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CRITERIA OF THE ASME BOILER AND PRESSURE VESSEL CODE FOR DESIGN BY ANALYSIS IN SECTIONS III AND VIII, DIVISION 2

THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS
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DESIGN

I. INTRODUCTION

The design philosophy of the present Section I (Power Boilers) and Division 1 of Section VIII (Pressure Vessels) of the ASME Boiler Code may be inferred from a footnote which appears in Division 1 of Section VIII on page 9 of the 1968 edition. This footnote refers to a sentence Par. UG-23 (c) which states, in effect, that the wall thickness of a vessel shall be such that the maximum hoop stress does not exceed the allowable stress. The footnote says:

> "It is recognized that high localized and secondary bending stresses may exist in vessels designed and fabricated in accordance with these rules. Insofar as practical, design rules for details have been written to hold such stresses at a safe level consistent with experience."

What this means is that Section I and Division 1 of Section VIII do not call for a detailed stress analysis but merely set the wall thickness necessary to keep the basic hoop stress below the tabulated allowable stress. They do not require a detailed evaluation of the higher, more localized stresses which are known to exist, but instead allow for these by the safety factor and a set of design rules. An example of such a rule is the minimum allowable knuckle radius for a torispherical head. Thermal stresses are given even less consideration. The only reference to them is Par. UG-22 where "the effect of temperature gradients" is listed among the loadings to be considered. There is no indication of how this consideration is to be given. In the othe hand, the Piping Code (USAS-B31.1) does give allowable values for the thermal stresses which are produced by the expansion of piping systems and even varies these allowable stresses with the number of cycles expected in the system.
lowable stresses with the number of cycles expected in the system.

The Special Committee to Review Code Stress Basis was originally established to investigate what changes in Code design philosophy might permit use of higher allowable stresses without reduction in safety. It soon became clear that one approach would be to make better use of modern methods of stress analysis. Detailed evaluation of actual stresses would permit substituting knowledge of localized stresses, and assignment of more rational margins, in place of a larger factor which really reflected lack of knowledge.

The ASME Special Committee dealt with these problems partly by the knowledge and experience of individual members and partly by the results of numerous analytical and experimental investigations. The Code Committee itself does not conduct research programs, but is able to derive much useful information from the Pressure Vessel Research Committee. PVRC is a private non-profit organization supported by subscription of interested fabricator and user groups and established to sponsor cooperative research programs aimed at improving the design, fabrication, and materials used in pressure vessels. Among other programs PVRC has sponsored considerable work on fatigue behavior in materials and vessels. Results of these experimental programs were studied by the ASME Special Committee and formed the basis for the design methods described in Section III and Appendix E of Division 2 of Section VIII for evaluation of fatigue behavior in vessels. The PVRC effort is now continuing in the even more difficult region of high temperature, in which the effects of cyclic loading are combined with the plastic deformation of creep.

The simplified procedures of Division 1 of Section VIII are for the most part conservative for pressure vessels in conventional service and a detailed analysis of many pressure vessels constructed to the rules of Division 1 of Section VIII would show where the design could be optimized to conserve metal. However, it is recognized that the designer may be required to provide additional design considerations for pressure vessels to be used in severe types of service such as vessels for highly cyclic types of operation, for services which require superior reliability, or for nuclear service where periodic inspection is usually difficult and sometimes impossible. The need for design rules for such vessels led to the preparation of Section III and Division 2 of Section VIII.

The development of analytical and experimental techniques has made it possible to determine stresses in considerable detail. When the stress picture is brought into focus, it is not reasonable to retain the same values of allowable stress for the clear detailed picture as had previously been used for the less detailed one. Neither is it sufficient merely to raise the allowable stresses to reasonable values for the peak stresses, since peak stress by itself is not an adequate criterion of safety. A calculated value of stress means little until it is associated with its location and distribution in the structure and with the type of loading which produced it. Different types of stress have different degrees of significance and must, therefore, be assigned different allowable values. For example, the average hoop stress through the thickness of the wall of a vessel due to internal pressure must be held to a lower value than the stress at the root of a notch in the wall. Likewise, a thermal stress can often be allowed to reach a higher value than one which is produced by dead weight or pressure. Therefore the Special Committee developed a new set of design criteria which shifted the emphasis away from the use of standard configurations and toward the detailed analyses of stresses. The setting of allowable stress values required dividing stresses into categories and assigning different allowable values to different groups of categories.

With its knowledge of the problems enhanced and its technical ability to solve them improved by its work on Section III, in 1963 the Special Committee returned to the objective inherent to its original assignment: the development of Alternative Rules for Pressure Vessels. More specifically, the objective was the development of rules which would be consistent with the higher stress levels of Section III but retain or enhance the degree of safety inherent in the prior rules and achieve balanced construction. The result of this effort was the publication of Division 2, Alternative Rules for Pressure Vessels, of Section VIII in 1968.

The design requirements of Division 2 consist of a text, comparable to the paragraphs on design in part UG of Division 1, and three appendices:

Appendix 4, Design Based on Stress Analysis

Appendix 5, Design Based on Fatigue Analysis

Appendix 6, Experimental Stress Analysis

These three appendices are essentially identical to the analysis requirements of Section III. They provide a means whereby one can evaluate those vessels subject to severe service stresses or which contain configurations not considered within the text, using the detailed engineering approach which modern methods of stress analysis have made possible.

For reasons discussed in Part V of this booklet, neither Section III nor Division 2 of Section VIII consider metal temperatures in the creep range, at this time.

Because of the prominent role played by stress analysis in designing vessels by the rules of Section III or by the appendices of Division 2, and because of the necessity to integrate the design and analysis efforts, the procedure may be termed *"design by analysis."* This document provides an explanation of the strength theories, stress categories, and stress limits on which these design procedures are presently based. It also provides an explanation of the methods used for determining the suitability of vessels and parts for cyclic application of loads. In these respects, this document replaces the "Criteria of Section III of the ASME Boiler and Pressure Vessel Code for Nuclear Vessels" published by ASME in 1964.

Definitions

When discussing various combinations of stresses produced by various types of loading, it is important to use terms which are clearly defined. For example, the terms "membrane stress" and "secondary stress" are often used somewhat loosely. However, when a limit is to be placed on membrane stress, it is imperative that there must be no question about what is meant. Therefore the Special Committee spent a considerable amount of time in preparing a set of definitions. These definitions are given in Par. N-412 of Section III and Appendix 4, Par. 4-112 of Division 2.

Strength Theories

The stress state at any point in a structure may be completely defined by giving the magnitudes and directions of the three principal stresses. When two or three of these stresses are different from zero, the proximity to yielding must be determined by means of a strength theory. The theories most commonly used are the maximum stress theory, the maximum shear stress theory (also known as the Tresca criterion), and the distortion energy theory (also known as the octahedral shear theory and the Mises criterion). It has been known for many years that the maximum shear stress theory and the distortion energy theory are both much better than the maximum stress theory for predicting both yielding and fatigue failure in ductile metals. Section I and Division 1 of Section VIII use the maximum stress theory, by implication, but Section III and Division 2 use the maximum shear theory. Most experiments show that the distortion energy theory is even more accurate than the shear theory, but the shear theory was chosen because it is a little more conservative, it is easier to apply, and it offers some advantages in some applications of the fatigue analysis, as will be shown later.

The maximum shear stress at a point is defined as one-half of the algebraic difference between the largest and the smallest of the three principal stresses. Thus, if the principal stresses are σ_1, σ_2, and σ_3, and $\sigma_1 > \sigma_2 > \sigma_3$ (algebraically), the maximum shear stress is $\frac{1}{2}(\sigma_1 - \sigma_3)$. The maximum shear stress theory of failure states that yielding in a component occurs when the maximum shear stress reaches a value equal to the maximum shear stress at the yield point in a tensile test. In the tensile test, at yield, $\sigma_1 = S_y$, $\sigma_2 = 0$, and $\sigma_3 = 0$; therefore the maximum shear stress is $S_y/2$. Therefore yielding in the component occurs when

$$\frac{1}{2}(\sigma_1 - \sigma_3) = \frac{1}{2} S_y.$$

In order to avoid the unfamiliar and unnecessary operation of dividing both the calculated and the allowable stresses by two before comparing them, a new term called "equivalent intensity of combined stress" or, more briefly, "stress intensity" has been used. The stress intensity is defined as twice the maximum shear stress and is equal to the largest algebraic difference between any two of the three principal stresses. Thus the stress intensity is directly comparable to strength values found from tensile tests.

For the simple analyses on which the thickness formulas of Section I and Division 1 of Section VIII are based, it makes little difference whether the maximum stress theory or the maximum shear stress theory is used. For example, in the wall of a thin-walled cylindrical pressure vessel, remote from any discontinuities, the hoop stress is twice the axial stress and the radial stress on the inside is compressive and equal to the internal pressure, p. If the hoop stress is σ, the principal stresses are:

$$\sigma_1 = \sigma$$

$$\sigma_2 = \sigma/2$$

$$\sigma_3 = -p \, .$$

According to the maximum stress theory, the controlling stress is σ, since it is the largest of the three principal stresses. According to the maximum shear stress theory, the controlling stress is the stress intensity, which is $(\sigma + p)$. Since p is small in comparison with σ for a thin-walled vessel, there is little difference between the two theories. When a more detailed stress analysis is made, however, the difference between the two theories often becomes important.

