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NUCLEAR METALLURGY LECTURES
CHAPTER 17

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NUCLEAR METALLURGY LECTURES

Chapter 17

Process Tubes

A. Philosophy of Factors of Safety

1. States of Stress

In uni-axial loading, yielding or permanent set will occur at some value of stress, which has been defined as the yield stress corresponding to an arbitrary strain. For ductile materials this value of strain has been set at 0.2 percent (see Figure 1). The designers use this value of yield stress in order to specify the load which a structure may support or, in case of a tube, to specify the wall thickness for a given pressure, or the pressure for a given wall thickness.

The yield stress for uni-axial load can easily be determined by a tensile or a compressive stress. Hence, a designer can adopt a factor of safety and determine the allowable or working stress. However, for combined stresses, the yield stress as determined from a tensile test does not necessarily correspond to yielding under these conditions (Figure 2). There are five theories for yielding under combined stresses and these are discussed below.

2. Theories for Yielding¹

a. Maximum Stress Theory (Rankine)

This theory is based on the assumption that yielding of the element in Figure 2 is dependent only on the largest of the two (or three) principal stresses; that is, yielding will occur when either σ_1 or σ_2 equals the yield stress determined from a simple tensile or compression test. This theory can be expressed as follows:

$$\sigma_1 = \pm \sigma_y, \sigma_2 = \pm \sigma_y, \sigma_1/\sigma_y = \pm 1, \sigma_2/\sigma_y = \pm 1.$$

This theory is illustrated graphically in Figure 3-a.

b. Maximum Shear Theory (Coulomb)

This theory assumes that yielding for combined stresses will occur when the maximum resulting shear stress equals the yield shear stress in simple tension. That is:

$$\tau_r = \tau_y = \frac{\sigma_y}{2}, \quad \text{where } \tau_r = \text{resulting shear stress,} \\ \tau_y = \text{yield shear stress,} \\ \sigma_y = \text{yield tensile stress.}$$

$$\text{Then, from the above equation and } \tau = \frac{\sigma_1 - \sigma_2}{2}, \quad \sigma_1 - \sigma_2 = \pm \sigma_y$$

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If the two-dimensional stresses are of opposite sign, the maximum shear stress is:

$$\tau = \frac{\sigma_1 - \sigma_2}{2}, \quad \sigma_1 - \sigma_2 = \pm \sqrt{y} \quad (\text{for yielding})$$

For stresses of the same sign, then

$$\sigma_1 = \pm \sqrt{y} \quad \text{or} \quad \sigma_2 = \pm \sqrt{y}, \quad \text{whichever is larger.}$$

This theory is illustrated graphically in Figure 3-b.

c. Maximum Strain Theory (St. Venant)

This theory assumes that yielding will occur when the maximum of the strains under bi-axial loading equals the strain at yield in simple tension, that is $\epsilon = \sigma_y/E$. The equations are:

$$\begin{aligned} \sigma_1 > \sigma_2: \quad \sigma_1 - \mu \sigma_2 &= \pm \sigma_y \\ \sigma_2 > \sigma_1: \quad \sigma_2 - \mu \sigma_1 &= \pm \sigma_y \end{aligned}$$

This theory is illustrated graphically in Figure 3-c.

d. Maximum Strain Energy Theory

This theory assumes that yielding will occur when the total strain energy of deformation under combined stresses exceeds the strain energy in simple tension, both for unit volume. The strain energy in simple tension is:

$$u = \sigma_y^2 / 2E$$

u = strain energy
 σ_y = stress at yield
 E = Young's Modulus

For a two-dimensional stress system, the strain energy is given by:

$$u = \frac{1}{2E} (\sigma_1^2 - 2\mu \sigma_1 \sigma_2 + \sigma_2^2)$$

e. Distortion energy theory (von Mises - Hencky)

This theory assumes that yielding will occur when the distortion energy produced by combined stresses, equals the distortion energy in simple tension at yield. (The strain energy discussed in (d) is made up of the energy required to change the volume of the element and the energy required to produce distortion).

For simple tension at yield, the distortion energy is:

$$u_d = \left(\frac{1+\mu}{3E} \right) \sigma_y^2$$

For bi-axial stress: $u_d = \left(\frac{1+\mu}{3E} \right) (\sigma_1^2 + \sigma_2^2 - \sigma_1 \sigma_2)$

Therefore, for yielding under combined stresses:

$$\sigma_1^2 - \sigma_1 \sigma_2 + \sigma_2^2 = \sigma_y^2$$

This theory is illustrated in Figure 3-e.

