
5. Weld repairs

In addition Section VIII, Division 2 contains data for smooth bar design fatigue curves and welded joint fatigue curves. The curves represent testing conducted in air, and can be adjusted for the effect of corrosive environments. In general, corrosive environments have a strong, detrimental effect on fatigue behavior. Fatigue cracks in corrosive environments can occur at lower stress ranges; they can occur earlier and propagate faster. Corrosion can cause pitting, non-uniform wall thickness, crevices, gaps, and cracks... all of which reduce fatigue life.

Fabrication tolerances can be equally important details. Normal tolerances for vessels are not adequate. The ASME tolerance for out of roundness of 1% is not always acceptable. Peaking and banding tolerances may need to be much lower than the Code allows. Offsets between plates should be carefully controlled.

Additional factors for carbon steel vessels in cyclic service:

1. Vessels shall be PWHT
2. Material to be normalized
3. Material to be fine grain practice (7 or finer)
4. Plate material shall be 100% UT examined
5. All welds full penetration
6. All main seams ground flush
7. All attachment welds, internal or external, shall be ground smooth or contoured

In actuality, the surface finish, geometry, welded condition, temperature, environmental properties, and non-uniform material properties are drastically different than the test samples. As such, the ASME Code modified the data from the smooth bar fatigue curves by 2 on strain (or stress), or 20 on the number of cycles, whichever was more conservative.

Histograms

A histogram should be developed for each vessel to better assess the quantity of cycles the vessel or component will be subjected to. The purpose of the histogram(s) is to break down the loading history into individual cycles. The loading histogram should be determined based on the specified loadings provided in the UDS (User's Design Specification). The loading histogram should include all significant operating loads and events that are applied to a component. Examples of various histograms are given below.

Cycle Counting Methods

Section VIII, Division 2, Annex 5-B gives procedures for developing a histogram as well as two procedures for counting cycles. There are four major methods used for cycle counting as follows;

1. Rainfall Cycle Counting Procedure
2. Max-Min Cycle Counting Procedure
3. Reservoir Cycle Counting Procedure
4. Simplified Cycle Counting Procedure

Items 1 and 2 are described in Section VIII, Division 2. Procedures 3 and 4 are from other sources. Details for counting are not provided here.

Screening Procedure

Section VIII, Division 1 does not have either a screening method or a procedure for performing a fatigue analysis. However, Section VIII, Division 2 has both. However, it is still acceptable to build and stamp a vessel in cyclic service to Section VIII, Division 1 but to do the fatigue analysis per Section VIII, Division 2. There is no requirement for a vessel to be built to Section VIII, Division 2 simply because it is in cyclic service. On the other hand, Section VIII, Division 1 vessels are not exempt from fatigue analysis because they are not built to Section VIII, Division 2.

There are three Screening Criteria detailed in Section VIII, Division 2 for determining whether a fatigue analysis must be performed or if the vessel is exempt. The three criteria are as follows;

1. Screening Based on Experience with Comparable Equipment
2. Screening Method A
3. Screening Method B

Detailed descriptions of the screening methods are the following;

Screening Based on Experience with Comparable Equipment: Section VIII, Division 2, Paragraph 5.5.2.2 allows a new vessel to be exempted from fatigue analysis if comparable equipment with successful experience and similar loadings is obtained.

The criteria are;

1. Successful experience over a sufficient time frame
2. Similar histogram
3. Addressed in UDS (User's Design Specification)
4. Comparable equipment
5. Similar or same operating conditions

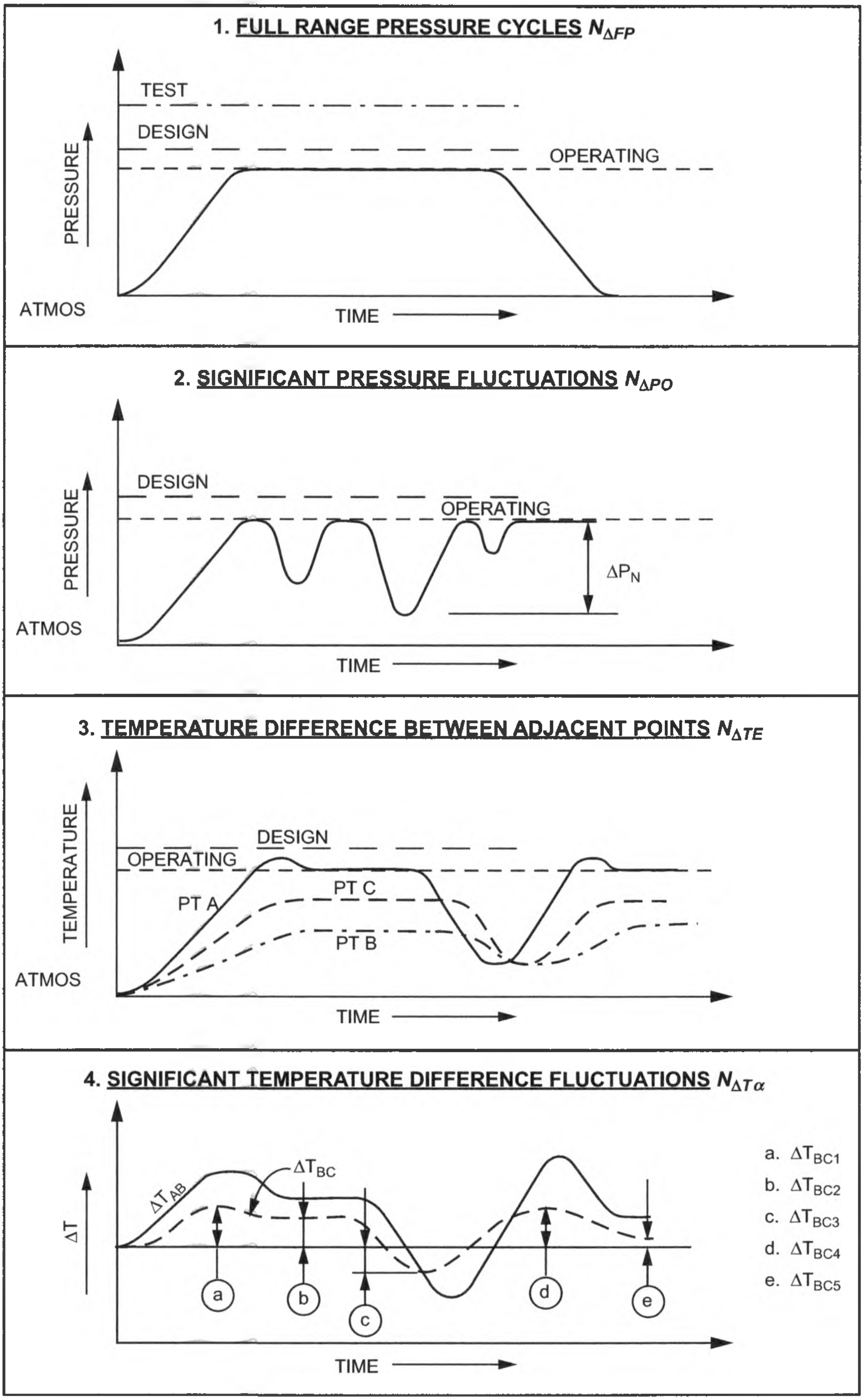
Screening Method A: Section VIII, Division 2, Paragraph 5.5.2.3 can only be used for vessels constructed of materials with an ultimate tensile strength less than or equal to 80,000 PSI (550 Mpa). It is a more simple method of A and B and is more conservative. The steps are as follows;

1. Determine load history... a histogram.
2. Determine the quantity of full range pressure cycles, $N_{\Delta FP}$, including start up and shut down. One full range pressure cycle would encompass starting up and shutting down.
3. Determine quantity of partial pressure fluctuations, $N_{\Delta PO}$. The cycles to be included in this category will vary depending on the type of construction; 20% of the design pressure for integral construction, and 15% for non-integral construction.
4. Determine the quantity of cycles for variation in temperature between adjacent points, $N_{\Delta TE}$. An adjacent point for a shell and dished head is $L = 2.5\,(Rt)^{1/2}$ or 3.5a for flat plates, where 'a' is equal to the radius of a hot spot or a heated area within a plate. This number is modified based on the range of temperatures by the procedure outlined in Section VIII, Division 2.
5. Determine the quantity of cycles for components having different coefficients of expansion, $N_{\Delta T\alpha}$. This does not include cladding.
6. Compare the total number of cycles, $N_T = N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha}$ with the criteria listed in Table 5.9 of ASME Section VIII, Division 2, shown in Table 1-3.