II. STRESS CATEGORIES AND STRESS LIMITS

The various possible modes of failure which confront the pressure vessel designer are:

1. Excessive elastic deformation including elastic instability.
2. Excessive plastic deformation.
3. Brittle fracture.
4. Stress rupture/creep deformation (inelastic).
5. Plastic instability – incremental collapse.
6. High strain – low cycle fatigue.
7. Stress corrosion.
8. Corrosion fatigue.

In dealing with these various modes of failure, we will assume that the designer has at his disposal a picture of the state of stress within the part in question. This would be obtained either through calculation or measurements of both the mechanical and thermal stresses which could occur throughout the entire vessel during transient and steady state operations. The question one must ask is what do these numbers mean in relation to the adequacy of the design? Will they insure safe and satisfactory performance of a component? It is against these various failure modes that the pressure vessel designer must compare and interpret stress values. For example, elastic deformation and elastic instability (buckling) cannot be controlled by imposing upper limits to the calculated stress alone. One must consider, in addition, the geometry and stiffness of a component as well as properties of the material.

The plastic deformation mode of failure can, on the other hand, be controlled by imposing limits on calculated stress, but unlike the fatigue and stress corrosion modes of failure, peak stress does not tell the whole story. Careful consideration must be given to the consequences of yielding, and therefore the type of loading and the distribution of stress resulting therefrom must be carefully studied. The designer must consider, in addition to setting limits for allowable stress, some adequate and proper failure theory in order to define how the various stresses in a component react and contribute to the strength of that part.

As mentioned previously, different types of stress require different limits, and before establishing these limits it was necessary to choose the stress categories to which limits should be applied. The categories and sub-categories chosen were as follows:

A. Primary Stress.
   (1) General primary membrane stress.
   (2) Local primary membrane stress.
   (3) Primary bending stress.

B. Secondary Stress.

C. Peak Stress.

Definitions of these terms are given in Table N-414 of Section III and Appendix 4, Table 4-120.1 of Division 2, but some justification for the chosen categories is in order. The major stress categories are primary, secondary, and peak. Their chief characteristics may be described briefly as follows:

(a) Primary stress is a stress developed by the imposed loading which is necessary to satisfy the laws of equilibrium between external and internal forces and moments. The basic characteristic of a primary stress is that is not self-limiting. If a primary stress exceeds the yield strength of the material through the entire thickness, the prevention of failure is entirely dependent on the strain-hardening properties of the material.

(b) Secondary stress is a stress developed by the self-constraint of a structure. It must satisfy an imposed strain pattern rather than being in equilibrium with an external load. The basic characteristic of a secondary stress is that it is self-limiting. Local yielding and minor distortions can satisfy the discontinuity conditions or thermal expansions which cause the stress to occur.

(c) Peak stress is the highest stress in the region under consideration. The basic characteristic of a peak stress is that it causes no significant distortion and is objectionable mostly as a possible source of fatigue failure.

The need for dividing primary stress into membrane and bending components is that, as will be discussed later, limit design theory shows that the calculated value of a primary bending stress may be allowed to go higher than the calculated value of a primary membrane stress. The placing in the primary category of local membrane stress produced by mechanical loads, however, requires some explanation because this type of stress really has the basic characteristics of a secondary stress. It is self-limiting and when it exceeds yield, the external load will be resisted by other parts of the structure, but this shift may involve intolerable distortion and it was felt that it must be limited to a lower value than other secondary stresses, such as discontinuity bending stress and thermal stress.

Secondary stress could be divided into membrane and bending components, just as was done for primary stress, but after the removal of local membrane stress to the primary category, it appeared that all the remaining secondary stresses could be controlled by the same limit and this division was unnecessary.

Thermal stresses are never classed as primary stresses, but they appear in both of the other categories, secondary and peak. Thermal stresses which can produce distortion of the structure are placed in the secondary category and thermal stresses which result from almost complete suppression of the differential expansion, and thus cause no significant distortion, are classed as peak stresses.

A special exception to these general rules is the case of the stress due to a radial temperature gradient in a cylindrical shell. It is specifically stated in N-412 (m) (2) (6) of Section III, and in 4-112 (1) (2) (6) of Appendix 4 of Division 2, that this stress may be considered a local thermal stress. In reality, the linear portion of this gradient can cause deformation, but it was the opinion of the Special Committee that this exception could be safely made.

One of the commonest types of peak stress is that produced by a notch, which might be a small hole or a fillet. The phenomenon of stress concentration is well-known and requires no further explanation here.

Many cases arise in which it is not obvious which category a stress should be placed in, and considerable judgement is required. In order to standardize this procedure and use the judgement of the writers of the Code rather than the judgement of individual designers, a table was prepared covering most of the situations which arise in pressure vessel design and specifying which category each stress must be placed in. This table appears as Table N-413 of Section III and Appendix 4, Table 4-120.1 of Division 2.

The grouping of the stress categories for the purpose of applying limits to the stress intensities is illustrated in Fig. N-414 of Section III and Fig. 4-130.1 of Appendix 4 of Division 2. This diagram has been called the "hopper diagram" because it provides a hopper for each stress category. The calculated stresses are made to progress through the diagram in the direction of the arrows. Whenever a rectangular box appears, the sum of all the stress components which have entered the box are used to calculate the stress intensity, which is then compared to the allowable limit, shown in the circle adjacent to the rectangle. The following points should be noted in connection with this diagram:

(a) The symbols P_m, P_l, P_b, Q and F do not represent single quantities, but each represents a set of six quantities, three direct stress and three shear stress components. The addition of stresses from different categories must be performed at the component level, not after translating the stress components into a stress intensity. Similarly, the calculation of membrane stress intensity involves the averaging of stresses across a section, and this averaging must also be performed at the component level.

(b) The stresses in Category Q are those parts of the total stress which are categorized as secondary, and do not include primary stresses which may also exist at the same point. It should be noted, however, that a detailed stress analysis frequently gives the combination of primary and secondary stresses directly, and this calculated value represents the total of P (or P_L) + P_b + Q and not Q alone. It is not necessary to calculate Q separately since the stress limit (to be described later) applies to the total stress intensity. Similarly, if the stress in Category F is produced by a stress concentration, the quantity F is the additional stress produced by the notch, over and above the nominal stress, but it is not necessary to calculate F separately.

The potential failure modes and various stress categories are related to the Code provisions as follows:

(a) The primary stress limits are intended to prevent plastic deformation and to provide a nominal factor of safety on the ductile burst pressure.

(b) The primary plus secondary stress limits are intended to prevent excessive plastic deformation leading to incremental collapse, and to validate the application of elastic analysis when performing the fatigue evaluation.

(c) The peak stress limit is intended to prevent fatigue failure as a result of cyclic loadings.

(d) Special stress limits are provided for elastic and inelastic instability.

Protection against brittle fracture is provided by material selection, rather than by analysis. Protection against environmental conditions such as corrosion and radiation effects are the responsibility of the designer. The creep and stress rupture temperature range will be considered in later editions.

Basic Stress Intensity Limits

The choice of the basic stress intensity limits for the stress categories described above was accomplished by the application of limit design theory tempered by some engineering judgement and some conservative simplifications. The principles of limit design which were used can be described briefly as follows.

The assumption is made of perfect plasticity with no strain-hardening. This means that an idealized stress-strain curve of the type shown in Fig. 1 is assumed. Allowable stresses based on perfect plasticity and limit design theory may be considered as a floor below which a vessel made of any sufficiently ductile material will be safe. The actual strain-hardening properties of specific materials will give them larger or smaller margins above this floor.

In a structure as simple as a straight bar in tension, a load producing yield stress, S_y, results in "collapse." If the bar is loaded in bending, collapse does not occur until the load has been increased by a factor known as the "shape factor" of the cross section; at that time a "plastic hinge" is formed. The shape factor for a rectangular section in bending is 1.5. When the primary stress in a rectangular section consists of a combination of bending and axial tension, the value of the limit load depends on the ratio between the tensile and bending loads. Fig. 2 shows the value of the maximum calculated stress at the outer fiber of a rectangular section which would be required to produce a plastic hinge, plotted against the average tensile stress across the section, both values expressed as multiples of the yield stress, S_y. When the average tensile stress, P_m, is zero, the failure stress for bending is 1.5 S_y. When the average tensile stress is S_y, no additional bending stress, P_b, may be applied.

