Gears and Gearing
Stress and Selection

Chapter Outline

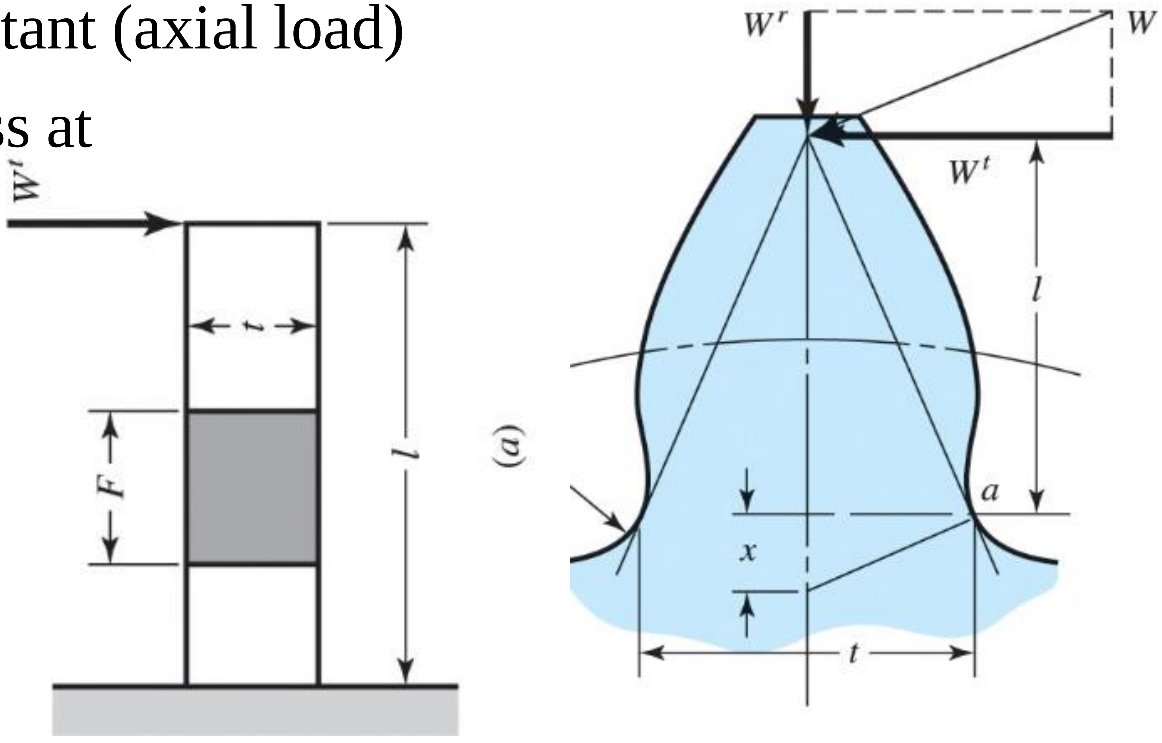
Stresses in Spur Gears

Lewis Equation for estimating bending stresses in gear teeth

- Consider worst case
- Contact Force (W^t) at top of tooth yields bending moment
- Similar to a cantilever beam
- W^r force not as important (axial load)
- Highest bending stress at root, pt. a

$$\sigma = \frac{M}{I/c} = \frac{6W^t l}{Ft^2}$$

F is the face width,
and t is the tooth width



(a) (b)

Cantilever Beam Model of Bending Stress in Gear Tooth

$$\sigma = \frac{M}{I/c} = \frac{6W^t l}{Ft^2}$$

Using geometry, similar triangles

$$\frac{t/2}{x} = \frac{l}{t/2} \quad or \quad x = \frac{t^2}{4l} \quad or \quad l = \frac{t^2}{4x}$$

Rearranging terms

$$\sigma = \frac{6W^t l}{Ft^2} = \frac{W^t}{F}\frac{1}{t^2/6l} = \frac{W^t}{F}\frac{1}{t^2/4l}\frac{1}{\frac{4}{6}}$$