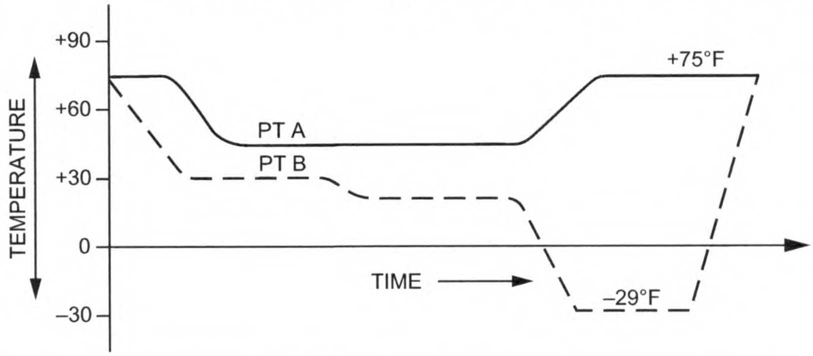
Table 1-3
Fatigue screening criteria for method A

	Description	
Integral Construction	Attachments and nozzles in the knuckle region of formed heads	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq 350$
	All other components	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq 1000$
Non-Integral Construction	Attachments and nozzles in the knuckle region of formed heads	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq 60$
	All other components	$N_{\Delta FP} + N_{\Delta PO} + N_{\Delta TE} + N_{\Delta T\alpha} \leq 400$

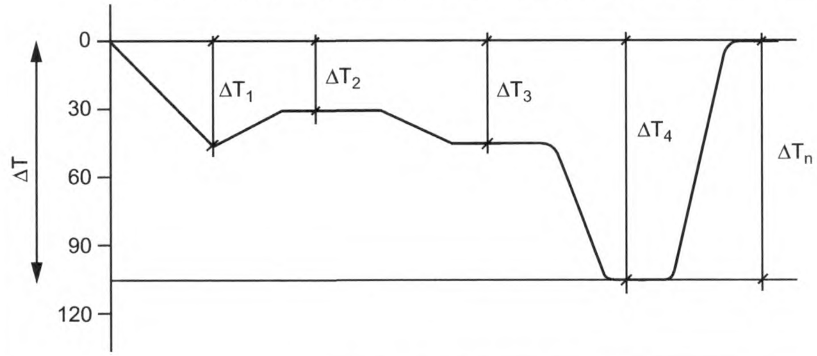
EXAMPLES OF HISTOGRAMS



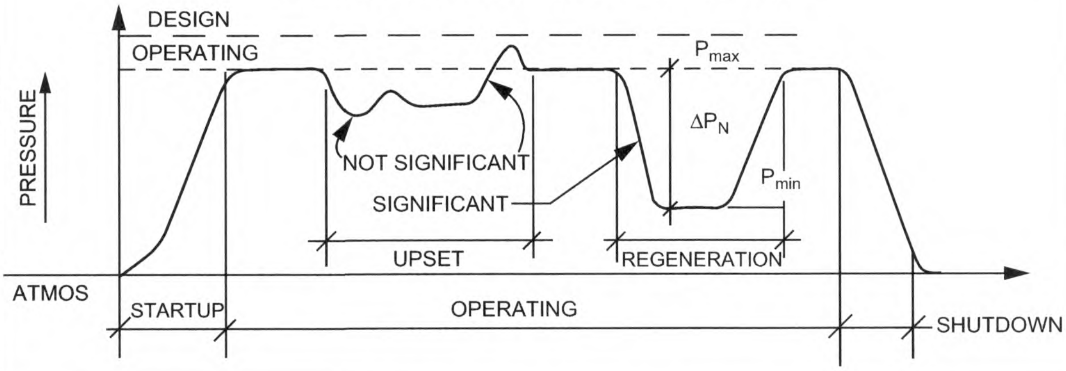
5. TEMPERATURE DIFFERENCE BETWEEN ADJACENT POINTS



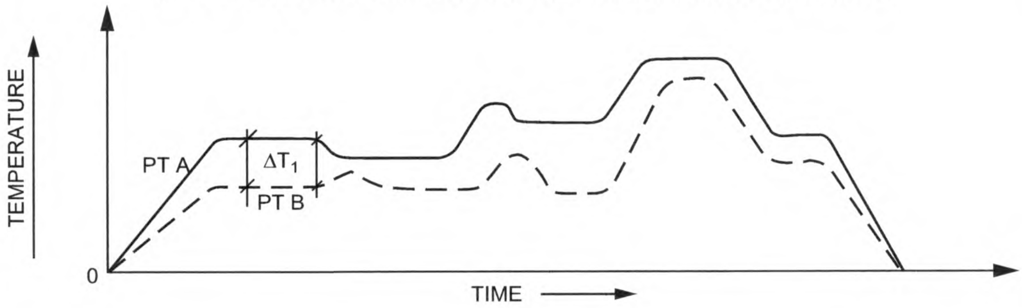
6. SIGNIFICANT TEMPERATURE DIFFERENCE FLUCTUATIONS



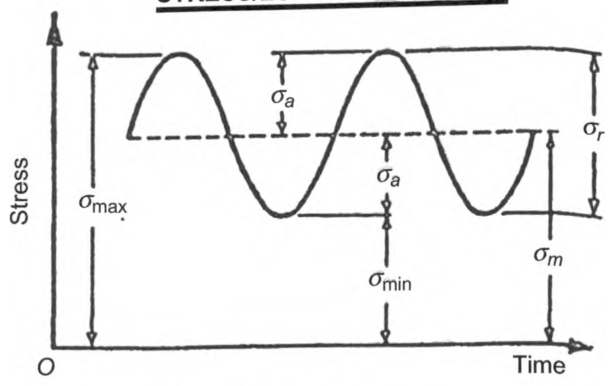
7. SIGNIFICANT PRESSURE FLUCTUATIONS



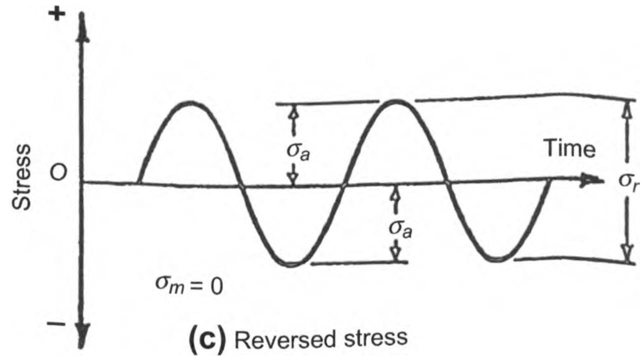
8. TEMPERATURE DIFFERENCE BETWEEN ADJACENT POINTS



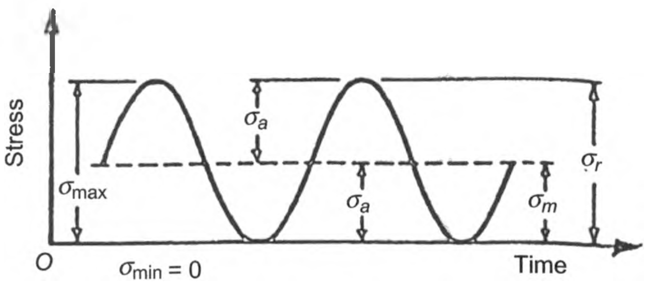
STRESS/LOADING DIAGRAMS



(a) Fluctuating stress



(c) Reversed stress



(b) Repeated Stress or Released Stress

DATA

σ_m = Mean stress; $.5(\sigma_{max} + \sigma_{min})$

σ_r = Stress range; $\sigma_{max} - \sigma_{min}$

σ_a = Stress amplitude; $.5(\sigma_{max} - \sigma_{min}) = .5\,\sigma_r$

σ_{min} = Stress minimum

σ_{max} = Stress maximum

Screening Method B: Section VIII, Division 2, Paragraph 5.5.2.4 indicates that the fatigue screening Method B may be used for all materials. It is not a simple, single procedure as is outlined in method A. Six of the steps require stress calculations, and not satisfying any of the six calculation steps results in a requirement to perform a detailed fatigue analysis.