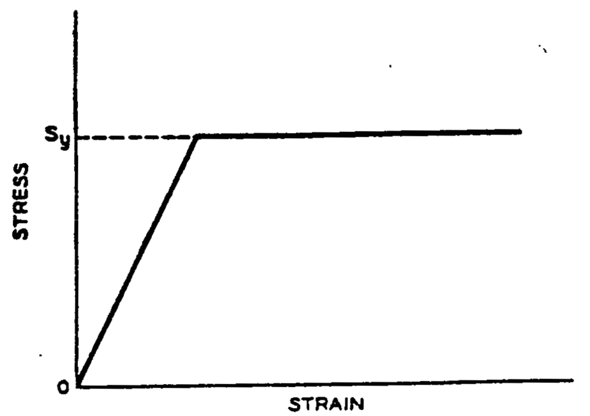


FIGURE 1. IDEALIZED STRESS – STRAIN RELATIONSHIP

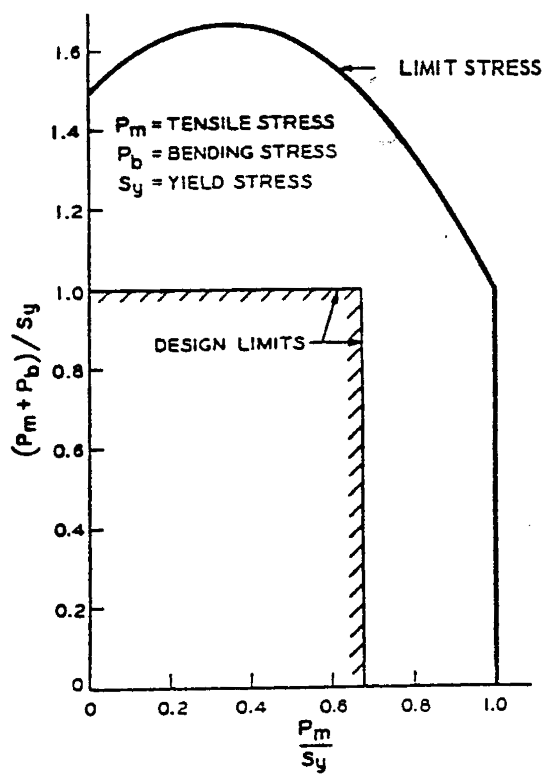
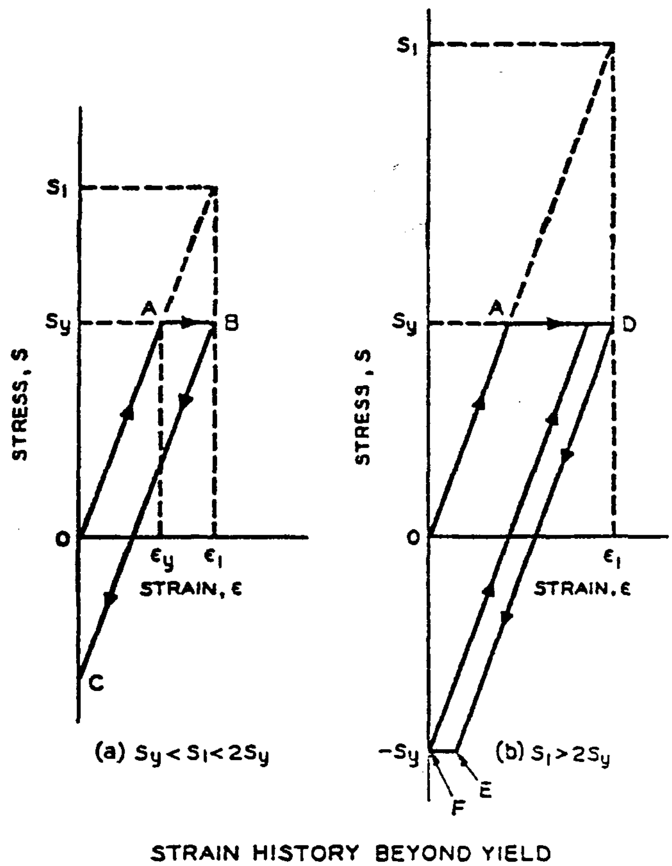


FIGURE 2. LIMIT STRESS FOR COMBINED TENSION AND BENDING (RECTANGULAR SECTION)

Figure 2 was used to choose allowable values, in terms of the yield stress, for general primary membrane stress, P_m, and primary membrane-plus-bending stress, $P_m + P_b$. It may be seen that limiting P_m to (2/3) S_y and $P_m + P_b$ to S_y provides adequate safety. The safety factor is not constant for all combinations of tension and bending, but a design rule to provide a uniform safety factor would be needlessly complicated.

In the study of allowable secondary stresses, a calculated elastic stress range equal to twice the yield stress has a very special significance. It determines the borderline between loads which, when repetitively applied, allow the structure to "shake down" to elastic action and loads which produce plastic action each time they are applied. The theory of limit design provides rigorous proof of this statement, but the validity of the concept can easily be visualized. Consider, for example, the outer fiber of a beam which is strained in tension to a strain value ϵ_1, somewhat beyond the yield strain as shown in Fig. 3 (a) by the path *OAB*. The calculated elastic stress would be $S = S_1 = E\epsilon_1$. Since we are considering the case of a secondary stress, we shall assume that the nature of the loading is such as to cycle the strain from zero to ϵ_1 and back to zero, rather than cycling the stress from zero to S_1, and back to zero. When the beam is returned to its undeflected position, *O*, the outer fiber has a residual compressive stress of magnitude $S_1 - S_y$. On any subsequent loading, this residual compression must be removed before the stress goes into tension and thus the elastic range has been increased by the quantity $S_1 - S_y$. If $S_1 = 2S_y$, the elastic range becomes $2S_y$, but if $S_1 > 2S_y$, the fiber yields in compression, as shown by *EF* in Fig. 3 (b) and all subsequent cycles produce plastic strain. Therefore, $2S_y$ is the maximum value of calculated secondary elastic stress which will "shake down" to purely elastic action.



STRAIN HISTORY BEYOND YIELD

FIGURE 3.

An important point to note from the foregoing discussion of primary and secondary stresses is that 1.5 S_y is the *failure* stress for primary bending, whereas for secondary bending $2S_y$ is merely the threshold beyond which some plastic action occurs. Therefore the allowable design stress for primary bending must be reduced below $1.5S_y$ to, say, $1.0S_y$, whereas $2S_y$ is a safe design value for secondary bending since a little plastic action during overloads is tolerable. The same type of analysis shows that $2S_y$ is also a safe design value for secondary membrane tension. As described previously, local membrane stress produced by mechanical load has the characteristics of a secondary stress but has been arbitrarily placed in the primary category. In order to avoid excessive distortion, it has been assigned an allowable stress level of S_y, which is 50 per cent higher than the allowable for general primary membrane stress but precludes excessive yielding.

We have now shown how the allowable stresses for the first four stress categories listed in the previous section should be related to the yield strength of the material. The last category, peak stress, is related only to fatigue, and will be discussed later. With the exception of some of the special stress limits, the allowables in Codes are not expressed in terms of the yield strength, but rather as multiples of the tabulated value S_m, which is the allowable for general primary membrane stress. In assigning allowable stress values to a variety of materials with widely varying ductilities and widely varying strain-hardening properties, the yield strength alone is not a sufficient criterion. In order to prevent unsafe designs in materials with low ductility and in materials with high yield-to-tensile ratios, the Code has always considered both the yield strength and the ultimate tensile strength in assigning allowable stresses. This principle has not been changed in Section III or Division 2 but the chosen fractions of the mechanical properties have been increased to two-thirds yield strength and one-third ultimate strength instead of five-eighths yield strength (for ferrous materials) and one-fourth ultimate strength. The Special Committee believed that this increase was quite safe because the detailed stress analysis required eliminates the need for a large safety factor to cover unanalyzed areas. The stress intensity limits for the various categories given are such that the multiples of yield strength described above are never exceeded.

The allowable stress intensity for austenitic steels and some non-ferrous materials, at temperatures above 100 F, may exceed $(2/3)S_y$ and may reach $0.9S_y$ at temperature. Some explanation of the use of up to $0.9S_y$ for these materials as a basis for S_m is needed in view of Figure 2 because this figure would imply that loads in excess of the limit load are permitted. The explanation lies in the different nature of these materials' stress strain diagram. These materials have no well-defined yield point but have strong strain-hardening capabilities so that their yield strength is effectively raised as they are highly loaded. This means that some permanent deformation during the first loading cycle may occur, however the basic structural integrity is comparable to that obtained with ferritic materials. This is equivalent to choosing a somewhat different definition of the "design yield strength" for those materials which have no sharply defined yield point and which have strong strain-hardening characteristics. Therefore, the S_m value in the code tables, regardless of material, can be thought of as being no less than 2/3 of the "design yield strength" for the material in evaluating the primary and secondary stresses.

Table I summarizes the basic stress limits and shows the multiples of yield strength and ultimate strength which these limits do not exceed.