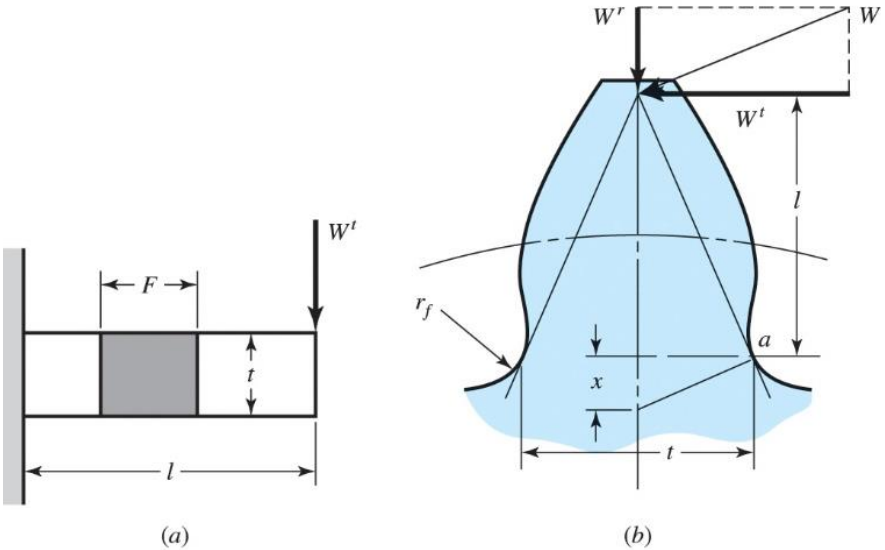


Fig. 14–1

Developing the Lewis Equation for estimating bending stresses in gear teeth, continued

$$\sigma = \frac{W^t p}{F\left(\frac{2}{3}\right) xp}$$

$y = 2x/(3p)$ y is called the Lewis form factor

$$\sigma = \frac{W^t}{Fpy} \qquad \textbf{(14-1)}$$

Lewis Equation[1]

$$\sigma = \frac{W^t}{Fpy} \qquad \textbf{(14-1)}$$

$$p = \pi / P \qquad y = Y / \pi$$

Lewis Equation $$\sigma = \frac{W^t P}{FY} \qquad \textbf{(14-2)}$$

Lewis Form Factor $$Y = \frac{2xP}{3} \qquad \textbf{(14-3)}$$

Values of Lewis Form Factor Y

Number of Teeth	*Y*	Number of Teeth	*Y*
12	0.245	28	0.353
13	0.261	30	0.359
14	0.277	34	0.371
15	0.290	38	0.384
16	0.296	43	0.397
17	0.303	50	0.409
18	0.309	60	0.422
19	0.314	75	0.435
20	0.322	100	0.447
21	0.328	150	0.460
22	0.331	300	0.472
24	0.337	400	0.480
26	0.346	Rack	0.485

Table 14–2

Dynamic Effects[1]

Effective load increases as velocity increases.

Velocity factor K_v accounts for this.

With pitch-line velocity V in feet per minute,

$$K_v = \frac{600 + V}{600} \quad \text{(cast iron, cast profile)} \qquad \textbf{(14-4}\boldsymbol{a}\textbf{)}$$

$$K_v = \frac{1200 + V}{1200} \quad \text{(cut or milled profile)} \qquad \textbf{(14-4}\boldsymbol{b}\textbf{)}$$

$$K_v = \frac{50 + \sqrt{V}}{50} \quad \text{(hobbed or shaped profile)} \qquad \textbf{(14-5}\boldsymbol{a}\textbf{)}$$

$$K_v = \sqrt{\frac{78 + \sqrt{V}}{78}} \quad \text{(shaved or ground profile)} \qquad \textbf{(14-5}\boldsymbol{b}\textbf{)}$$

Dynamic Effects 2

With pitch-line velocity *V* in meters per second,

$$K_v = \frac{3.05 + V}{3.05} \qquad \text{(cast iron, cast profile)} \qquad \mathbf{(14\text{-}6}\boldsymbol{a}\mathbf{)}$$

$$K_v = \frac{6.1 + V}{6.1} \qquad \text{(cut or milled profile)} \qquad \mathbf{(14\text{-}6}\boldsymbol{b}\mathbf{)}$$

$$K_v = \frac{3.56 + \sqrt{V}}{3.56} \qquad \text{(hobbed or shaped profile)} \qquad \mathbf{(14\text{-}6}\boldsymbol{c}\mathbf{)}$$

$$K_v = \sqrt{\frac{5.56 + \sqrt{V}}{5.56}} \qquad \text{(shaved or ground profile)} \qquad \mathbf{(14\text{-}6}\boldsymbol{d}\mathbf{)}$$

Lewis Equation 2

The Lewis equation including velocity factor.