1. Determine load history… a histogram.
2. Determine screening criteria factors, C_1 and C_2, where for integral construction of attachments and nozzles in the knuckle region of formed heads, $C_1 = 4$ and $C_2 = 2.7$, and for other components $C_1 = 3$ and $C_2 = 2$. For non-integral construction, $C_1 = 5.3$ and $C_2 = 3.6$, and for other components $C_1 = 4$ and $C_2 = 2.7$.
3. Determine the quantity of full range pressure cycles, $N_{\Delta FP}$, including start up and shut down. One full range pressure cycle would encompass starting up and shutting down. N(S) is the number of cycles from the fatigue curve at the stress amplitude.

$$N_{\Delta FP} \leq N(C_1 S)$$

4. Determine the maximum range of pressure fluctuation (excluding startups and shutdowns), ΔP_N, and the corresponding number of cycles, $N_{\Delta P}$. Note that P is the design pressure and S is the allowable stress at design temperature.

$$\Delta P_N \leq \frac{P}{C_1}\left(\frac{S_a(N_{\Delta P})}{S}\right)$$

5. Determine the maximum temperature difference between two adjacent points of the vessel during normal operation, and during startup and shutdown, ΔT_N, and the corresponding number of cycles, $N_{\Delta TN}$. The value α is the coefficient of thermal

expansion, and E_{ym} is the modulus of elasticity at the mean temperature of the cycle.

$$\Delta T_N \leq \left(\frac{S_a(N_{\Delta TN})}{C_2 E_{ym}\alpha}\right)$$

6. Determine the maximum temperature difference between two adjacent points of the vessel during normal operation, and during startup and shutdown, ΔT_R, and the corresponding number of cycles, $N_{\Delta TR}$.

$$\Delta T_R \leq \left(\frac{S_a(N_{\Delta TR})}{C_2 E_{ym}\alpha}\right)$$

7. Determine the range of temperature difference between two adjacent points for components made from different materials during normal operation, ΔT_M, and the corresponding number of cycles, $N_{\Delta TM}$. The values E_{y1} and E_{y2} are the moduli of elasticity for materials one and two, respectively, at the mean temperature of the cycle.

$$\Delta T_M \leq \left(\frac{S_a(N_{\Delta TM})}{C_2(E_{y1}\alpha_1 - E_{y2}\alpha_2)}\right)$$

8. Determine the equivalent stress range from the full mechanical loads, excluding pressure but including piping reactions, ΔS_{ML}, and the corresponding number of cycles, $N_{\Delta S}$.

$$\Delta S_{ML} \leq S_a(N_{\Delta S})$$

Fatigue Assessment: Section VIII, Division 2, Part 5 contains methods for performing an actual fatigue analysis. They are as follows:

1. Elastic Stress Analysis and Equivalent Stresses. This is based off of the pre-2007 Section VIII, Division 2 methodology. Stress ranges will be the output values using this analysis.
2. Elastic-Plastic Stress Analysis and Equivalent Strains. Both stress and strain ranges will be the output using this analysis.
3. Elastic Analysis and Structural Stress (for welds). This method was incorporated into Section VIII, Division 2, to allow for a treatment of welded joints. Stress ranges will be the output values using this analysis.

Ratcheting: Protection against ratcheting shall be performed even if the fatigue screening criteria are met. Ratcheting is progressive incremental inelastic strain that is a result of either mechanical or thermal stress (where thermal stress ratcheting is used to indicate that thermal stresses are mostly responsible for ratcheting action). As ratcheting causes cyclic straining of the material, it can lead to failure by fatigue or collapse. If the loading results in primary plus non-cyclic secondary stresses, ratcheting will be avoided. Shakedown would occur if only initial plastic deformations occurred but upon unloading and reloading only elastic primary and secondary stresses were developed, hence the term, 'shakedown to elastic action'. The methods outlined in Section VIII, Division 2, Part 5 are as follows:

1. Elastic Stress Analysis – Elastic Ratcheting Analysis Method. If the limits of $P_L + P_b + Q$ and including general thermal effects are met by limiting this value to S_{PS}, which is equal to the greater of 3S or $2S_y$, where S is the allowable tensile stress and S_y is the yield strength and where the average value between the specified highest and lowest temperatures are used.
2. Elastic Stress Analysis – Simplified Elastic-Plastic Method. This method may be used in the case where the method indicated above shows the $P_L + P_b + Q$ stress limits are not satisfied, but indicates that the $P_L + P_b + Q$ range and excluding thermal effects must be less than S_{PS}. Additionally, the effective alternating equivalent stress amplitude must include the fatigue penalty factor, $K_{e,k}$, which is based off of the simplified elastic-plastic criteria from the pre-2007 Section VIII, Division 2. Finally, a thermal stress ratcheting assessment must be made.
3. Elastic Stress Analysis – Thermal Stress Ratcheting Assessment. This section will evaluate the allowable limit on the secondary stress range from cyclic thermal loading.

General

The fatigue exemption is performed on a component or part basis since the stress level varies on a component by component basis. One component may be exempt, while another may not be. For any component that is not exempt, a fatigue analysis must be performed for that component.

It should be noted that not all vessels will have a 20 or 30 year life. Some vessels will have a significantly shorter life. The number of allowable cycles for the most highly stressed component will determine the life of the vessel. After the design life of the vessel is reached, the owner or user must then determine whether to retire the vessel, or apply some periodic inspections and NDE to determine if the vessel can continue in service.

If the screening method determines that a fatigue analysis is required, ASME Section VIII, Division 2, Part 5 gives detailed methods for performing this analysis. Basically the fatigue analysis process consists of a rigorous stress analysis of the whole vessel to find the points of highest stress. The highest stress is then used on a fatigue curve for that material to find the allowable number of cycles. If the allowable number of cycles exceeds the actual number of cycles, then that part and vessel are acceptable. If, on the other hand, the actual number of cycles exceeds the allowable number of cycles, then the design must be altered until an acceptable result is achieved, or a shortened design life is acceptable.

Fatigue Curves

Fatigue curves are used to determine the number of allowable cycles. The fatigue curve is also known as the S – N diagram, because one axis represents stress, S, and the other axis represent number of cycles, N. Each material group has their own fatigue curve based on test results and are shown in ASME Section VIII, Division 2, Annex 3-F.

The fatigue curves can be used in several ways as follows;

1. If the number of cycles is known, then you can determine the maximum allowable alternating stress that corresponds to that number of cycles. As long as the actual stress is less than this value, then the design is acceptable.
2. If the actual alternating stress is known, then you can determine the maximum number of allowable cycles based on that stress. If this quantity is greater than the actual number of cycles designed for, then the design is acceptable. As an alternative, the design life can be determined by this method, given the allowable number of cycles.

Permissible Number of Cycles

The permissible number of cycles, N, is based on the alternating equivalent stress amplitude taken from the applicable fatigue curve. Corrections for temperature should be made as follows;

$$N = 10^{X}(E_T/E_{FC})$$

Where;

X = Exponent used to compute the permissible number of cycles
E_T = Modulus of elasticity at temperature evaluated
E_{FC} = Modulus of elasticity used to establish the design fatigue curve.

Effective Alternating Equivalent Stress

The value of alternating stress taken from the fatigue curve is subject to other factors given by the ASME Code. The ultimate allowable stress for a given number of cycles should be adjusted for these factors as follows (for elastic stress analysis and equivalent stresses;

$$S_a = \frac{K_f \cdot K_{e,k} \cdot \Delta S_{P,k}}{2}$$

Where;
K_f = Fatigue strength reduction factor
$K_{e,k}$ = Fatigue penalty factor
$\Delta S_{P,k}$ = Effective equivalent stress range

Cumulative Usage (Damage), U

Often vessels are not subjected to the same range of stress throughout its entire life, but rather to a variety of stress levels for different periods. The results are additive to each other and known as total cumulative damage. S_E in the figure indicates the endurance limit. The check of total cumulative damage can be determined by the following procedure as follows;

Let n_1 = Number of cycles endured at S_1 where N_1 is the fatigue life at that stress

Let n_2 = Number of cycles endured at S_2 where N_2 is the fatigue life at that stress

Let n_3 = Number of cycles endured at S_3 where N_3 is the fatigue life at that stress

Then; $D_{f,k} = [\, n1 / N1 + n2 / N2 + n3 / N3 + \ldots + nk / Nk] =$ Cumulative Damage ≤ 1

Definitions

Adjacent Points: Any two points less than distance $2(R_m t)^{1/2}$ apart.

Cycle: A cycle is a relationship between stress and strain that is established by a specified loading at a specific location. More than one stress-strain cycle can be produced at a given location.

Endurance Limit: The value of stress below which a material can presumably endure an infinite number of cycles, S_E.

Fatigue Life: The number of stress cycles which can be sustained for a given condition, N_A.

Nominal Stress: The stress calculated by simple theory without taking into account variations in stress caused by holes, grooves, fillets and other structural discontinuities.

Peak: The point at which the stress histogram goes from positive to negative.

Stress Cycle: The smallest part of the stress-time function which is repeated periodically and identically. A stress cycle is a condition in which the alternating stress goes from an initial value to a maximum value, then to a minimum value and then returns to the initial value. A single operational cycle may result in one or more stress cycles.