TABLE I. BASIC STRESS INTENSITY LIMITS

Stress Intensity	Tabulated Value	Yield Strength	Ultimate Tensile Strength
General primary membrane (P_m)	S_m	$\leqslant \frac{2}{3} S_y$	$\leqslant \frac{1}{3} S_u$
Local primary membrane (P_l)	$1.5\ S_m$	$\leqslant S_y$	$\leqslant \frac{1}{2} S_u$
Primary membrane plus bending ($P_l + P_b$))	$1.5\ S_m$	$\leqslant S_y$	$\leqslant \frac{1}{2} S_u$
Primary plus secondary ($P_l + P_b + Q$)	$3\ S_m$	$\leqslant 2S_y$	$\leqslant S_u$

Stresses Above Yield Strength

The primary criterion of the structural adequacy of a design, is that the stresses, as determined by calculation or experimental stress analysis, shall not exceed the specified allowable limits. It frequently happens that both the calculated stress and allowable stress exceeds the yield strength of the material. Nevertheless, unless stated specifically otherwise, it is expected that calculations be made on the assumption of elastic behavior.

Allowable stresses higher than yield appear in the values for primary-plus-secondary stress and in the fatigue curves. In the case of the former, the justification for allowing calculated stresses higher than yield is that the limits are such as to assure shake-down to elastic action after repeated loading has established a favorable pattern of residual stresses. Therefore the assumption of elastic behavior is justified because it really exists in all load cycles subsequent to shake-down.

In the case of fatigue analysis, plastic action can actually persist throughout the life of the vessel, and the justification for the specified procedure is somewhat different. Repetitive plastic action occurs only as the result of peak stresses in relatively localized regions and these regions are intimately connected to larger regions of the vessel which behave elastically. A typical example is the peak stress at the root of a notch, in a fillet, or at the edge of a small hole. The material in these small regions is strain-cycled rather than stress-cycled (as will be discussed later) and the elastic calculations give numbers which have the dimensions of stress but are really proportional to the strain. The factor of proportionality for uniaxial stress is, of course, the modulus of elasticity. The fatigue design curves have been specially designed to give numbers comparable to these fictitious calculated stresses. The curves are based on strain-cycling data and the strain values have been multiplied by the modulus of elasticity. Therefore stress intensities calculated from the familiar formulas of strength-of-materials texts are directly comparable to the allowable stress values in the fatigue curves.

III. FATIGUE ANALYSIS

One of the important innovations in Section III and Division 2 as compared to Sections I and Division 1 of Section VIII, is the recognition of fatigue as a possible mode of failure and the provision of specific rules for its prevention. Fatigue has been a major consideration for many years in the design of rotating machinery and aircraft, where the expected number of cycles is in the millions and can usually be considered infinite for all practical purposes. For the case of large numbers of cycles, the primary concern is the endurance limit, which is the stress which can be applied an infinite number of times without producing failure. In pressure vessels, however, the number of stress cycles applied during the specified life seldom exceeds 10^5 and is frequently only a few thousand. Therefore, in order to make fatigue analysis practical for pressure vessels, it was necessary to develop some new concepts not previously used in machine design [1,2].

Use of Strain-Controlled Fatigue Data

The chief difference between high-cycle fatigue and low-cycle fatigue is the fact that the former involves little or no plastic action, whereas failure in a few thousand cycles can be produced only by strains in excess of the yield strain. In the plastic region large changes in strain can be produced by small changes in stress. Fatigue damage in the plastic region has been found to be a function of plastic strain and therefore fatigue curves for use in this region should be based on tests in which strain rather than stress is the controlled variable. As a matter of convenience, the strain values used in the tests are multiplied by the elastic modulus to give a fictitious stress which is not the actual stress applied but has the advantage of being directly comparable to stresses calculated on the assumption of elastic behavior.

The general procedure used in evaluating the strain-controlled fatigue data was to obtain a "best fit" for the quantities A and B in the equation

$$S = \frac{E}{4\sqrt{N}} \ln \frac{100}{100 - A} + B \tag{1}$$

where E = elastic modulus (psi)
N = number of cycles to failure
S = strain amplitude times elastic modulus

It is possible to estimate the fatigue properties by taking A as the percentage reduction of area in a tensile test, RA, and B as the endurance limit, S_e.

The use of strain instead of stress and the consideration of plastic action have necessitated some additional departures from the conventional methods of studying fatigue problems. It has been common practice in the past to use lower stress concentration factors for small numbers of cycles than for large numbers of cycles. This is reasonable when the allowable stresses are based on stress-fatigue data, but is not advisable when strain-fatigue data are used. Fig. 4 shows typical relationships between stress, S, and cycles-to-failure, N, from (A) strain cycling tests on unnotched specimens, (B) stress-cycling tests on unnotched specimens, and (C) stress-cycling tests on notched specimens. The ratio between the ordinates of curves (B) and (C) decreases with decreasing cycles-to-failure, and this is the basis for the commonly-accepted practice of using lower values

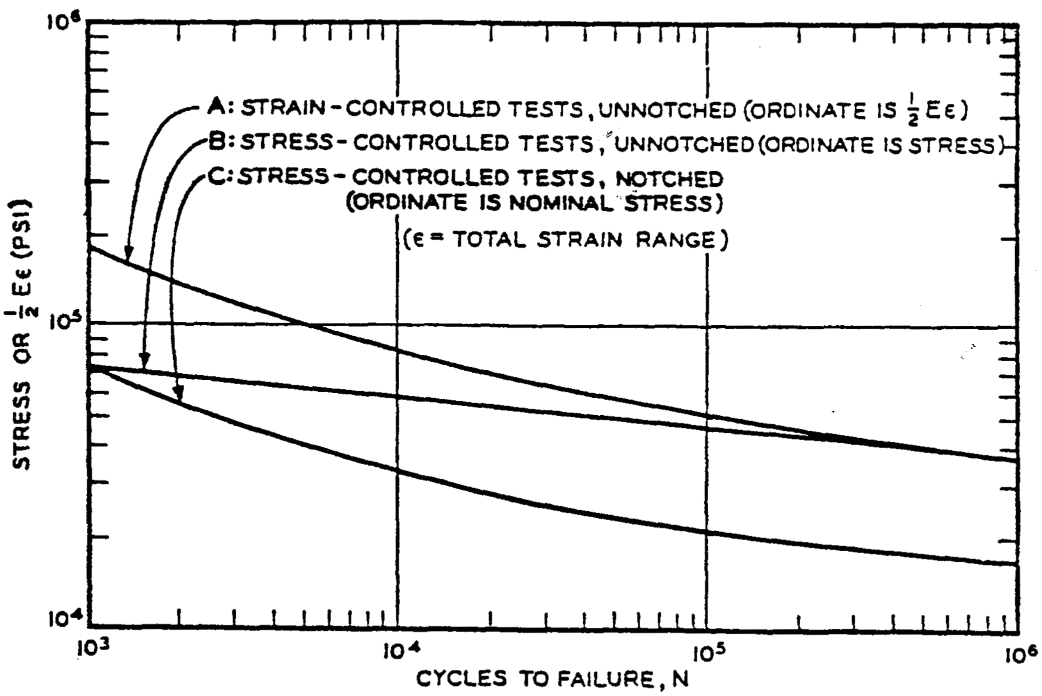


FIG. 4. TYPICAL RELATIONSHIP BETWEEN STRESS, STRAIN, AND CYCLES-TO-FAILURE.

of K (stress concentration factor) for lower values of N. In (C), however, although nominal stress is the controlled parameter, the material in the root of the notch is really being strain cycled, because the surrounding material is at a lower stress and behaves elastically. Therefore it should be expected that the ratio between curves (A) and (C) should be independent of N and equal to K. For this reason it is recommended in Section III and Division 2 of Section VIII that the same value of K be used regardless of the number of cycles involved.

The choice of an appropriate stress concentration factor is not an easy one to make. For fillets, grooves, holes, etc. of known geometry, it is safe to use the theoretical stress concentration factors found in such references as [3] and [4], even though strain concentrations can sometimes exceed the theoretical stress concentration factors. The use of the theoretical factor as a safe upper limit is justified, however, since strain concentrations significantly higher than the stress concentrations only occur when gross yielding is present in the surrounding material, and this situation is prevented by the use of basic stress limits which assure shake-down to elastic action. For very sharp notches it is well known that the theoretical factors grossly overestimate the true weakening effect of the notch in the low and medium strength materials used for pressure vessels. Therefore no factor higher than 5 need ever be used for any configuration allowed by the design rules and an upper limit of 4 is specified for some specific constructions such as fillet welds and screw threads. When fatigue tests are made to find the appropriate factor for a given material and configuration, they should be made with a material of comparable notch sensitivity and failure should occur in a reasonably large number of cycles (> 1000) so that the test does not involve gross yielding.