- U.S. Customary version.

$$\sigma = \frac{K_v W^t P}{FY} \qquad \textbf{(14-7)}$$

- Metric version.

$$\sigma = \frac{K_v W^t}{FmY} \qquad \textbf{(14-8)}$$

Acceptable for general estimation of stresses in gear teeth.

Forms basis for AGMA method, which is preferred approach.

Generally Selecting Gears, Follow the Manufacturer's Guide

Let's look at the Boston Gear Catalog:

https://www.bostongear.com/products/open-gearing/stock-gears/spur-gears/spur-gears

In particular the Rotary Drive Products catalog from Boston Gear

https://www.bostongear.com/-/media/Files/Literature/Brand/boston-gear/catalogs/p-1930-bg.ashx

2 Important Sections:

1) Standard products & Selection Guide
2) Engineering data

Spur Gears Standard Product Data: pg 17-62

- All the different gears (materials, P, dimensions, etc.)
- Selection procedure & hints
- Horsepower ratings

Gear Selection Procedure

Selection Procedure

1. Determine service factor.
   a. Using application Classification Chart, pages 331-332, determine service factor or
   b. With knowledge of operating conditions and load classification, select service factor from Table 1 below.

Design HP = Application Load X Service Factor (Table 1)

3. Select spur gear pinion with horsepower capacity equal to (or greater than) design horsepower determined in Step 2. 14½° Pressure Angle Spur Gears—Page 50 to Page 57. 20° Pressure Angle Spur Gears— Page 58 to Page 62.
4. Select a driven spur gear with a catalog rating equal to or greater than the horsepower determined in Step 2. All ratings are predicated on gears properly lubricated and maintained.

Selection Hints

A. Select pinion having pitch diameter at least twice the shaft diameter.

B. Pinion number of teeth should be greater than 16 for 14½°PA and 13 for 20°PA to avoid excessive under-cutting.

C. For tooth numbers or RPMs not on Chart, interpolation of horsepower is adequate.

D. Pitchline velocities above 1000 FPM are not recommended for metallic spur gears. The Selection Chart reflects this in the lack of ratings for larger numbers of teeth at higher RPM's. Ratings to the right of heavy line are not recommended, as suggested maximum velocity is exceeded, and should be used for interpolation purposes only.

TABLE 1

Service Factor	Operating Conditions
.8	Uniform — not more than 15 minutes in 2 hours.
1.0	Moderate Shock — not more than 15 minutes in 2 hours. Uniform — not more than 10 hours per day.
1.25	Moderate Shock — not more than 10 hours per day. Uniform — more than 10 hours per day.
1.50	Heavy Shock — not more than 15 minutes in 2 hours. Moderate Shock — more than 10 hours per day.
1.75	Heavy Shock — not more than 10 hours per day.
2.0	Heavy Shock — more than 10 hours per day.

Heavy shock loads and/or severe wear conditions may require the use of higher service factors. Consultation with factory is recommended in these applications.

Horsepower ratings

Spur Gears

Approximate Horsepower and Torque* Ratings

For Class I Service (Service Factor = 1.0)