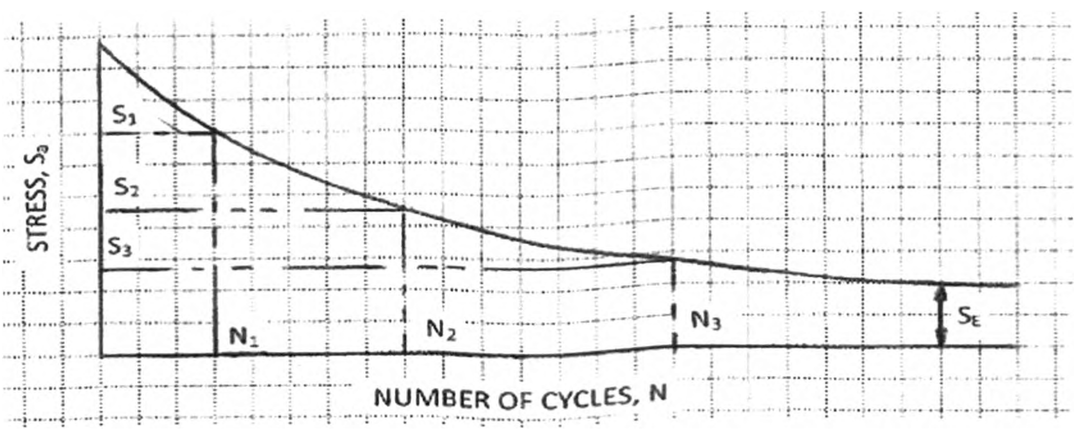
Stress Cycles Endured: The number of cycles endured at any stage of life, N_T.

S-N Diagram: AKA fatigue curve. A plot of alternating stress, S_a, against the maximum number of allowable of cycles, N_A.

Valley: The point at which the stress histogram goes from negative to positive.

Notation

C_1, C_2 = Factors for fatigue screening B
D_f = Cumulative fatigue damage
K_f = Fatigue strength reduction factor
$K_{e,k}$ = Fatigue penalty factor
ΔP_n = Fluctuating pressure
$\Delta S_{P,k}$ = Effective equivalent stress range
ΔS_{ML} = Equivalent stress range from the full range of mechanical load cycles (excluding pressure)
ΔT = Operating temperature range
ΔT_M = Temperature difference between two adjacent points for components made from difference materials
ΔT_N = Temperature difference between two adjacent points for components including startup and shutdown
ΔT_R = Temperature difference between two adjacent points for components including startup and shutdown
$N_{\Delta FP}$ = Number of full range pressure cycles including startup and shutdown
$N_{\Delta P}$ = Number of significant partial pressure cycles
$N_{\Delta PO}$ = Number of significant pressure fluctuation cycles, not including startup or shutdown.
$N_{\Delta S}$ = Number of significant full range of mechanical load cycles (excluding pressure)



$N_{\Delta TE}$ = Number of cycles in metal temperature between adjacent points.
$N_{\Delta TM}$ = Allowable number of cycles for temperature fluctuations for components fabricated from different materials
$N_{\Delta TN}$ = Allowable number of cycles for maximum temperature difference between any two adjacent points
$N_{\Delta TR}$ = Allowable number of cycles for range of temperature fluctuations
$N_{\Delta T\alpha}$ = Number of cycles for components having different coefficients of expansion
S = Maximum allowable stress, tension, ASME VIII-1
S_a = Maximum alternating stress, PSI
S_m = Maximum allowable stress, tension, ASME VIII-2

Creep

Creep is a time dependent phenomena by which the material permanently deforms under stress, and occurs at elevated temperatures. It is a function of the material, stress, temperature, and time. There are several failure modes associated with creep, namely creep rupture, creep fatigue, and creep buckling. Higher stress levels or higher temperatures will result in faster creep rates.

Creep occurs in any metal or alloy at a temperature slightly above the recrystallization temperature. At this temperature the atoms become sufficiently mobile to allow time dependent rearrangement of the metallurgical structure.

Rupture has different mechanisms dependent on the temperature. At room temperature, failure in metals occurs through the grains. This is known as "transcrystalline". At elevated temperatures however, failure occurs around the grains. This is called "intercrystalline". Simply put, at room temperature there is greater strength in the grain boundaries then in the grains themselves. However as temperature is increased, a point is reached where the grains and grain boundaries have equal strength. This temperature is known as the "equicohesive temperature". Below the "equicohesive temperature", initial deformation is elastic. Above the equicohesive temperature, the deformation is plastic.

Creep damage begins with microscopic voids in the material. In time the voids link up into fissures, and finally cracks. The voids, fissures and cracks form at grain boundaries. Creep consists of three distinct stages, primary, secondary and tertiary. Descriptions are as follows;

1. Primary Creep or First Stage: After initial elastic strain, the rate of creep decreases since the effects of strain hardening of the material are greater than the effect of annealing.
2. Secondary Creep or Second Stage: Also known as "steady state creep". In second stage, there is a constant creep rate. The effects of strain hardening and annealing counteract each other.
3. Tertiary Creep or Third Stage: The creep rate rapidly increases. There is a drastically increased strain rate associated with a rapid deformation, terminating in stress rupture. As the cross sectional area of the material reduces (necking) the stress level is increased.

For any given alloy, a coarse grain size has greater strength at elevated temperatures than its fine grain counterpart. Relative slight changes in alloy composition can also alter creep strength appreciably. An increase in alloy content will generally result in better creep resistance. The order of selection of materials is as follows;

1. Carbon Steel
2. C-1/2 Mo
3. 1 Cr
4. 1-1/4 Cr
5. 2-1/4 Cr

Creep begins at different temperatures for different materials and the following may be used as a very general guideline;

1. Aluminum
   200°F to 400°F
2. Titanium
   600°F
3. Carbon Steel
   700°F
4. Low Alloy
   700°F

5. Stainless Steels
   800°F
6. Nickel Alloys
   800°F
7. Refractory metals & alloys
   1800°F to 2800°F

Creep behavior is such that even at constant stress and temperature, strain will develop as shown on the creep curve. According to the Norton-Bailey power law, which models primary and secondary creep, creep strain, ε_c, is more affected by stress than other factors. The creep strain increases no more than linearly with time, but increases exponentially with stress. Creep strain depends on, and is influenced by each of the following four factors:

1. Metallurgy – Alloy and grain size
2. Temperature
3. Time
4. Stress

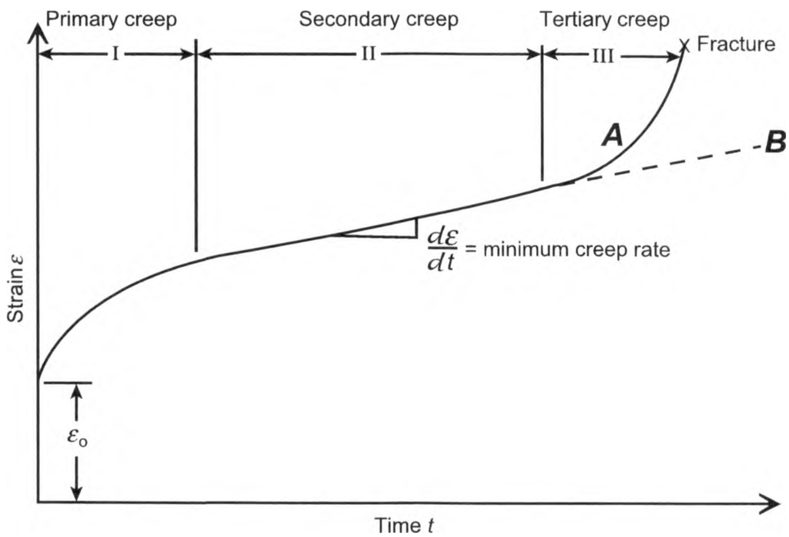


Figure 1-5. Typical creep curve showing the three steps of creep. Curve *A*, constant-load test; curve *B*, constant-stress test.

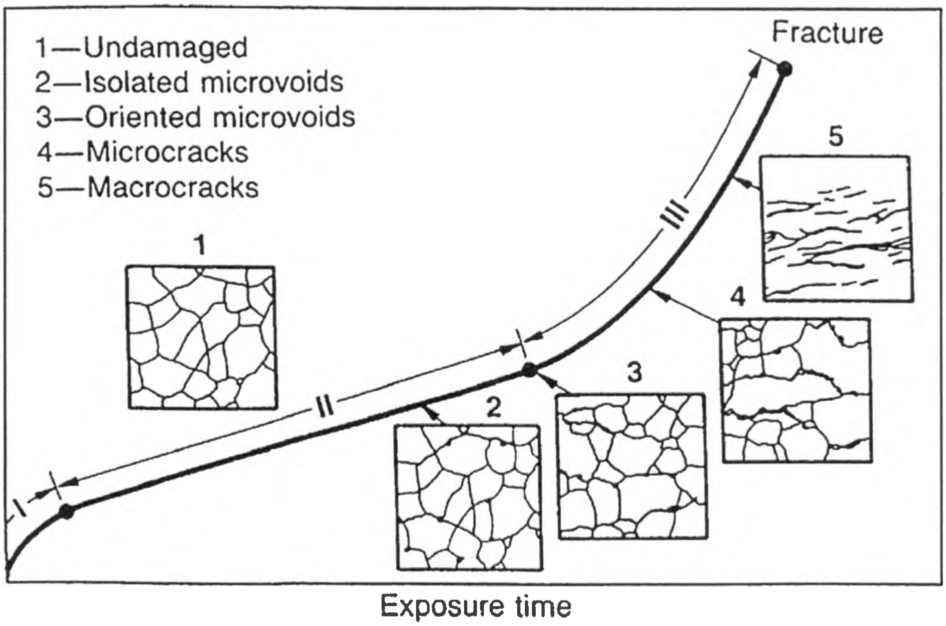


Figure 1-6. Idealized creep curve and corresponding microstructural damage.