Effect of Mean Stress

Another deviation from common practice occurs in the consideration of fluctuating stress, which is a situation where the stress fluctuates around a mean value different from zero, as shown in Fig. 5. The evaluation of the effects of mean stress is commonly accomplished by use of the modified Goodman diagram, as shown in Fig. 6, where mean

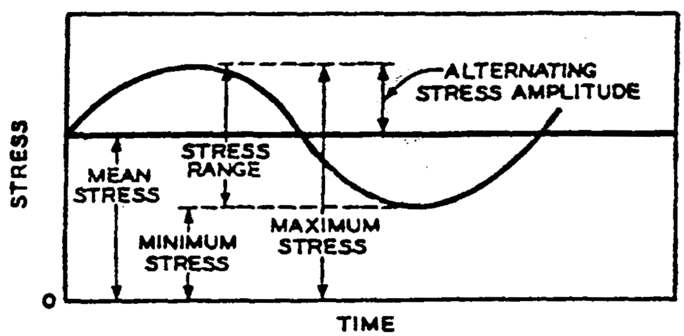


FIG. 5. STRESS FLUCTUATION AROUND A MEAN VALUE.

stress is plotted as the abscissa and the amplitude (half range) of the fluctuation is plotted as the ordinate. The straight line joining the endurance limit, S_e, (where $S_N = S_e$) on the vertical axis (point E) with the ultimate strength, S_u, on the horizontal axis (point D) is a conservative approximation of the combinations of mean and alternating stress which produce failure in large numbers of cycles. A little consideration of this diagram shows that not all points below the "failure" line, ED, are feasible. Any combination of mean and alternating stresses which results in a stress excursion above the yield strength will produce a shift in the mean stress which keeps the maximum stress during the cycle at the yield value. This shift has already been illustrated by the strain history shown in Fig. 3. The feasible combinations of mean and alternating stress are all contained within the 45 degree triangle AOB or on the vertical axis above A, where A is the yield strength on the vertical axis and B is the yield strength on the horizontal axis. Regardless of the conditions under which any test or service cycle is started, the true conditions after the application of a few cycles must fall within this region because all combinations above AB have a maximum stress above yield and there is a consequent reduction of mean stress which shifts the conditions to a point on the line AB or all the way to the vertical axis.

It may be seen from the foregoing discussion that the value of mean stress to be used in the fatigue evaluation is not always the value which is calculated directly from the imposed loading cycle. When the loading cycle produces calculated stresses which exceed

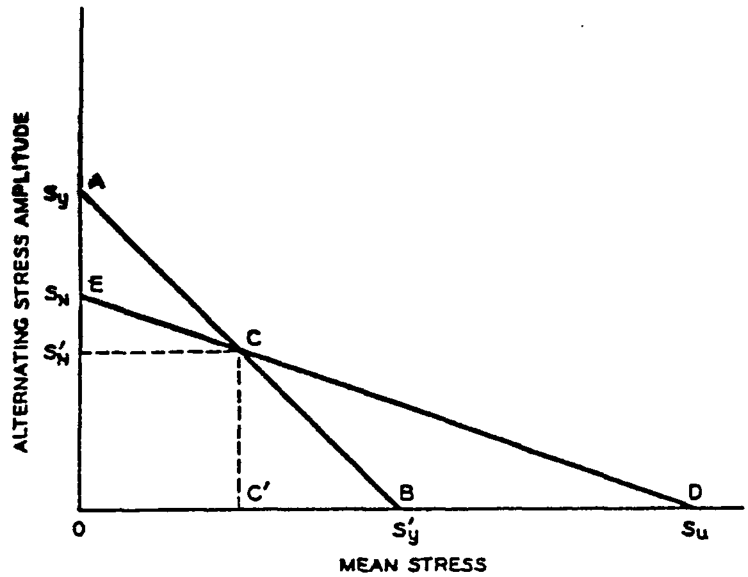


FIG. 6. MODIFIED GOODMAN DIAGRAM.

the yield strength at any time, it is necessary to calculate an adjusted value of mean stress before completing the fatigue evaluation. The rules for calculating this adjusted value when the modified Goodman diagram is applied may be summarized as follows:

Let S'_{mean} = basic value of mean stress (calculated directly from loading cycle)
S_{mean} = adjusted value of mean stress
S_{alt} = amplitude (half range) of stress fluctuation
S_y = yield strength

$$\left.\begin{array}{l} \text{If } S_{alt} + S'_{mean} \leq S_y\,,\ S_{mean} = S'_{mean} \\ \text{If } S_{alt} + S'_{mean} > S_y \text{ and } S_{alt} < S_y,\ S_{mean} = S_y - S_{alt} \\ \text{If } S_{alt} \geq S_y,\ S_{mean} = 0. \end{array}\right\} \qquad (2)$$

The fatigue curves are based on tests involving complete stress reversal, that is, $S_{mean} = 0$. Since the presence of a mean stress component detracts from the fatigue resistance of the material, it is necessary to determine the equivalent alternating stress component for zero mean stress before entering the fatigue curve. This quantity, designated S_{eq}, is the alternating stress component which produces the same fatigue damage at zero mean stress as the actual alternating stress component, S_{alt}, produces at the existing value of mean stress. It can be obtained graphically from the Goodman diagram by projecting a line as shown in Fig. 7 from S_u through the point (S_{mean}, S_{alt}) to the vertical axis. It is usually easier, however, to use the simple formula

$$S_{eq} = \frac{S_{alt}}{1 - \dfrac{S_{mean}}{S_u}}. \qquad (3)$$

S_{eq} is the value of stress to be used in entering the fatigue curve to find the allowable number of cycles.

The foregoing discussion of mean stress and the shift which it undergoes when yielding occurs leads to another necessary deviation from standard procedures. In applying stress concentration factors to the case of fluctuating stress, it has been the common practice to apply the factor to only the alternating component. This is not a logical procedure, however, because the material will respond in the same way to a given load regardless of whether the load will later turn out to be steady or fluctuating. It is more logical to apply the concentration factor to both the mean and the alternating component and then consider the reduction which yielding produces in the mean component. It is important to remember that the concentration factor must be applied before the adjustment for yielding is made. The following example shows that the common practice of applying the concentration factor to only the alternating component gives a rough approximation to the real situation but can sometimes be unconservative.

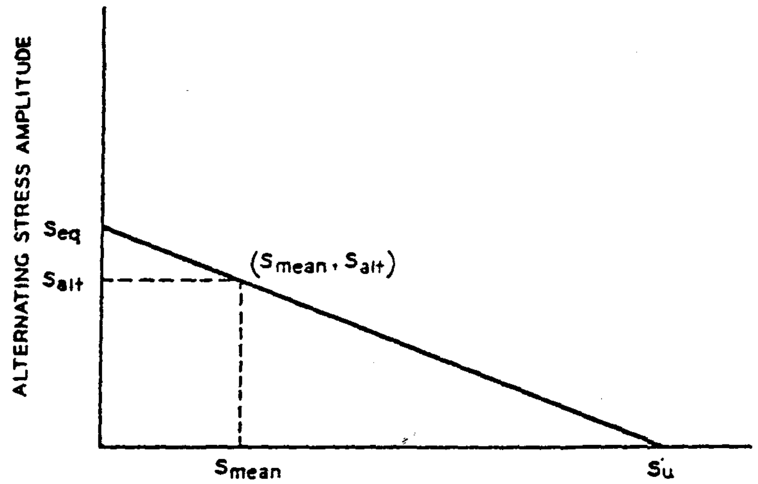


FIG. 7. GRAPHICAL DETERMINATION OF S_{eq}.

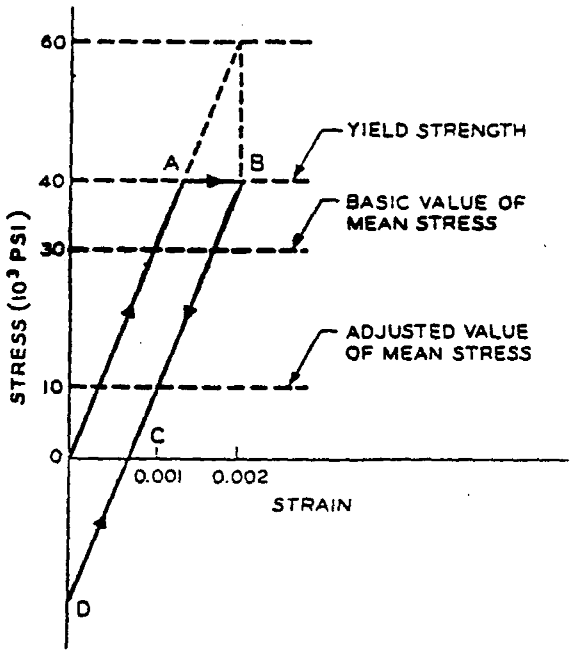


FIGURE 8. IDEALIZED STRESS VS STRAIN HISTORY

Take the case of a material with 80,000 psi tensile strength, 40,000 psi yield strength and 30×10^6 psi modulus made into a notched bar with a stress concentration factor of 3. The bar is cycled between nominal tensile stress values of 0 and 20,000 psi. Common practice would call S_{mean}, the mean stress, 10,000 psi and S_{alt}, the alternating component, $(1/2) \times 3 \times 20,000 = 30,000$ psi. The stress-strain history of the material at the root of the notch would be, in idealized form, as shown in Fig. 8. The calculated maximum stress, assuming elastic behavior, is 60,000 psi. The basic value of mean stress, S'_{mean}, is 30,000 psi, but since $S_{alt} + S'_{mean} = 60,000 \text{ psi} > S_y$ and $S_{alt} = 30,000 \text{ psi} < S_y$,

$$S_{mean} = S_y - S_{alt} = 40,000 - 30,000 = 10,000 \text{ psi}$$

and

$$S_{eq} = \frac{30,000}{1 - \dfrac{10,000}{80,000}} = 34,300 \text{ psi.}$$

It so happens that, for the case chosen, the common practice gives exactly the same result as the proposed method. Thus, the yielding during the first cycle is seen to be the justification for the common practice of ignoring the stress concentration factor when determining the mean stress component. The common practice, however, would have given the same result regardless of the yield strength of the material, whereas the proposed method gives different mean stresses for different yield strengths. For example, if the yield strength had been 50,000 psi, S_{mean} would have been 20,000 psi and S_{eq} by the proposed method would have been 40,000 psi. The common practice would have given 34,300 psi for S_{eq} and too large a number of cycles would have been allowed.