A

5 DIAMETRAL PITCH STEEL **20° PRESSURE ANGLE** **2-1/2" FACE** **REFERENCE PAGE 46.**

No. Teeth	25 RPM		50 RPM		100 RPM		200 RPM		300 RPM		600 RPM		900 RPM		1200 RPM		1800 RPM		3600 RPM	
	H.P.	Torque	H.P.	Torque	H.P.	Torque	H.P.	Torque	H.P.	Torque	H.P.	Torque	H.P.	Torque	H.P.	Torque	H.P.	Torque	H.P.	Torque
12	1.14	2865	2.22	2794	4.22	2662	7.71	2431	10.65	2237	17.19	1805	21.61	1513	24.80	1302	29.09	1019		
14	1.49	3756	2.89	3647	5.47	3449	9.87	3110	13.48	2832	21.25	2233	26.31	1843	29.87	1569	34.53	1209		
15	1.67	4198	3.23	4069	6.08	3833	10.90	3435	14.81	3112	23.11	2427	28.41	1990	32.09	1686	36.87	1291		
16	1.81	4565	3.50	4416	6.58	4146	11.72	3693	15.85	3329	24.47	2570	29.89	2093	33.61	1765				
18	2.12	5332	4.08	5138	7.60	4789	13.38	4216	17.92	3766	27.15	2852	32.77	2295	36.56	1920				
20	2.44	6141	4.68	5894	8.66	5456	15.07	4750	20.02	4205	29.79	3129	35.58	2492	39.41	2070				

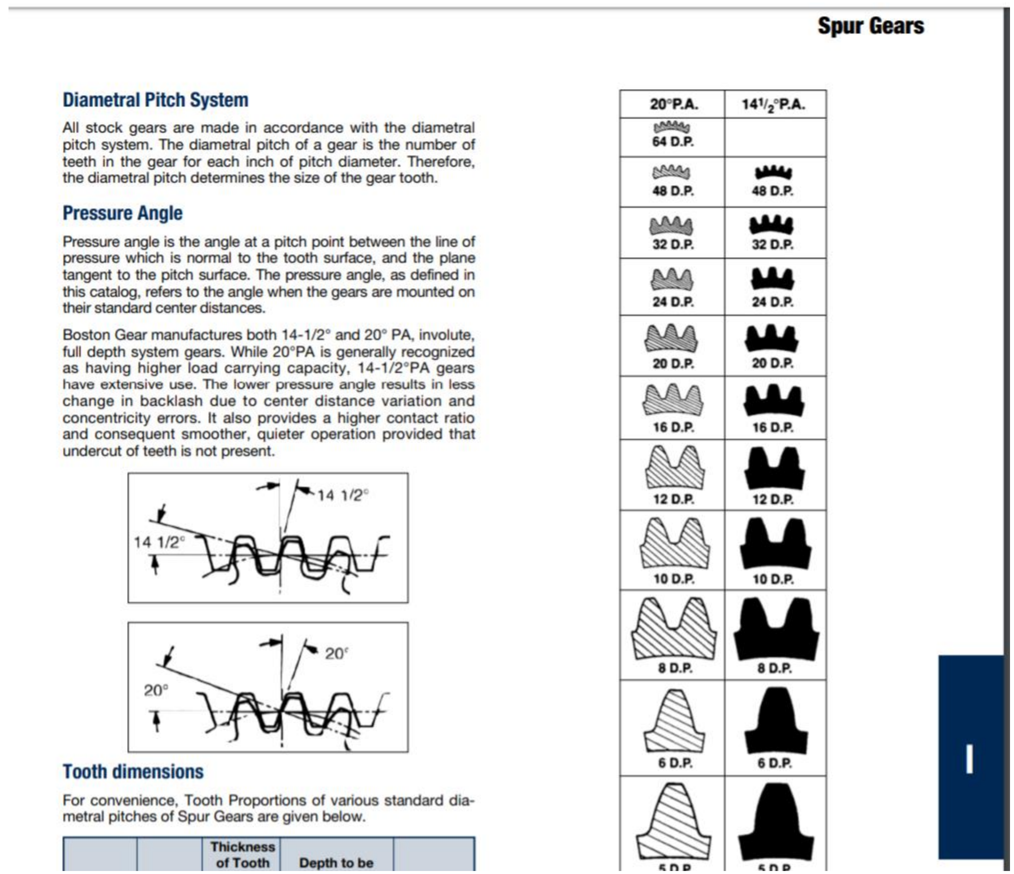
5 DIAMETRAL PITCH CAST IRON **20° PRESSURE ANGLE** **2-1/2" FACE** **REFERENCE PAGE 46.**