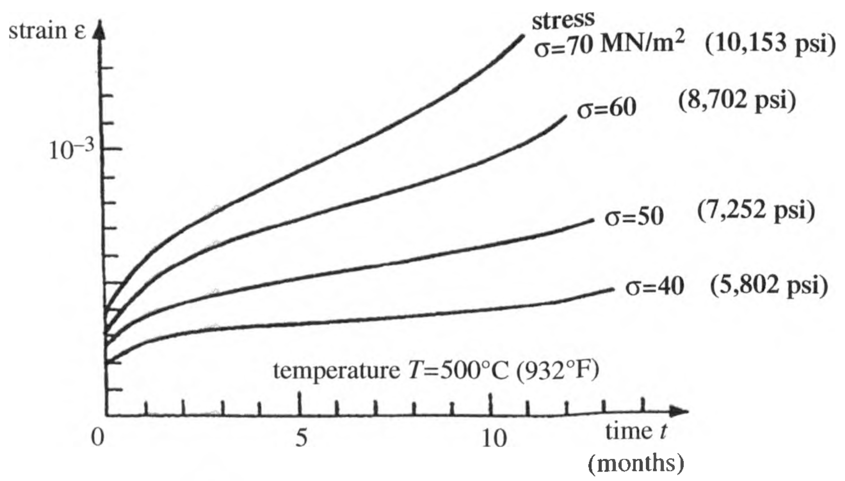
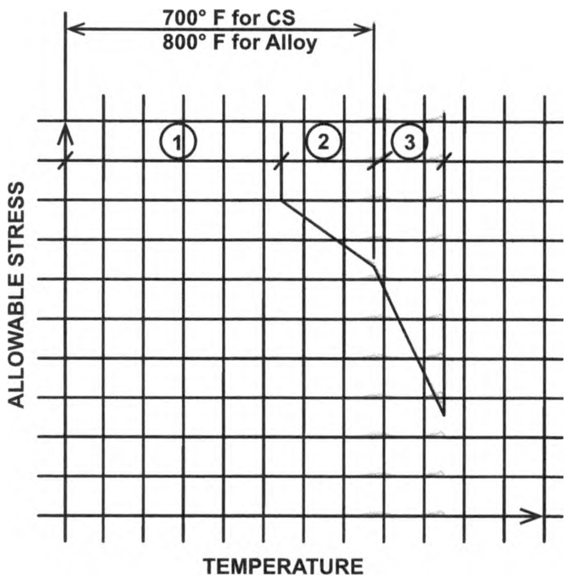


Figure 1-7. Creep curves for carbon steel (Hult, 1966).



① UTS / FS

② $2F_y/3$

③ 1% CREEP IN 100,000 HOURS

Figure 1-8. Idealized graph of allowable stress criteria.

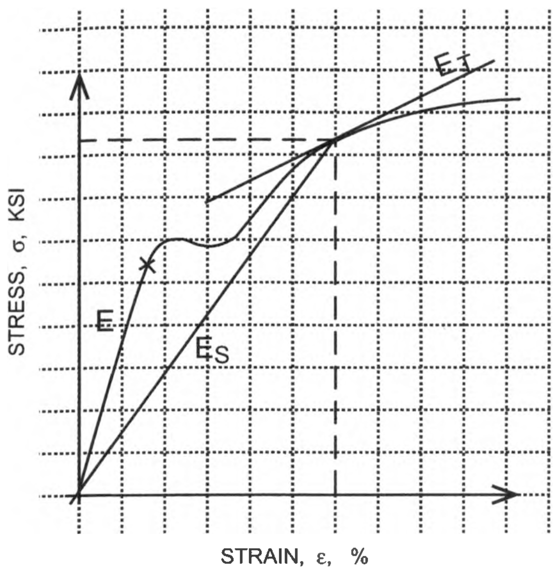


Figure 1-9. Stress-strain curve showing tangent and secant modulus.

There are several design methods utilized in the industry for analyzing creep problems. Three of the most prevalent are the Larson-Miller relation, the Norton-Bailey power law, and the MPC Omega method. The Larson-Miller relation attempts to extrapolate creep rupture data from experimental results. The Norton-Bailey power law is used to determine creep strain and creep strain rate. The MPC Omega method can be used to calculate accumulated and future strain, total damage and damage rate, creep rate, and remaining life. In addition there are several industry standards that provide guidelines, criteria, and design data for materials in the creep range. These are API-530, API-579, and WRC-443.

There are three distinct cases where creep is a factor in the design of equipment. These are as follows:

1. Continuous service in creep range
2. Creep Fatigue: Non-continuous, or intermittent service in the creep range
   a. Low-cycle fatigue
   b. High-cycle fatigue
   c. Thermal fatigue
3. Creep Buckling: Based on geometry of the component operating in the creep range.

Allowable Tensile Stress in the Creep Range

The ASME Code establishes allowable stresses at temperatures where creep and stress rupture govern based on the following:

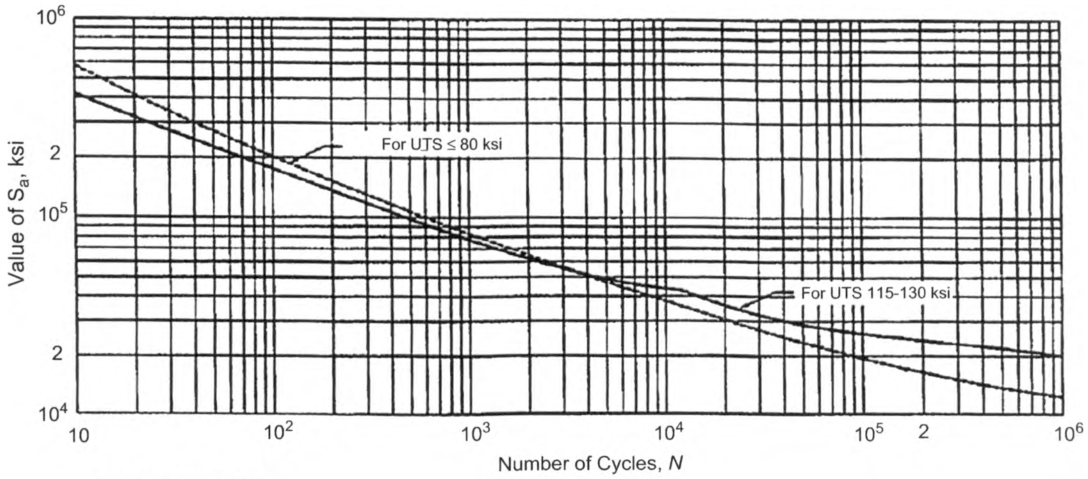
1. 100% of the average stress to yield a creep rate of 0.01%/1,000 hours,
2. 100% of the average stress to produce rupture in 100,000 hours (11.41 years),
3. 80% of the minimum stress to cause rupture in 100,000 hours.

In the ASME Code, Section II, Part D, Table 2A, the allowable tensile stress values for the materials that are governed where creep and stress rupture govern are indicated by *italics*. However this is not the sole criteria for determining if creep should be a consideration. Creep temperature limits are also given inadvertently in the fatigue curves of ASME Section VIII, Division 2. For the materials shown in Appendix 5, Fig. 5-110.1, the limit is 700°F. For materials listed in Fig. 5-110.4, the limit is 800°F. ASME Section III, Subsection NH, Appendix T also shows similar temperatures for the onset of creep.

Generally, misunderstanding arises when the design temperature is greater than these temperature limits, 700°F for carbon and low alloy materials and 800°F for alloy materials. The confusion arises because some of the materials above these temperature limits are not shown in *italics*. Many interpret this to mean that creep is not a consideration. However the magnitude of creep strain at any given temperature depends not just upon the stress level, but repeated applications and time at elevated temperature. The Code acknowledges this by limiting the temperatures when using the fatigue charts/tables.