For parts of the structure, particularly if welding is used, the residual stress may produce a value of mean stress higher than that calculated by the procedure. Therefore it would be advisable and also much easier to adjust the fatigue curve downward enough to allow for the maximum possible effect of mean stress. It will be shown here that this adjustment is small for the case of low and medium-strength materials.

As a first step in finding the required adjustment of the fatigue curve, let us find how the mean stress affects the amplitude of alternating stress which is required to produce fatigue failure. In the modified Goodman diagram of Fig. 6 it may be seen that at zero mean stress the required amplitude for failure in N cycles is designated S_N. As the mean stress increases along OC', the required amplitude of alternating stress decreases along the line EC. If we try to increase the mean stress beyond C', yielding occurs and the mean stress reverts to C'. Therefore C' represents the highest value of mean stress which has any effect on fatigue life. Since S_N' in Fig. 6 is the alternating stress required to produce failure in N cycles when the mean stress is at C', S_N' is the value to which the point on the fatigue curve at N cycles must be adjusted if the effects of mean stress are to be ignored. From the geometry of Fig. 6, it can be shown that

$$S_N' = S_n \left[\frac{S_u - S_y}{S_u - S_N} \right] \quad \text{for } S_N < S_y \tag{4}$$

When N decreases to the point where $S_N \geqslant S_y$, then $S_N' = S_N$ and no adjustment of this region of the curve is required.

Figures 9, 10 and 11 show the fatigue data which were used to construct the design fatique curves for certain materials. In each case the solid line is the best-fit failure curve for zero mean stress and the dotted line is the curve adjusted in accordance with (4). Fig. 11 for stainless steel and nickel-chrome-iron alloy has no dotted line because the fatigue limit is higher than the yield strength over the whole range of cycles. A single design curve is used for carbon and low-alloy steel below 80,000 psi ultimate tensile strength because, as may be noted from Figs. 9 and 10, the adjusted curves for these classes of material were nearly identical.

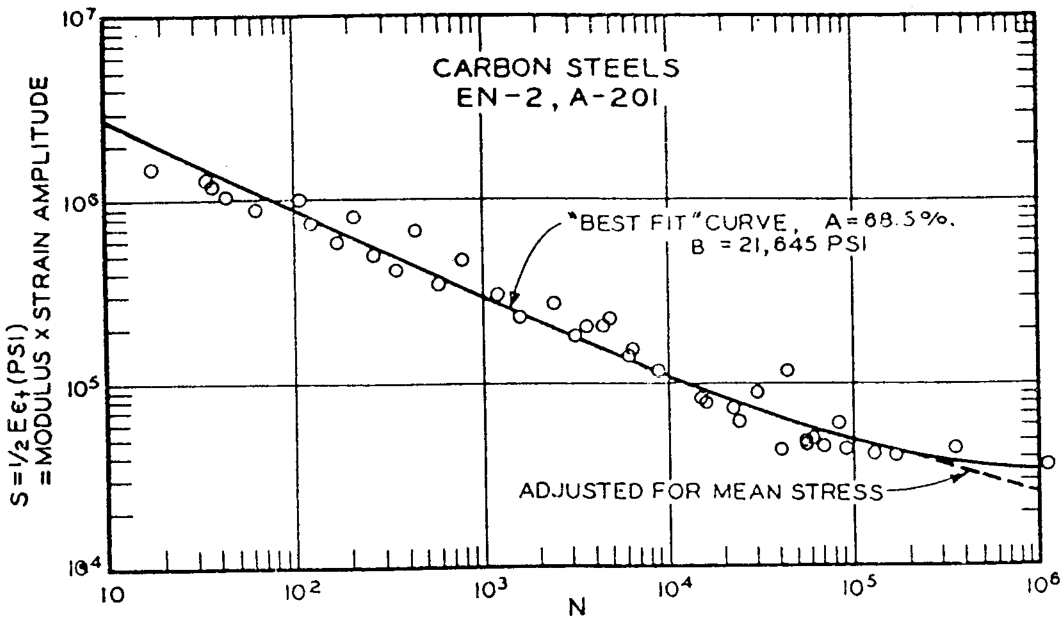


FIG. 9. FATIGUE DATA – CARBON STEELS.

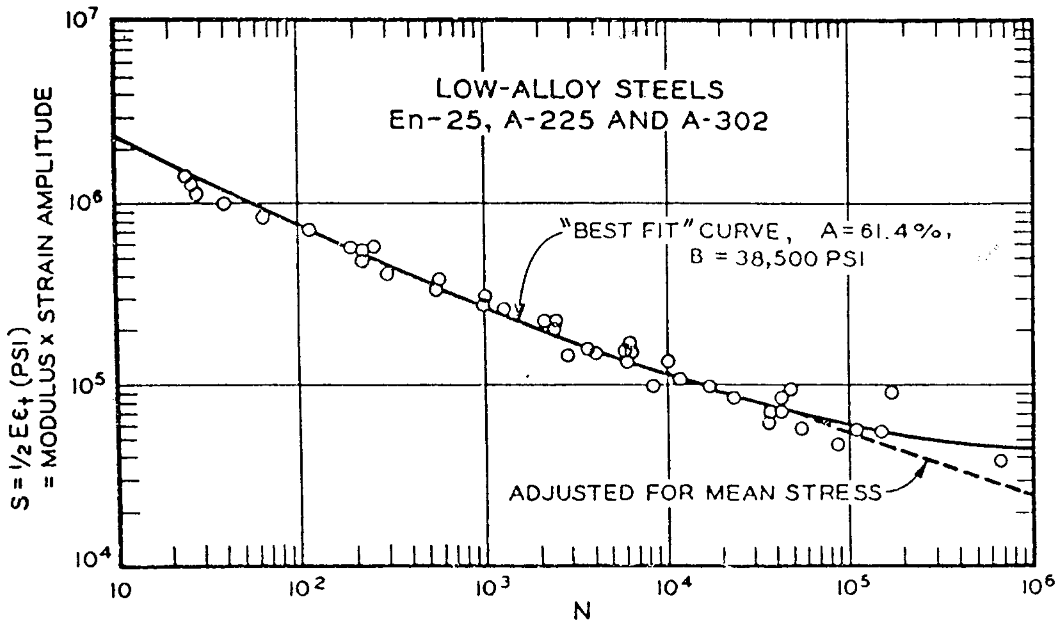


FIG. 10. FATIGUE DATA – LOW-ALLOY STEELS.

For the case of high-strength, heat-treated, bolting materials, the heat treatment increases the yield strength of the material much more than it increases either the ultimate strength, S_u, or the fatigue limit, S_N. Inspection of (4) shows that for such cases, S_N' becomes a small fraction of S_N and thus the correction for the maximum effect of mean stress becomes unduly conservative.

Test data indicate that use of the Peterson cubic equation

$$S_{eq} = \frac{7S_a}{8 - \left(1 + \frac{S_{mean}}{S_a}\right)^3} \tag{5}$$

results in an improved method for high strength bolting materials, and this equation has been used in preparing design fatigue curves for such bolts [10].