No. Teeth	25 RPM		50 RPM		100 RPM		200 RPM		300 RPM		600 RPM		900 RPM		1200 RPM		1800 RPM		3600 RPM	
	H.P.	Torque	H.P.	Torque	H.P.	Torque	H.P.	Torque	H.P.	Torque	H.P.	Torque	H.P.	Torque	H.P.	Torque	H.P.	Torque	H.P.	Torque
24	1.82	4599	3.48	4381	6.35	4002	10.82	3411	14.15	2972	20.41	2144	23.95	1677						
25	1.91	4826	3.64	4588	6.63	4177	11.24	3542	14.64	3075	20.97	2203	24.51	1716						
28	2.21	5571	4.18	5267	7.54	4750	12.60	3970	16.23	3410	22.81	2396	26.37	1847						
30	2.40	6050	4.52	5700	8.10	5108	13.42	4230	17.18	3609	23.86	2506	27.41	1920						
35	2.97	7477	5.54	6982	9.78	6164	15.85	4995	19.98	4199	27.04	2840								
40	3.47	8737	6.42	8087	11.17	7040	17.75	5593	22.08	4639	29.21	3068								
45	3.98	10040	7.31	9216	12.56	7916	19.59	6174	24.09	5060	31.26	3284								
50	4.38	11046	7.98	10056	13.53	8528	20.75	6540	25.25	5304	32.22	3384								
60	5.32	13399	9.53	12008	15.78	9944	23.48	7400	28.05	5892	34.81	3657								
70	6.27	15794	11.06	13945	17.93	11300	26.00	8192	30.58	6425										
80	7.23	18229	12.59	15869	20.00	12605	28.34	8932	32.92	6916										
100	8.71	21969	14.78	18630	22.67	14288	30.92	9745												
110	9.68	24409	16.22	20449	24.50	15439	32.88	10362												
120	10.38	26168	17.19	21669	25.58	16125	33.85	10666												
140	11.70	29508	18.97	23910	27.50	17334	35.49	11182												
160	13.30	33526	21.13	26631	29.94	18870	37.83	11921												
180	14.49	36534	22.61	28495	31.40	19787	38.97	12281												

Ratings are based on strength calculation. Basic static strength rating, or for hand operation of above gears is approximately 3 times the 100 RPM rating.

Spur Gears: Engineering Data, Section I, pg 306-

Basics of Gears (involute profile, pressure angles, diametral pitch,…)



Spur Gears

Diametral Pitch System

All stock gears are made in accordance with the diametral pitch system. The diametral pitch of a gear is the number of teeth in the gear for each inch of pitch diameter. Therefore, the diametral pitch determines the size of the gear tooth.

Pressure Angle

Pressure angle is the angle at a pitch point between the line of pressure which is normal to the tooth surface, and the plane tangent to the pitch surface. The pressure angle, as defined in this catalog, refers to the angle when the gears are mounted on their standard center distances.

Boston Gear manufactures both 14-1/2° and 20° PA, involute, full depth system gears. While 20°PA is generally recognized as having higher load carrying capacity, 14-1/2°PA gears have extensive use. The lower pressure angle results in less change in backlash due to center distance variation and concentricity errors. It also provides a higher contact ratio and consequent smoother, quieter operation provided that undercut of teeth is not present.

Tooth dimensions

For convenience, Tooth Proportions of various standard diametral pitches of Spur Gears are given below.

		Thickness of Tooth	Depth to be	

20°P.A.	14½°P.A.
64 D.P.	
48 D.P.	48 D.P.
32 D.P.	32 D.P.
24 D.P.	24 D.P.
20 D.P.	20 D.P.
16 D.P.	16 D.P.
12 D.P.	12 D.P.
10 D.P.	10 D.P.
8 D.P.	8 D.P.
6 D.P.	6 D.P.
5 D.P.	5 D.P.

I

Spur Gears: Engineering Data, Section I, pg 306-

Formulas

Engineering Information

Spur Gears

Backlash

Stock spur gears are cut to operate at standard center distances. The standard center distance being defined by:

$$\text{Standard Center Distance} = \frac{\text{Pinion PD} + \text{Gear PD}}{2}$$

When mounted at this center distance, stock spur gears will

Diametral Pitch	Backlash (Inches)	Diametral Pitch	Backlash (Inches)
3	.013	8-9	.005
4	.010	10-13	.004
5	.008	14-32	.003
6	.007	33-64	.0025
7	.006		

have the following average backlash:

An increase or decrease in center distance will cause an increase or decrease in backlash.