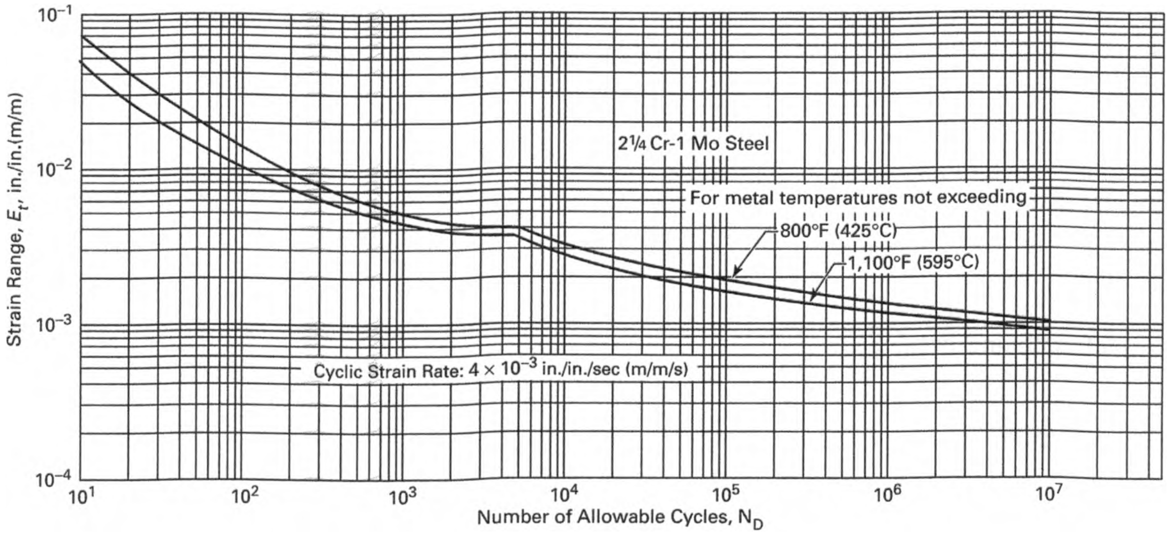
Allowable Compressive Stress in the Creep Range

The ASME Code gives temperature limitations for most groups of materials under external pressure and axial compression for cylinders, and for external pressure for spheres. The limitations are shown in graphical form in Section II, Part D, Mandatory Appendix 3, Figures



GENERAL NOTES:
(a) $E = 30 \times 10^6$ psi.
(b) Interpolate for UTS 80.0-115.0 ksi.
(c) Table 5-110.1 contains tabulated values and a formula for an accurate interpolation of these curves.

Figure 1-10. Design fatigue curves for carbon, low alloy, series 4XX, high alloy steels and high tensile steels for temperatures not exceeding 700°F and $N \leq 10^6$ (Use Fig. 5-110.1.1 for $N \geq 10^6$). (*Reprinted by permission, ASME*).



N_d, Number of Cycles [Note (1)]	ϵ_t, Strain Range (in./in.) (m/m) at Temperature	
	800°F (425°C)	900-1,000°F (480-595°C)
10^1	0.056	0.040
4×10^1	0.023	0.0163
10^2	0.013	0.0097
2×10^2	0.0094	0.0070
4×10^2	0.0070	0.0056
10^3	0.0052	0.0042
2×10^3	0.0044	0.0039
4×10^3	0.0040	0.0035
10^4	0.0032	0.00265
2×10^4	0.0026	0.00215
4×10^4	0.0023	0.00182
10^5	0.00195	0.00158
2×10^5	0.00173	0.00142
4×10^5	0.00155	0.00130
10^6	0.00137	0.00118

NOTE:
(1) Cycle strain rate: 4×10^{-3} in./in./sec (m/m/s).

Figure 1-11. Design Fatigue Strain Range, ε_t, for 2 ¼ Cr-1Mo Steel. (*Reprinted by permission, ASME.*)

3-500.1, 3-500.2 and 3-500.3. These temperatures are also a function of radius to thickness ratio. As the vacuum charts are based on short-term tensile tests, they are limited to certain temperatures since creep reduces the critical buckling stress. Since "time at temperature" is a critical factor for elevated temperature designs, it may be possible to design for short term loadings using this method.

For Code Case 2286 (now adopted into Section VIII, Division 2), the temperature limits are indicated within the said document.

At elevated temperatures, the effect of creep is to reduce the critical buckling stress. The critical buckling stress is dependent on the magnitude of the load and the time at load. Either load or time at elevated temperature will have a detrimental effect.

Additional information and design limits for elevated temperature buckling and instability are given in ASME III, Division 1, Subsection NH, Appendix T, Para T-1500.

Creep-Fatigue

For vessels in cyclic service, below the creep range, the fatigue-stress curves of Section VIII, Division 2 may be used to determine the number of acceptable cycles.

Above the creep range, a fatigue-strain curve is used. Within Section VIII, Division 2, a fatigue screening analysis is required. If the component fails the screening, a fatigue evaluation is required by the ASME Code. For vessels in the creep range, data for a fatigue evaluation is contained in Section III, Division 1, Subsection NH, Appendix T.

The methodology in Section III uses an interaction diagram, whereby a total fraction of creep damage is compared to a total fraction of fatigue damage using a material specific graph. For the creep damage, the fraction is calculated by determining the actual time at some temperature and dividing by an allowable time at some temperature and stress condition. For the fatigue damage, the fraction is calculated by determining the actual number of cycles at some at some temperature by an allowable number of cycles at that temperature. The summation of both total fractions must be less than or equal to a value as determined graphically from the interaction graph and is a function of material.

Often times, vessels are subjected to short term loadings that overstress some portion of the vessel for a very short duration. This short duration, overstressed condition, may in fact be acceptable providing the number of cycles at that overstressed condition is acceptable. A fatigue evaluation can validate this situation. The Code allowable stresses are based on continuous service up to 100,000 hours. If the vessel or component is going to be exposed to the overstressed condition for less than 100,000 hours, then the acceptability hinges on the number of allowable cycles rather than allowable stress.

The "design life" based on the allowable number of cycles must be monitored by the end user. At the end of life, the vessel must be retired, changed service or a systematic NDE program established for the continued safe operation of the equipment.

There is no general empirical or mathematical equation to relate creep strain to metallurgy or temperature since it is a combination of all these factors. One common equation for creep strain is the Norton-Bailey power law;

$$\varepsilon_c = A\,\sigma^m\,t^n$$

where,

ε_c = Creep strain, in/in
A = Material constant calculated from isochronus curves
σ = Applied stress, psi
t = Time, hours
m = Material constant calculated from isochronus curves. Always greater than 1
n = Time hardening coefficient, typically between 1/3 and 1/2.

Isochronus curves: These curves are available for many materials from ASME Section III, Division 1, Subsection NH, Class 1 for components in elevated temperature service. The word "isochronus" is derived from the words "iso", meaning equal, and "chronous" meaning time. It is a stress-strain curve for a given material, temperature, and time duration.

Creep-Buckling

One of the failure modes for cylindrical or conical under axial compression is buckling. In cylindrical components, buckling is a phenomenon that occurs when the cylinder fails in compression before the ultimate compressive strength is reached. It is a function of geometry, material properties, and is affected by imperfections in shape.

There are two kinds of failure due to buckling. The first is "general buckling" and involves bending of the axis of the cylinder resulting in instability. This is the type addressed by Euler and designed for by a "slenderness ratio" method.

The other type of buckling is a result of local instability that may or may not result in a change in the axis of the cylinder. This type is known as "local buckling" and the stability against local buckling is dependent on D/t ratios.

Creep-buckling is an issue when the cylindrical component operates in the creep range. Again, creep is a function of stress, temperature, and time.

Since the compressive loads in Section II, Part D do not account for time dependent loads, the Welding Research Council (WRC) published Bulletin 443 for calculating design limits for elevated temperature buckling based on theory as well as factors from Section III, Division 1- Subsection NH. Bulletin 443 presents equations for cylinders under axial compression as well as external pressure, and spheres under external pressure.

Creep-buckling is an issue when the cylindrical component operates in the creep range. The protection against buckling is determined by calculating a critical stress or failure point and applying a safety factor. This safety factor is a variable in the design of such components.