The areas bounded by the stress ratios σ_1/σ_y and σ_2/σ_y represent elastic behavior of the material; that is, in these regions, strains reverse to zero upon removal of stresses. Outside of these regions, the material is in the plastic range.

The maximum stress theory does not agree with test results; however, it is used automatically when designers do not take into account the influence of combined stresses (Figure 3-a).

The maximum shear theory is used for ductile materials and agrees fairly well with test results. If the two stresses σ_1 and σ_2 are alike in sign (both tension or both compression), the maximum stress and maximum shear stress theories result in the same criteria for yielding. (see Figure 3a and 3b). If they are unlike in sign, then the two theories differ considerably. This theory has been adopted by ASME and some government agencies for various design systems, due to the fact that test results are on the conservative side.

Of the five theories discussed above, the distortion energy theory (Figure 3-e) approximates test results most accurately.

Cadwell² applied these theories to process tubes subjected to various stress states. He pointed out that if a tube is bent, a compressive stress is introduced, in addition to the circumferential tensile stress; hence, by the distortion energy theory, $\sigma_1 = 0.65 \sigma_y$, (see Figure 3-e), and yielding will occur at only 65 percent of the yield strength. This, of course, is an undesirable condition.

In order to check this theory, Taylor and Pedersen³, bent short cylinders and subjected these to hydrostatic pressures. They found that the bursting pressure remained constant, essentially, irrespective of magnitude of plastic deformation in bending. Their experiments, of course, do not prove or disprove these theories, since bursting pressure, rather than yield strength, was used as a measure of failure. However, from practical point of view, they found that all failures occurred at the thinnest part of the cylinder. Hence, irrespective of the theories, the designer must allow a measure of safety to account for variables in manufacture, etc.

3. Factor of Safety

Marin¹ has made a critical analysis of the variables which should be considered in the selection of the factor of safety. Such items as fabrication, homogeneity of the material, stress analysis, material properties and many others must be integrated into a factor of safety which will optimize the working or design stress. For static structures, the simple "beefing up" of a beam or a column, except for the extra cost of the material, can be done. However, for mobile equipment, increase in weight of the system detracts from its pay load. For reactor design, two other factors must be considered: thermal stress and neutron cross-section; increasing the cross-section to compensate for unknown factors in design leads to larger thermal gradients, where parts operate at elevated temperatures, and to larger losses in neutrons. Also, unfortunately, materials with low neutron cross sections are the most expensive; hence, to use more material than absolutely necessary becomes costly, as we shall see later.

Most accepted codes for design of engineering systems are conservative, as the data in Table I illustrates. For our applications, a re-evaluation of such codes is necessary.

8. Mechanical Properties of Available Materials

1. Effect of Temperature

Recently, Wallace⁴ summarized the elevated temperature mechanical properties of aluminum and its alloys, zirconium, stainless steels, and other materials. These data are reproduced in Figures 5 - 8. From strengths and reactivity considerations aluminum alloys are the most economical to use up to temperatures 150 - 200 C; above these temperatures, zircaloy 2, in the cold worked condition, is the most economical material.

The pressure induced stresses for three pressures and tube sizes and the yield strengths for three materials at 315 C are illustrated graphically in Figure 5; the independent variable in this figure is wall thickness. A range of safety has been calculated for several cases; for example, the yield strength of cold worked zircaloy 2 is 60,000 psi at 315 C; for a KER tube of 2.40" O.D. with 0.120" wall thickness, the pressure induced stress is 20,000 psi, hence, the range of safety is 60,000 - 20,000 = 40,000 psi.

The tensile and yield strengths of various materials at several temperatures are illustrated in Figures 6 and 7. It will be noted that stainless steels, particularly the precipitation hardenable grades, have the highest tensile and yield strengths.

2. Effect of Time at Temperature

The data on the effect of time at temperature, that is, creep and stress rupture data, are available for aluminum and its alloys in the annealed and strain and precipitation hardened conditions. On stainless steels, data of this type are limited and on zirconium, they are non-existent. Hence, it is necessary to infer the effect of time on zircaloy 2 in the cold worked condition from available data on aluminum and steel. In the case of aluminum alloys, the creep strengths up to 150 C are raised by strain hardening, and up to 200 C, by precipitation hardening; however, above these temperatures, these effects disappear.

C. Costs

Figure 8 presents data on costs of zircaloy 2 tubing. The base material costs are approximately as illustrated. The fabrication costs can be reduced substantially as more fabrication experience is acquired.