Since, in practice, some deviation from the theoretical standard center distance is inevitable and will alter the backlash, such deviation should be as small as possible. For most applications, it would be acceptable to limit the deviation to an increase over the nominal center distance of one half the average backlash. Varying the center distance may afford a practical means of varying the backlash to a limited extent.

The approximate relationship between center distance and backlash change of 14-1/2° and 20° pressure angle gears is shown below:

For 14-1/2°–Change in Center Distance = 1.933 x Change in Backlash
For 20° –Change in Center Distance = 1.374 x Change in Backlash

From this, it is apparent that a given change in center distance, 14-1/2° gears will have a smaller change in backlash than 20° gears. This fact should be considered in cases where backlash is critical.

Undercut

When the number of teeth in a gear is small, the tip of the mating gear tooth may interfere with the lower portion of the tooth profile. To prevent this, the generating process removes material at this point. This results in loss of a portion of the

Spur Gear Formulas

FOR FULL DEPTH INVOLUTE TEETH

To Obtain	Having	Formula
Diametral Pitch (P)	Circular Pitch (p)	$P = \frac{3.1416}{p}$
	Number of Teeth (N) & Pitch Diameter (D)	$P = \frac{N}{D}$
	Number of Teeth (N) & Outside Diameter (D_o)	$P = \frac{N+2}{D_o}$ (Approx.)
Circular Pitch (p)	Diametral Pitch (P)	$p = \frac{3.1416}{P}$
Pitch Diameter (D)	Number of Teeth (N) & Diametral Pitch (P)	$D = \frac{N}{P}$
	Outside Diameter (D_o) & Diametral Pitch (P)	$D = D_o - \frac{2}{P}$
Base Diameter (D_b)	Pitch Diameter (D) and Pressure Angle (ø)	$Db = D\cos ø$
Number of Teeth (N)	Diametral Pitch (P) & Pitch Diameter (D)	$N = P \times D$
Tooth Thickness (t) @Pitch Diameter (D)	Diametral Pitch (P)	$t = \frac{1.5708}{P}$
Addendum (a)	Diametral Pitch (P)	$a = \frac{1}{P}$
Outside Diameter (D_o)	Pitch Diameter (D) & Addendum (a)	$D_o = D + 2a$
Whole Depth (h_t) (20P & Finer)	Diametral Pitch (P)	$h_t = \frac{2.2}{P} + .002$
Whole Depth (h_t) (Courser than 20P)	Diametral Pitch (P)	$h_t = \frac{2.157}{P}$
Working Depth (h_k)	Addendum (a)	$h_k = 2(a)$
Clearance (c)	Whole Depth (h_t) Addendum (a)	$c = h_t - 2a$
Dedendum (b)	Whole Depth (h_t) & Addendum (a)	$b = h_t - a$
Contact Ratio (M_C)	Outside Radii, Base Radii, Center Distance and Pressure Angle+C.P.	

$$M_C = \frac{\sqrt{R_o^2 - R_b^2} + \sqrt{r_o^2 - r_b^2} - C\sin ø^*}{p \cos ø}$$

Pitch Diameter (D)

Spur Gears: Engineering Data
Lewis Formula

Good for determining

F, for your application

Engineering Information

Spur Gears

Lewis Formula (Barth Revision)

Gear failure can occur due to tooth breakage (tooth stress) or surface failure (surface durability) as a result of fatigue and wear. Strength is determined in terms of tooth-beam stresses for static and dynamic conditions, following well established formula and procedures. Satisfactory results may be obtained by the use of Barth's Revision to the Lewis Formula, which considers beam strength but not wear. The formula is satisfactory for commercial gears at Pitch Circle velocities of up to 1500 FPM. It is this formula that is the basis for all Boston Spur Gear ratings.