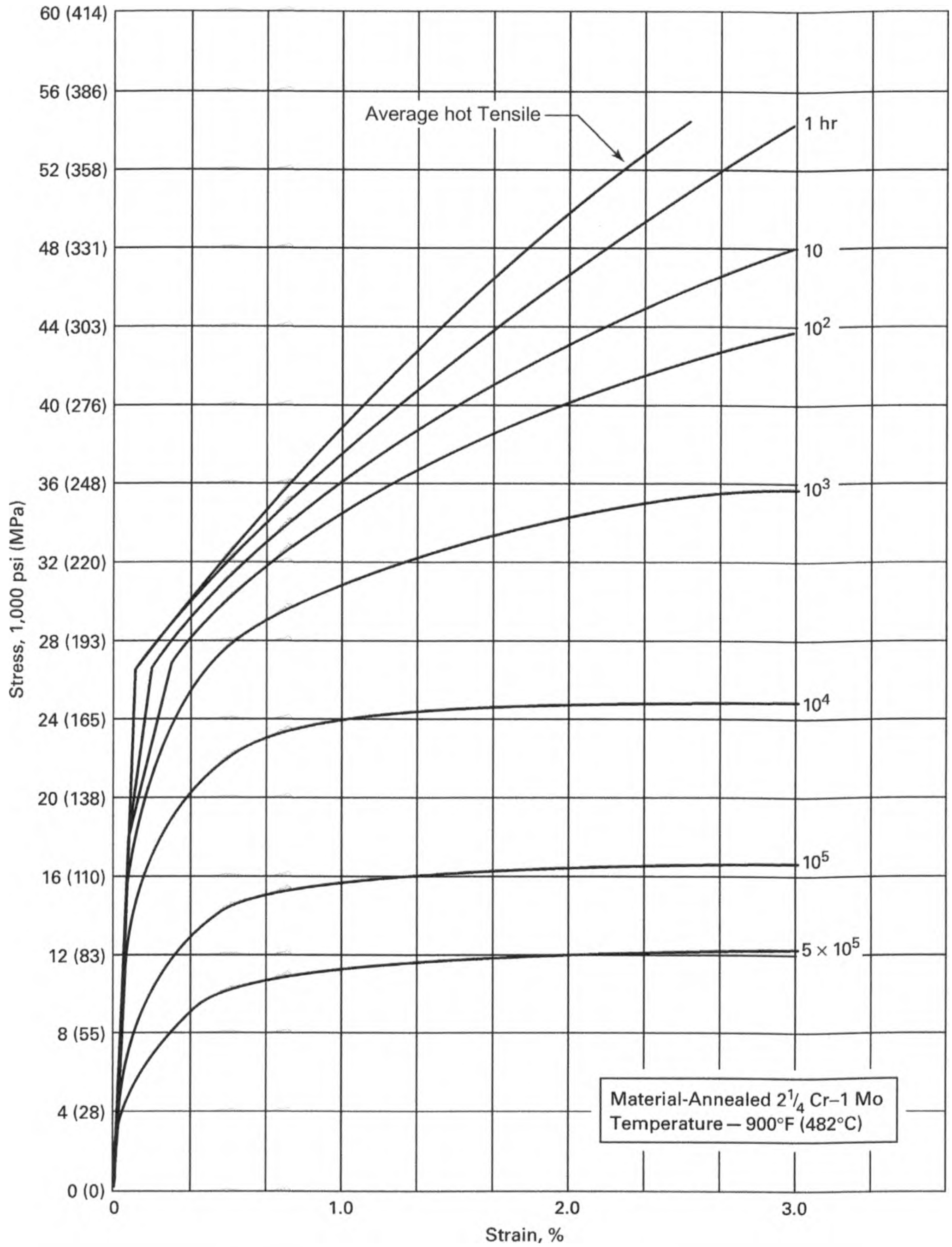


Figure 1-12. Average Isochronous Stress-Strain Curves. (*Reprinted by permission, ASME.*)

ASME Section II Part D, Mandatory Appendix 3 requires one to perform a creep-buckling analysis to obtain the allowable compressive stress at elevated temperature. Welding Research Council (WRC) Bulletin 443 has the following formula to calculate the critical buckling stress at elevated temperatures for a cylinder under axial compression:

$$\sigma_c = \left[\frac{E_t E_s}{3(1-\mu^2)}\right]^{1/2} \times \frac{t}{r_o}$$

Data

σ_c = critical buckling stress, axial, psi
t = wall thickness, in.
r_o = outside radius, in
ν = Poisson's ratio (0.3 for steel)
μ = $1/2-(1/2-\nu)E_s/E$
E = modulus of elasticity, psi
B = ASME Code B factor, psi
ε = strain, in/in
ε_{cr} = critical strain, in/in
α = factor for imperfection of cylinder

E_t, tangent modulus: When the critical stress exceeds the proportional limit of the material, the modulus of elasticity at that particular point on the curve is no longer valid. Instead the modulus of elasticity decreases to the local tangent value, E_t. The tangent modulus is used for buckling stresses above the proportional limit. Beyond this transition point, yield or creep will govern. The tangent modulus may be found from the isochronus stress-strain curve for that time or by using the MPC Project Omega data. The modulus is the slope of the line, tangent to the curve at the point of % strain.

E_s, secant modulus: The secant modulus is the slope of the line drawn from the origin to the point of % strain. The secant modulus may be found from the isochronus stress-strain curve for that time or by using the MPC Project Omega data.

E_e, equivalent modulus.

Per Mandatory Appendix 3-500, paragraph (c)(3), the tangent modulus, E_t, shall be used for σ_c above the proportional limit (transition point). The above formula can be used for both above and below the proportional limit.

Up to the proportional limit the σ_c formula reduces to:

$$B = \frac{AE}{2}$$

where $A = 0.125\ t/r_o$

In the σ_c formula, let $E_tE_s = E_e^2$ where E_e is the equivalent modulus.

$$\sigma_c = \left[\frac{E_e^2}{3(1-\nu^2)}\right]^{1/2} \times \frac{t}{r_o}$$

Up to the proportional limit, $E_e = E$, $\mu = \nu = 0.3$ so that:

$$\sigma_c = \left[\frac{E^2}{3(1-0.3^2)}\right]^{1/2} \times \frac{t}{r_o}$$

$$\sigma_c = \frac{E}{[3(1-0.3^2)]^{1/2}} \times \frac{t}{r_o}$$

$$\sigma_c = (0.605)(E)\left(\frac{t}{r_o}\right) \quad \text{(Timoshenko)}$$

The Code formula for a cylinder under axial compression is based on an imperfection factor, $\alpha = 0.207$ and a design factor, D.F. = 2. The factor, α, accounts for theoretical (for a perfect shape) versus test results (imperfect shape):

$$B = (\alpha)\left(\frac{\sigma_c}{2}\right) = \frac{(0.207)(0.605)(E)\left(\frac{t}{r_o}\right)}{2}$$

$$B = (0.125)\left(\frac{t}{r_o}\right)\left(\frac{E}{2}\right)$$

$$B = \frac{AE}{2} \quad \text{(Code formula)}$$

Therefore, the σ_c formula is the continuation of the Code formula value up to the proportional limit. The steps for calculating σ_c are as follows.

1. Calculate the proportional limit, σ_{prop}, by the isochronous curve.
2. Calculate the critical stress, σ_c, for $E = E_t = E_s$ using E from Section II, Part D. If the calculated value of σ_c is less or equal to σ_{prop} then the value of this is the value of σ_c.
3. If the calculated value of σ_c is greater than σ_{prop} then calculate the values of E_t and E_s to determine E_e and μ.
4. Use E_e and μ to calculate the value of σ_c and the corresponding value of ε_{cr}.
5. Calculate the allowable axial compressive stress using σ_c, a modified imperfection factor, α, and the appropriate design factor.

The following procedure should be used for determining the critical buckling stress for items operating in the creep range. ASME Code Vacuum charts stop at

800°F for carbon and low alloy steel. Above this temperature an alternative method must be used to determine the critical buckling stress.

This method shown below is referenced in ASME Code, Section II, Part D, Para 3-500(a) (3) which in turn refers to WRC Bulletin 443. The procedure is as follows;

Data

σ_c = critical buckling stress, psi
η = calculated per equation
E_s = secant modulus, psi
E_t = tangent modulus, psi
μ = calculated per equation
ν = Poisson's ratio, 0.3 for steel
r = radius of vessel, in.
t = thickness, in.

Sample Problem # 1, Creep Buckling (WRC – 443 Method)

Cone dimensions and properties:

Material = SA-240-304H
Design temp. = 1,150°F
Cone diameter at large end = 6.6284 in.
Cone diameter at small end = 2.6284 in.
Corroded thickness = 0.1036 in.
Length of cone = 12 in.
Equivalent length of cone = 12.166 in.
Half apex angle = 9.46°
Equivalent diameter of cone = 4.966 in.
Equivalent radius of cone = 2.483 in
Poisson's ratio = 0.3
Modulus of elasticity, E = 21,600 ksi
Tangent modulus, E_t = 8,431 ksi
Secant modulus, E_s = 17,700 ksi

$$\mu = 1/2 - (1/2 - \nu)E_s/E$$

$$\mu = 0.3361$$

$$\eta = 1 - \frac{(1 - E_t/E_s)}{1 + (1 - 4\mu^2)E_t/3E_s}$$

$$\eta = 0.5183$$

$$\sigma_c = \frac{\eta E_s t^2}{4(1 - \mu^2)r^2}$$

$$\sigma_c = 4,501 \cdot ksi$$

Cryogenic Applications

The word "cryogenics" comes from two Greek words, "kryos" meaning icy cold, and "genes" meaning to form. The term cryogenics was first used in 1875 and has come into general usage since 1955. Cryogenic temperatures are normally considered as below (–)150°F.

This fundamental characteristic of cryogenic technology has found its way into almost every major industry. Cryogenic applications are found in the steel, space, refining, welding, chemical, glass, cement, food, electronics and medical industries. In terms of volume of products, the steel and chemical industries represent the largest consumers of these products.