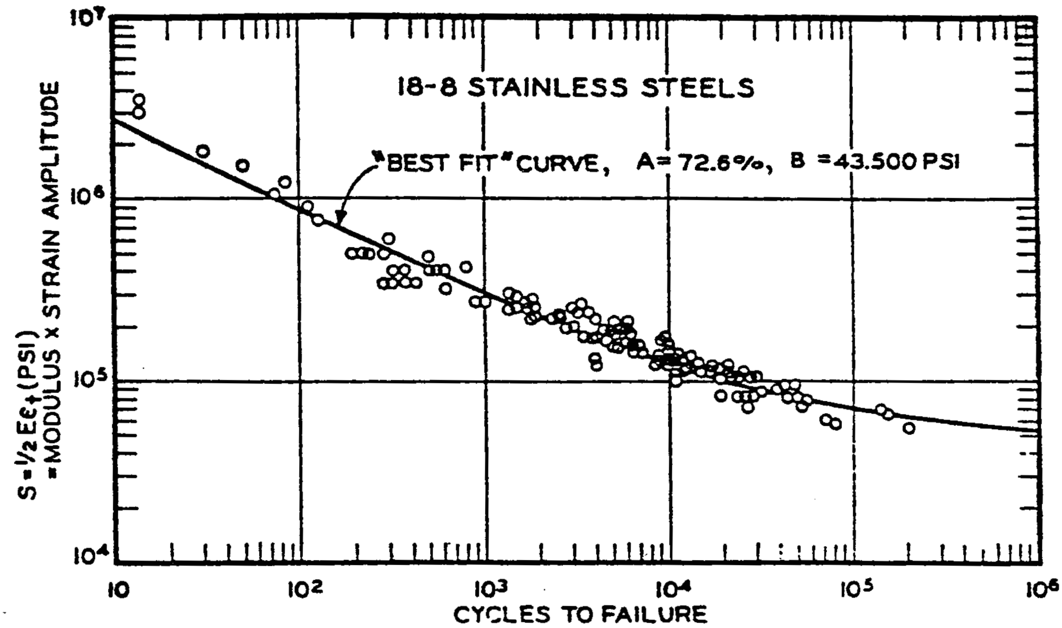


FIG. 11. FATIGUE DATA – STAINLESS STEELS.

Procedure for Fatigue Evaluation

The step-by-step procedure for determing whether or not the fluctuation of stresses at a given point is acceptable is given in detail in Par. N-415.2 of Section III and Appendix 5 of Division 2. The procedure is based on the maximum shear stress theory of failure and consists of finding the amplitude (half full range) through which the maximum shear stress fluctuates. Just as in the case of the basic stress limits, the stress differences and stress intensities (twice maximum shear stress) are used in place of the shear stress itself.

At each point on the vessel at any given time there are three principal stresses, σ_1, σ_2, and σ_3 and three stress differences, S_{12}, S_{23}, and S_{31}. The stress intensity is the largest of the three stress differences and is usually considered to have no direction or sign, just as for the strain energy of distortion. When considering fluctuating stresses, however, this concept of non-directionality can lead to errors when the sign of the shear stress changes during the cycle. Therefore the range of fluctuation must be determined from the stress differences in order to find the full algebraic range. The alternating stress intensity, S_{alt}, is the largest of the amplitudes of the three stress differences. This feature of being able to maintain directionality and thus find the algebraic range of fluctuation is one reason why the maximum shear stress theory rather than the strain energy of distortion theory was chosen.

When the directions of the principal stresses change during the cycle (regardless of whether the stress differences change sign), the non-directional strain energy of distortion theory breaks down completely. This has been demonstrated experimentally by Findley and his associates [5] who produced fatigue failures in a rotating specimen compressed across a diameter. The load was fixed while the specimen rotated. Thus the principal stresses rotated but the strain energy of distortion remained constant. The procedure outlined in Par. N-415.2(b) and 5-110(b) is consistent with the results of Findley's tests and uses the range of shear stress on a fixed place as the criterion of failure. The procedure brings in the effect of rotation of the principal stresses by considering only the *changes* in shear stress which occur in each plane between the two extremes of the stress cycle.

Cumulative Damage

In many cases a point on a vessel will be subjected to a variety of stress cycles during its lifetime. Some of these cycles will have amplitudes below the endurance limit of the material and some will have amplitudes of varying amounts above the endurance limit. The cumulative effect of these various cycles is evaluated by means of a linear damage relationship in which it is assumed that if N_1 cycles would produce failure at a stress level S_1, then n_1 cycles at the same stress level would use up the fraction n_1/N_1 of the total life. Failure occurs when the cumulative usage factor, which is the sum $n_1/N_1 + n_2/N_2 + n_3/N_3 + \ldots$ is equal to 1.0. Other hypotheses for estimating cumulative fatigue damage have been proposed and some have been shown to be more accurate than the linear damage assumption. Better accuracy could be obtained, however, only if the sequence of the stress cycles were known in considerable detail, and this information is not apt to be known with any certainty at the time the vessel is being designed. Tests have shown [6] that the linear assumption is quite good when cycles of large and small stress magnitude are fairly evenly distributed throughout the life of the member, and therefore this assumption was considered to cover the majority of cases with sufficient accuracy. It is of interest to note that a concentration of the larger stress cycles near the beginning of life tends to accelerate failure, whereas if the smaller stresses are applied first and followed by progressively higher stresses, the cumulative usage factor can be "coaxed" up to a value as high as 4 or 5.

When stress cycles of various frequencies are intermixed through the life of the vessel, it is important to identify correctly the range and number of repetitions of each type of cycle. It must be remembered that a small increase in stress range can produce a large decrease in fatigue life, and this relationship varies for different portions of the fatigue curve. Therefore the effect of superposing two stress amplitudes cannot be evaluated by adding the usage factors obtained from each amplitude by itself. The stresses must be added before calculating the usage factors. Consider, for example, the case of a thermal transient which occurs in a pressurized vessel. Suppose that at a given point the pressure stress is 20,000 psi tension and the added stress from the thermal transient is 70,000 psi tension. If the thermal cycle occurs 10,000 times during the design life and the vessel is pressurized 1000 times, the usage factor should be based on 1000 cycles with a range from zero to 90,000 psi and 9000 cycles with a range from 20,000 psi to 90,000 psi. Another example, is given in N-415.2(d)(1) and in 5-110(e).

Exemption from Fatigue Analysis

The fatigue analysis of a vessel is quite apt to be one of the most laborious and time-consuming parts of the design procedure and this engineering effort is not warranted for vessels which are not subjected to cyclic operation. However, there is no obvious borderline between cyclic and non-cyclic operation. No operation is completely non-cyclic, since startup and shutdown is itself a cycle. Therefore, fatigue cannot be completely ignored, but Par. N-415 and AD-160 gives a set of rules which may be used to justify the by-passing of the detailed fatigue analysis for vessels in which the danger of fatigue failure is remote. The application of these rules requires only that the designer know the specified pressure fluctuations and that he have some knowledge of the temperature differences which will exist between different points in the vessel. He does not need to determine stress concentration factors or to calculate cyclic thermal stress ranges. He must, however, be sure that the basic stress limits of N-414.1 to 414.4 or of 4-131 to 4-134 are met, which may involve some calculation of the most severe thermal stresses.

The rules for exemption from fatigue analysis are based on a set of assumptions, some of which are highly conservative and some of which are not conservative, but is believed that the conservatisms outweigh the unconservatisms. These assumptions are:

(1) The worst geometrical stress concentration factor to be considered is 2. This assumption is unconservative since $K = 4$ is specified for some geometries.

(2) The concentration factor of 2 occurs at a point where the nominal stress is $3S_m$, the highest allowable value of primary-plus-secondary stress. This is a conservative assumption. The net result of assumptions 1 and 2 is that the peak stress due to pressure is assumed to be $6S_m$, which appears to be a safe assumption for a good design.

(3) *All* significant pressure cycles and thermal cycles have the same stress range as the *most severe* cycle. This is a highly conservative assumption. (A "significant" cycle is defined as one which produces a stress amplitude higher than the endurance limit of the material).

(4) The highest stress produced by a pressure cycle does not coincide with the highest stress produced by a thermal cycle. This is unconservative and must be balanced against the conservatism of assumption 3.

(5) The calculated stress produced by a temperature difference ΔT between two points does not exceed $2E\alpha\Delta T$, but the peak stress is raised to $4E\alpha\Delta T$ because of the assumption that a K value of 2 is present. This assumption is conservative, as evidenced by the following examples of thermal stress:

(a) For the case of a linear thermal gradient through the thickness of a vessel wall, if the temperature difference between the inside and the outside of the wall is ΔT, the stress is

$$\sigma = \frac{E\alpha\Delta T}{2(1-\nu)} = .715\, E\alpha\Delta T \text{ (for } \nu = 0.3\text{)} .$$

(b) When a vessel wall is subjected to a sudden change of temperature, ΔT, so that the temperature change only penetrates a short distance into the wall thickness, the thermal stress is

$$\sigma = \frac{E\alpha\Delta T}{1-\nu} = 1.43\, E\alpha\Delta T \text{ (for } \nu = 0.3\text{)} .$$

(c) When the average temperature of a nozzle is ΔT degrees different from that of the rigid wall to which it is attached, the upper limit to the magnitude of the discontinuity stress is

$$\sigma = 1.83\, E\alpha\Delta T \text{ (for } \nu = 0.3\text{)}.$$

Thus the coefficient of $E\alpha\Delta T$ is always less than the assumed value of 2.0.

When the two points in the vessel whose temperatures differ by ΔT are separated from each other by more than $2\sqrt{Rt}$, there is sufficient flexibility between the two points to produce a significant reduction in thermal stress. Therefore only temperature differences between "adjacent" points need be considered.

