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Chapter 2

2-1 From Tables A-20, A-21, A-22, and A-24c,
(a) UNS G10200 HR. $S_{ut} = 380$ (55) MPa (kpsi), $S_y = 210$ (30) MPa (kpsi) .Ans.
(b) SAE 1050 CD. $S_{ut} = 690$ (100) MPa (kpsi), $S_y = 580$ (84) MPa (kpsi) .Ans.
(c) AISI 1141 Q&T at 540C (1000F). $S_{ut} = 896$ (130) MPa (kpsi), $S_y = 765$ (111) MPa (kpsi) .Ans.
(d) 2024-T4. $S_{ut} = 446$ (64.8) MPa (kpsi), $S_y = 296$ (43.0) MPa (kpsi) .Ans.
(e) Ti-6Al-4V annealed. $S_{ut} = 900$ (130) MPa (kpsi), $S_y = 830$ (120) MPa (kpsi) .Ans.

2-2 (a) Maximize yield strength: Q&T at 425C (800F) .Ans.

(b) Maximize elongation: Q&T at 650C (1200F) .Ans.

2-3 Conversion of kN/m³ to kg/m³ multiply by $1(10^3) / 9.81 = 102$
AISI 1018 CD steel: Tables A-20 and A-5
$$\frac{S_y}{\gamma} = \frac{370 \cdot 10^3}{76.5 \cdot 102} = 47.4 \text{ kN·m/kg} \quad .Ans.$$
2011-T6 aluminum: Tables A-22 and A-5
$$\frac{S_y}{\gamma} = \frac{169 \cdot 10^3}{26.6 \cdot 102} = 62.3 \text{ kN·m/kg} \quad .Ans.$$
Ti-6Al-4V titanium: Tables A-24c and A-5
$$\frac{S_y}{\gamma} = \frac{830 \cdot 10^3}{43.4 \cdot 102} = 187 \text{ kN·m/kg} \quad .Ans.$$
ASTM No. 40 cast iron: Tables A-24a and A-5 Does not have a yield strength. Using the
ultimate strength in tension
$$\frac{S_{ut}}{\gamma} = \frac{42.5 \cdot 6.89 \cdot 10^3}{70.6 \cdot 102} = 40.7 \text{ kN·m/kg} \quad .Ans.$$

2-4 AISI 1018 CD steel: Table A-5
$$E \quad \frac{30.0 \cdot 10^6}{0} \qquad 282$$