The principle gases include propane, ammonia, CO_2, argon, oxygen, helium, hydrogen, nitrogen, ethane, ethylene, methane and chlorine. Oxygen, nitrogen, argon and helium are mainly produced by the process of liquefaction.

There is a variety of equipment associated with separation and liquefaction of gases. These include heat exchangers, cold box equipment, distillation columns, storage vessels and compressors. In addition the storage, transportation, distribution and ultimate consumption of cryogenic fluids also require an array of mechanical equipment, piping, valves and instrumentation.

Due to the brittle nature of carbon steel at low temperature, this material is not suitable for cryogenic applications. Carbon steels can be utilized in low temperature service down to only about (–) 50°F, and then with the right testing and precautions. This is well above what would be considered a cryogenic application.

Most nonferrous metals are suitable for low temperature service. Essentially, all copper, aluminum, and high nickel based alloys remain tough and ductile in the cryogenic range. Low temperature applications also utilize low nickel alloys, such as 2-1/4 Ni and 3-1/2 Ni. Cryogenic applications utilize 9% Ni, stainless steel and aluminum. Austenitic stainless steels are capable of exposure to temperatures to absolute zero, (–) 459°F.

One of the major benefits of handling gases in liquefied form is the enormous reduction in volume resulting from the liquefaction of gas. Savings are achieved because one cubic foot of liquefied gas is equivalent to many hundreds of cubic feet of gas volume at normal pressure and temperature. Thus the handling of cryogenic fluids requires less container space.

Typical containers for cryogenic liquids are pressure vessels, spheres or tanks. In any case the normal storage device consists of a double wall, much like a thermos. Between the two walls is an insulation barrier to keep the cold within the inner vessel. Where the service requirements become critical in terms of minimizing heat loss, evacuation of the inner space becomes economically justifiable. The maintenance of this vacuum becomes one of the long term service problems. Vacuum requirements frequently begin for temperatures below (−) 200°F.

In double wall vessels and tanks the inner wall is subjected to the low temperature and the material required must correspond to that temperature. The outer vessel shell is used to support the insulation, contain the vacuum, if required, and support the total weight of the tank, contents and insulation. The outer vessel may be ordinary carbon steel, since it is a structural part only. The outer vessel does not need to be ASME Code stamped unless requested.

The inner vessel can be supported off the outer vessel in a number of ways but must minimize heat loss paths. The typical way to support the inner vessel is through a series of sway rods and support rods. These rods as well as the nozzles that penetrate both inner and outer shells, cause the most amount of heat loss. This heat loss results in boil-off of the cryogenic liquid and subsequent product loss.

If not properly designed, heat losses through the attachments to the inner shell, will account for more heat loss than the rest of the surfaces. Proper design will limit heat loss through these components to approximately 20% of the total calculated heat loss.

Many of the developments in the cryogenic field have only been possible because of the development of high efficiency insulation. Unlike most of our processes which are insulated to keep the heat in, cryogenic insulation is designed to keep the heat out.

Insulation becomes less important as the size of the vessel increases. The most important factor for heat loss is the surface area to volume ratio. As the size of the vessel increases, this ratio decreases and heat loss is less.

There are four general types of insulation used for cryogenic applications;

1. High Vacuum Insulation
2. Multiple Layer (super-insulation with vacuum)
3. Powder (perlite or silica aerogel used with no vacuum, partial vacuum, or full vacuum)
4. Rigid Foam (does not require a vacuum)

High Vacuum insulation is used for applications below (−) 200°F in double wall cryogenic vessels, as opposed to insulation between the inner and outer vessel walls. Surface reflectivity and emissivity are important factors for these applications and are a function of surface treatment, finish and cleanliness. High vacuum systems are designed to 10^{-9} torr and less. The better the surface finish and the higher emissivity, the better the insulating properties.

Specifications

A partial list of relevant Codes, Standards and Regulations is as follows;

US Std's & Codes:
Note CGA stands for "Compressed Gas Association"
CGA P-8.3
CGA P-12
CGA P-25
ASTM C740-97
European Std's & Codes:
EN 1252 CV
EN 12213
EN 12300
EN 13458 CV
EN 14197 CV
IGC DOC6/02/E
IGC DOC21/85/E/F/D
ISO/WD 21009-1
BS 5429:1976
BS 7777-4:1993

Service Considerations

The following is a list of services that require special consideration with regard to vessel design and/or material selection;

1. Cyclic Service: Fatigue
2. Lethal Service
3. Vibration Service
4. Shock Service; Thermal or Impulse loading
5. Low Temperature Service
6. Cryogenic
7. High Temperature Service
8. Creep
9. Creep –Fatigue
10. Corrosion Service
11. Contents: The contents of the vessel may require special design considerations and/or material selection. The following is a list of some of these special services;
    a. Hydrogen Service
    b. Wet H_2S Service (Sour Water)
    c. Caustic Service
    d. Ammonia Service
    e. Chloride Service
    f. Amine Service
    g. Sulfuric Acid
    h. Hydrochloric Acid
    i. Hydrofluoric Acid
    j. Polythionic Acid
12. Types of Hydrogen Service Failures;
    a. SSC: Sulfide Stress Cracking
    b. ASSC: Alkaline Sulfide Stress Cracking
    c. SZC: Soft Zone Cracking
    d. HIC: Hydrogen Induced Cracking
    e. SOHIC: Stress Oriented HIC
    f. SCC: Stress Corrosion Cracking
    g. GHSC: Galvanic Hydrogen Stress Cracking
    h. EC: Environmental Cracking
13. Types of Stress Corrosion Cracking (SCC);
    a. Chloride
    b. Polythionic
    c. Caustic
    d. MEA/DEA
    e. Sulfide
    f. Ammonia

Miscellaneous Design Considerations

TABLE 1-4
Guidelines for establishing design pressure & temperatures

Design Pressure		
Item	**Maximum Operating Pressure (PSIG)**	**Design Pressure (PSIG)**
1	Vacuum (Full or Partial)	50 PSIG Internal + Full Vacuum
2	0 to 5	50 PSIG Internal + Full Vacuum
3	6 to 35	50
4	36 to 100	Operating + 15
5	101 to 250	Operating + 25
6	251 to 500	Operating + 10 %
7	501 to 1,000	Operating + 50
8	over 1,000	Operating + 5%

Design Temperature		
Item	**Maximum Operating Temperature (°F)**	**Design Temperature (°F)**
1	Below (−) 10	Operating (−) 25
2	(−) 9 to 14	Operating (−) 20
3	15 and up	Operating + 50

TABLE 1-5
General size categories of vessels

Type	Small		Medium		Large	
HORIZONTAL / VERTICAL	DIA	<10 Ft	DIA	10 to 15 Ft	DIA	>15 Ft
	WEIGHT	<25 Tons	WEIGHT	25 to 50 Tons	WEIGHT	>50 tons
TRAYED COLUMNS	DIA	<10 Ft	DIA	10 to 15 Ft	DIA	15 to 25 Ft
	WEIGHT	<30 Tons	WEIGHT	30 to 100 Tons	WEIGHT	>100 Tons
	LENGTH	<50 Ft	LENGTH	50 to 100 Ft	LENGTH	>100 Ft
REACTORS & HIGH PRESSURE VESSELS	DIA	6 to 9 Ft	DIA	10 to 14 Ft	DIA	>15 Ft
	WEIGHT	<200 Tons	WEIGHT	200 to 500 Tons	WEIGHT	>500 Tons
	THK	2" to 4"	THK	4" to 6"	THK	>6"

Items to be Included in a User's Design Specification (UDS) for ASME VIII-2 Vessels

The following items should be included as part of the UDS.

1. Title to include Vessel name, Description and Item number
2. Project Name and Description
3. Scope
4. User
5. List of all Design Codes and Standards
6. List of all Specifications
7. List of all Design Drawings
8. List of all Standard Drawings

9. Design Basis
10. Design Data
11. Description of Loads, Load Cases, and Load Combinations (e.g. Wind, Seismic, Snow, Erection, etc.)
12. Location & Site Data
13. Fatigue Evaluation
14. Creep Evaluation
15. Design Life
16. Operating Data
17. Material Specifications
18. Inspections
19. Pressure Relief Device and Location
20. Special Service (Sour water, Hydrogen, Lethal, Cyclic, etc.)
21. Heat Treatment
22. Pressure Tests
23. Conflicts
24. Nozzle Loads
25. Design Certification

References

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[6] Hicks EJ, editor. Pressure Vessels—A Workbook for Engineers, Pressure Vessel Workshop, Energy—Sources Technology Conference and Exhibition. Houston: American Society of Petroleum Engineers; January 19–21, 1981.

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[11] Criteria of the ASME Boiler and Pressure Vessel Code for Design by Analysis in Sections III and VIII, Division 2, American Society of Mechanical Engineers.