Experimental Verification of Design Fatigue Curves

The design fatigue curves are based primarily on strain-controlled fatigue tests of small polished specimens. A best-fit to the experimental data was obtained by applying the method of least squares to the logarithms of the experimental values. The design stress values were obtained from the best-fit curves by applying a factor of two on stress or a factor of twenty on cycles, whichever was more conservative at each point. These factors were intended to cover such effects as environment, size effect, and scatter of data, and thus it is not to be expected that a vessel will actually operate safely for twenty times its specified life.

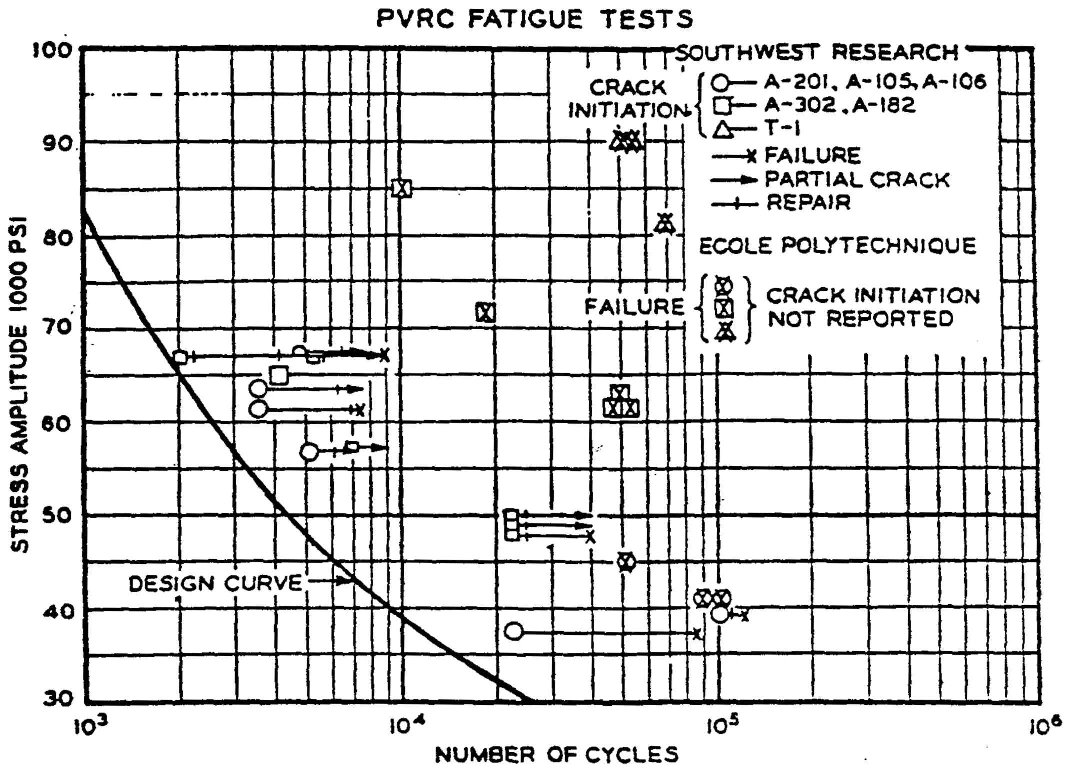


FIG. 12. PVRC FATIGUE TESTS.

The appropriateness of the chosen safety factors for fatigue has recently been demonstrated by tests conducted by the Pressure Vessel Research Committee [7,8]. In these tests 12-inch diameter model vessels and 3-foot diameter full-size vessels were tested by cyclic pressurization after a comprehensive strain gage survey was made of the peak stresses. Fig. 12 shows a summary of the PVRC test results compared to the recommended design fatigue curve of Section III for carbon and low-alloy steel. It may be seen that no crack initiation was detected at any stress level below the allowable stress, and no crack progressed through a vessel wall in less than three times the allowable number of cycles. The large scatter of the data does indicate that further research on specific materials and further studies of nozzle stresses could eventually lead to less restrictive rules for some materials and some nozzle designs. Additional data are included in Reference [9].

IV. SPECIAL STRESS LIMITS

Paragraph N-417 of Section III and Paragraphs 4-136 through 4-138 of Appendix 4 and Paragraphs 5-130 and 5-140 of Appendix 5 of Division 2 of Section VIII contain special stress limits. These deviations from the basic stress limits are provided to cover special operating conditions or configurations. Some of these deviations are less restrictive and some more restrictive than the basic stress limits. In cases of conflict, the special stress limits take precedence for the particular situations to which they apply.

The common coverage of the two Codes includes:

(a) A modified Poisson's ratio value to be used when computing local thermal stresses.

(b) Provisions for waiving certain stress limits if a plastic analysis is performed and shakedown is demonstrated.

(c) Provisions for Limit Analyses as a substitute for meeting the prescribed basic limits on local membrane stresses and on primary membrane plus primary bending stresses.

(d) A limit on the sum of the three principal stresses.

(e) Special rules to be applied at the transition between a vessel nozzle and the attached piping.

(f) Requirements to prevent thermal stress ratchet growth of a shell subjected to thermal cycling in the presence of a static mechanical load.

(g) Requirements to prevent progressive distortion on non-integral connections.

In addition, Paragraphs N-417.1 and N-417.2 of Section III and Paragraphs AD-132.1 and AD-132.2 of Div. 2 provide rules for Bearing Loads and Pure Shear, respectively.

The first three of these special rules and the rules associated with item (f) provide recognition of the growing significance of plastic analysis to the evaluation of pressure components. The shakedown analysis provides a means whereby the limit on primary plus secondary stress limits may be exceeded. This particular limit is the one with which most difficulty has been experienced in vessels subject to severe thermal transients. Unfortunately, the slow progress in developing practical methods of shakedown analysis has made this provision difficult to apply, and alternate methods are under study.

The limit analysis provision is essential when evaluating formed heads of large diameter to thickness ratio. Such heads develop significant hoop compressive stresses and meridional tensile stresses in the knuckle regions over an area in excess of that permitted by the rules for classification as local membrane stresses. A limit analysis such as that by Drucker and Shield [11] is essential and has been used to develop Figure AD-204.1 of Division 2. These techniques represent an extension to more complex geometries of the principles applied to the development of Figure 2.

The problem of potential thermal ratchet growth has been described by Miller [12], and this paper provides the basis for the Code rules.

Since the "stress intensity" limit used in these Codes is based upon the maximum shear stress criterion, there is no limit on the "hydrostatic" component of the stress. Therefore, a special limit on the algebraic sum of the three principal stress is required for completeness.

V. CREEP AND STRESS-RUPTURE

It is an observed characteristic of pressure vessel materials that in service above a certain temperature, which varies with the alloy composition, the materials undergo a continuing deformation (creep) at a rate which is strongly influenced by both stress and temperature. In order to prevent excessive deformation and possible premature rupture it is necessary to limit the allowable stresses by additional criteria on creep-rate and stress-rupture. In this creep range of temperatures these criteria may limit the allowable stress to substantially lower values than those suggested by the usual factors on short time tensile and yield strengths. Satisfactory empirical limits for creep-rate and stress-rupture have been established and used in Section I and Section VIII, Div. 1.

Creep behavior complicates the detailed stress analysis because the distribution of stress will vary with time as well as with the applied loads. The difficulties are particularly noticeable under cyclic loading. It has not yet been possible to formulate complete design criteria and rules in the creep range, and the present application of Section III and Division 2 of Section VIII is restricted to temperatures at which creep will not be significant. This has been done by limiting the tabulated allowable stress intensities to below the temperature of creep behavior. The Subgroup on Elevated Temperature is studying this problem.

VI. SUMMARY

The design criteria of Section III and Division 2 of Section VIII differ from those of Section I and Division 1 of Section VIII in the following respects:

(a) Section III and Division 2 use the maximum shear stress (Tresca) theory of failure instead of the maximum stress theory

(b) Section III and the Appendices of Division 2 require the detailed calculation and classification of all stresses and the application of different stress limits to different classes of stress, whereas Section I and Division 1 of Section VIII give formulas for minimum allowable wall thickness.

(c) Section III and Division 2 require the calculation of thermal stresses and give allowable values for them, whereas Section I and Division 1 do not.

(d) Section III and Division 2 consider the possibility of fatigue failure and give rules for its prevention, whereas Section I and Division 1 do not.

The stress limits of Section III and Division 2 are intended to prevent three different types of failure, as follows:

(a) Bursting and gross distortion from a single application of pressure are prevented by the limits placed on primary stresses.

(b) Progressive distortion is prevented by the limits placed on primary-plus-secondary stresses. These limits assure shake-down to elastic action after a few repetitions of the loading.

(c) Fatigue failure is prevented by the limits placed on peak stresses.

The design criteria described here were developed by the joint efforts of the members of the Special Committee to Review the Code Stress Basis and its Task Groups over a period of several years. It is not to be expected that this paper will answer all the questions which will be asked, but it is hoped that it will give sufficient background to justify the rules which have been given.