WP Wallace
Pile Technology
ENGINEERING DEPARTMENT

WP Wallace/bfm

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1. Marin, J., Engineering Materials, Prentice-Hall, Inc. N.Y., 1952.
2. Cadwell, J. J., "Process Tube Materials - Stress Factors", HW-23581, February 21, 1952, Secret.
3. Taylor, A. T., and Pedersen, L.M., "Failures of Stressed Cylinders", HW-35717, Secret.
4. Wallace, W. P., "Process Tube Materials for High Temperature Reactor Service", HW-34344, January 7, 1955, Secret.

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TABLE 1
MECHANICAL PROPERTIES AND ALLOWABLE STRESSES

Type of Material	Temp. F	0.2% Yield Strength σ_y (X1000 psi)	Tensile Strength σ_T (X1000 psi)	Creep Strength σ_c (X1000 psi)	Seamless σ_{A1} (X1000 psi)	Fusible Welded σ_{A2} (X1000 psi)	Resistance Welded σ_{A3} (X1000 psi)	$\frac{\sigma_{A1}}{\sigma_T}$	$\frac{\sigma_{A1}}{\sigma_y}$	$\frac{\sigma_{A1}}{\sigma_c}$
Low Carbon Steel	70	39.0	69.0		15.0	12.4	12.7	0.25	0.39	
	300	36.5	67.0		14.1	11.6	12.0	0.21	0.39	
	500	32.0	67.5		12.9	10.6	11.0	0.19	0.40	
	600	29.5	64.0	28.0	12.3	10.2	10.4	0.19	0.42	0.44
1/2% Mn Steel	70	40.0	64.0		11.0			0.17	0.27	
	300	36.0	70.0		11.0			0.17	0.30	
	500	31.5	72.5		11.0			0.15	0.35	
	600	30.0	71.5	42.0	11.0			0.15	0.37	0.26
5% Cr & 1/2 Mn Steel	70	31.0	71.0		12.0			0.17	0.33	
	300	27.0	65.0		12.0			0.18	0.44	
	500	24.0	61.0		12.0			0.20	0.50	
	600	17.5	68.0	30.0	12.0			0.20	0.52	0.40
Type 304 (18-8) Stainless Steel	70	34.0	85.0							
	300	23.0	73.0							
	500	19.0	69.0							
	600	17.5	68.0	42.0						
Type 347 Stainless Steels	70	39.5	91.0		15.0		12.7	0.16	0.38	
	300	34.0	74.5		15.0		12.7	0.20	0.44	
	500	32.0	69.0		15.0		12.7	0.20	0.44	
	600	32.0	68.0	46.0	15.0		12.7	0.22	0.47	0.33

Allowable stresses (σ_{A1} , σ_{A2} , and σ_{A3}) taken from Tables 3 and 3a, Code for Pressure Piping, ASME, 1951.

σ_y , σ_T , and σ_c values taken from Steels for Elevated Temperature Service, U. S. Steel Corp.

σ_c (creep) = stress for creep rate of 0.0001% per hour, extrapolated to 600 F from 800 - 1000F range.

THE CONVENTIONAL YS AND 0.7 SLOPE YS (σ_2) ARE EQUAL.

FIG. 1 - STRESS-STRAIN RELATIONSHIPS

FIG. 2 - STRESS STATES

FIG. 3 - THEORIES FOR YIELDING

- A - MAXIMUM STRESS THEORY
- B - MAXIMUM SHEAR THEORY
- C - MAXIMUM STRAIN THEORY
- D - MAXIMUM STRAIN ENERGY THEORY
- E - DISTORTION ENERGY THEORY

FIGURE 1

THEORIES FOR YIELDING - COMBINED

SECRET

FIGURE 5

VARIATION OF PRESSURE - INDUCED STRESS WITH WALL
THICKNESS FOR THREE TUBE SIZES AND PRESSURES

FIGURE 6

SHORT TIME TENSILE STRENGTHS OF SEVERAL METALS
IN ANNEALED, HOT ROLLED AND COLD ROLLED CONDITIONS

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FIGURE 7

SHORT TIME YIELD STRENGTHS OF SEVERAL METALS IN
ANNEALED, NOT ROLLED & COLD ROLLED CONDITIONS

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FIGURE 8

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WEIGHT AND BASE MATERIAL COST PER FOOT OF LENGTH OF THREE
TUBE SIZES. (BASE COST ASSUMED AS \$26.00 PER LB. OF ZR-2)

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