design factor method, and sometimes called the classical method of design. The fundamental equation is Eq. (1¢ÃÂÂ1) where nd is called the design factor. All loss-offunction modes must be analyzed, and the mode leading to the smallest design factor governs. After the design is completed, the actual design factor may change as a result of changes such as rounding up to a standard size for a cross section or using off-theshelf components with higher ratings instead of employing what is calculated by using the design factor. The factor is then referred to as the factor of safety, n. The factor of safety has the same definition as the factor, but generally differs numerically. Since stress may not vary linearly with the load (see Section 3 äiation 19), the use of the load as a parameter of functional loss may not be acceptable. It is more common therefore to express the design factor in terms of stress and significant strength. So the EQ. (1 "1) can be rewritten as ND 5 resistance to the loss of function S 5 eligible stress s (or t) (1 peeled" 3) the terms of stress and resistance in the EQ. (1 "3) must be of the same type and unit. Furthermore, stress and strength must apply to the same critical position as the part. Example 1-2 solution a barrel with a transversal area of A and loaded in Tension with an axial force of P 5 2000 LBF suffers a stress of S 5 Pya. Using a resistance to the material of 24 kpsi and a 3.0 design factor, determine the minimum diameter of a solid circular rod. Using the table to â,¬ â € œ17, select a favorite fractional diameter and determines the safety factor of the auction. From a 5 Pd 2y4, S 5 Pya and from the EQ. (1â € âvelop â € œ3), s 5 Synd, then S5 P P S 5 5 ND A PD 2 Y4 Solve for D5A Reply 4PND 1Y2 4 (2000) 3 1y2 B 5A B 5 0.564 in PS P (24 000) from the table to aâ € âvelop â € œ17 , the next higher favorite size is 58 in 5 0.625 inches. Therefore, when ND is replaced with N in the equation developed above, the safety factor is the response N5 P (24 000) 0.6252 PSD 2 5 5 3.6 8 4P 4 (2000) Ringing thus the diameter increased the actual design factor. It is attractive to offer some recommendations regarding the assignment of the design factor for a certain application. 7 The problem in doing it is with evaluation 7 if the reader wants some examples of values of the design factor assignment of David G. Ullman, The mechanical design process, 4th ed., Mcgraw-Hill, New York, 2010, app. C. .itatlusir .itatlusir i onnarezneulfni ehc irottaf i ittut id aznairav al erageipis id eratnet eved atsittegorp li ehc ¨Å Atlaer aL .enoiznuf id atidrep id Atiladom ella etaicossa ezzetrecni esoremun elled 91 acinaccem airengegni id enoizattegorp alla The designer must rely on the experience, business policies and the numerous codes that may concern the application (e.g. ASME boiler and pressure vessel code) to arrive at an appropriate design factor. An example could help clarify the complexity of a design factor assignment. Example 1â € "3 A vertical round rod must be used to support a suspended weight. A person will position the weight at the end without dropping it. The diameter of the rod can be produced within 61 percent of its nominal size. The support ends can be centered within 61.5 percent of the nominal diameter size. Weight is known within 62 percent of nominal weight. The strength of the material is known within 63.5 percent of the nominal resistance value. If the designer uses nominal values and nominal stress equation, SNOM 5 PYA (as in the previous example), determines which design factor should be used so that stress does not exceed strength. Solution There are two hidden factors to consider here. The first, due to the possibility of eccentric loading, the maximum stress is not S 5 PYA (revision Cap. 3). Secondly, the person may not gradually position the weight on the end of the rod support and the load application would therefore be considered dynamic. Consider eccentricity first. With eccentricity, there will be a bending moment giving an additional stress of S 5 32 MY (PD 3) (see Sez. 3â€ "10". The bending moment is given by M 5 PE, where E is eccentricity. Therefore, the maximum solicitation in the rod is given by S5 32PE P 32PE P 1 3 5 2 A PD Pd 3 Pd Y4 (1) Since the tolerance to eccentricity is expressed according to the diameter, we will write eccentricity as a percentage of d. Let E 5 Ked, where Ke is a constant. 1 1 1[2 monS661.1 5]I10,0 2 1 (D(d(D(D(D(D(D(D(D(D(D(D(D(D(D(D(D(D(D(D(D(D[P)20,0 1 1(p4 5 xamS Äd eregmuiggar ²Åup sserts ol ehc omissam li erenetto rep eznarellot el odnacilppA)2(2 dP 2 dP 3 dP)ek8 1 1(5 1 p4 p4 dekp23 5S emoc ottircsir eneiv)1(.qE 5 1.166 a 4P b pd 2 (3) Suddenly applied loading is covered in Sec. 4¢ÃÂÂ17. If a weight is dropped from a height, h, from the support end, the maximum load in the rod is given by Eq. (4¢ÃÂÂ59) which is F 5 W 1 W a1 1 hk 1y2 b W where F is the force in the rod, W is the weight, and k is the rod¢ÃÂÂs spring constant. Since the person is not dropping the weight, h 5 0, and with W 5 P, then F 5 2P. This assumes the person is not gradually placing the weight on, and there is no 20 Mechanical Engineering Design damping in the rod. Thus Eq. (3) is modified by substituting 2P for P and the maximum stress is smax 5 2(1.166) snom 5 2.332 snom The minimum strength is Smin 5 (1 2 0.035) Snom 5 0.965 Snom Equating the maximum stress to the minimum strength gives 2.332 snom 5 0.965 Snom From Eq. (1¢ÃÂÂ3), the design factor using nominal values should be Answer nd 5 Snom 2.332 5 5 2.42 snom 0.965 Obviously, if the designer takes into account all of the uncertainties in this example and accounts for all of the tolerances in the stress and strength in the calculations, a design factor of one would suffice. In practice, the designer would probably use the nominal geometric and strength values with the simple s 5 PyA calculation. The designer would probably not go through the calculations given in the example and would assign a design factor. This is where the experience factor comes in. The designer should make a list of the loss-of-function modes and estimate a factor, ni, for each. For this example, the list would be Loss-of-function Estimated accuracy ni Geometry dimensions Good tolerances 1.05 Dynamic load Not gradual loading 2.0* Bending Slight possibility 1.1 Well known 1.05 Stress calculation Strength data *Minimum Each term directly affects the results. Therefore, for an estimate, we evaluate the product of each term n d 5 q ni 5 1.05(2.0)(1.1)(1.05) 5 2.43 1¢ÃÂÂ12 Reliability and Probability of Failure In these days of Increase the number of causes of responsibility and the need to conform to the regulations issued by government agencies such as EPA and Osha, it is very important for the designer and the producer to know the reliability of their product. The method of reliability of the design is one in which we obtain the distribution of stresses and the distribution of strengths and then we recommend them to reach an acceptable success rate. The statistical measure of the probability that a mechanical element does not fail in use is called the reliability of this element and as we will see, it is related to the probability of bankruptcy, pf. Introduction to the mechanical design figure 1-4 f (x) 21 f (x) the form of normal curves; Distribution curve: (a) small s the (b) large s x m x m (a) (b) figure 1-5 function of normal distribution transformed of table a-10. f (z) f (zâ £) â £ 0 z pro z probability of bankruptcy The probability of bankruptcy, PF, is obtained from the function of density of probability (pdf), which represents the distribution of events within A given range of values. A series of discrete and continuous standard probability distributions are commonly applicable to engineering problems. The two more important distributions of continuous probability for our use in this text are the Gaussian distribution (normal) and the distribution of Weibull. We will describe the normal distribution in this section and in sec. 2-2. Weibull's distribution is widely used in the design of laminated contact bearings and will be described in chap. 11. The Gaussian distribution continues (normal) is important whose function of probability density (pdf) is expressed in terms of average, MX, and its standard deviation8 S, x 1 x 2 mx 2 f (x) 5 Exp C 2 a b d (1â € "4) 2 s, x 22p the lots of eq. (1â € "4) are shown in figure 1â € " 4 for small and large standard deviations. The curve a everb everb ¹Áip e fo s id irolav iloccip rep atterts ¹Áip e atla ¹Áip ¨Å anapmac id 0 5.93Z 1.1 "€ Å Å Å Å Å itali Å Å ¢ z) Z (F 6â" € Å ¢ 1 Erugif .ispk 5.93 Naht Ssel HTGNERTS A GNIVAH NO NOALTALUPOP EHT FO NOITROPORP EHT STNESERPER) SDOR 43

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