METALLIC SPUR GEARS

$$W = \frac{SFY}{P}\left(\frac{600}{600 + V}\right)$$

W = Tooth Load, Lbs. (along the Pitch Line)
S = Safe Material Stress (static) Lbs. per Sq. In. (Table II)
F = Face Width, In.
Y = Tooth Form Factor (Table I)
P = Diametral Pitch
D = Pitch Diameter
V = Pitch Line Velocity, Ft. per Min. = .262 x D x RPM

For NON-METALLIC GEARS, the modified Lewis Formula shown below may be used with (S) values of 6000 PSI for Phenolic Laminated material.

$$W = \frac{SFY}{P}\left(\frac{150}{200 + V} + .25\right)$$

TABLE I TOOTH FORM FACTOR (Y)

Number of Teeth	14-1/2° Full Depth Involute	20° Full Depth Involute
10	0.176	0.201
11	0.192	0.226
12	0.210	0.245
13	0.223	0.264
14	0.236	0.276
15	0.245	0.289
16	0.255	0.295
17	0.264	0.302

TABLE II–VALUES OF SAFE STATIC STRESS (s)

Material		(s) Lb. per Sq. In.
Plastic		5000
Bronze		10000
Cast Iron		12000
Steel	.20 Carbon (Untreated)	20000
Steel	.20 Carbon (Case-hardened)	25000
Steel	.40 Carbon (Untreated)	25000
Steel	.40 Carbon (Heat-treated)	30000
Steel	.40 C. Alloy (Heat-treated)	40000

Max. allowable torque (T) that should be imposed on a gear will be the safe tooth load (W) multiplied by $\frac{D}{2}$ or $T = \frac{W \times D}{2}$

The safe horsepower capacity of the gear (at a given RPM) can be calculated from $HP = \frac{T \times RPM}{63{,}025}$ or directly from (W) and (V);

$$HP = \frac{WV}{33{,}000}$$

For a known HP, $T = \frac{63025 \times HP}{RPM}$

Other important catalog data

Couplings

Engineering formulas (horsepower, pitch line speed, etc.)

Application classifications for determining service factors

Engineering Information

Application Classification for Various Loads

Type of Machine To Be Driven	Chart I For All Drives — Service Factor Loading: Not More Than 15 Mins. in 2 Hrs.	Not More Than 10 Hrs. per Day	More Than 10 Hrs. Per Day
AGITATORS			
Pure Liquid	0.80	1.00	1.25
Semi-Liquids, Variable Density	1.00	1.25	1.50
BLOWERS			
Centrifugal and Vane	0.80	1.00	1.25
Lobe	1.00	1.25	1.50
BREWING AND DISTILLING			
Bottling Machinery	0.80	1.00	1.25
Brew Kettles–Continuous Duty	–	–	1.25
Cookers – Continuous Duty	–	–	1.25
Mash Tubs – Continuous Duty	–	–	1.25
Scale Hopper – Frequent Starts	–	1.25	1.50
CAN FILLING MACHINES	–	1.00	–
CANE KNIVES	–	1.50	–
CAR DUMPERS	–	1.75	–
CAR PULLERS	–	1.25	–
CLARIFIERS	–	1.00	1.25
CLASSIFIERS	–	1.25	1.50
CLAY WORKING MACHINERY			
Brick Press & Briquette Machine	–	1.75	2.00
Extruders and Mixers	1.00	1.25	1.50
COMPRESSORS			

Type of Machine To Be Driven	Chart I For All Drives — Service Factor Loading: Not More Than 15 Mins. in 2 Hrs.	Not More Than 10 Hrs. per Day	More Than 10 Hrs. Per Day
ELEVATORS			
Bucket – Uniform Load	–	1.00	1.25
Bucket – Heavy Load	–	1.25	1.50
Centrifugal Discharge	–	1.25	1.50
Freight	–	1.25	1.50
Gravity Discharge	–	1.00	1.25
FANS			
Centrifugal – Light (Small Diam.)	–	1.00	1.25
Large Industrial	–	1.25	1.50
FEEDERS			
Apron – Belt – Screw	–	1.25	1.50
Disc	–	1.00	1.25
Reciprocating	–	1.75	2.00
FOOD INDUSTRY			
Beet Slicer	–	1.25	1.50
Cereal Cooker	–	1.00	1.25
Dough Mixer – Meat Grinder	–	1.25	1.50
GENERATORS (NOT WELDING)	–	1.00	1.25
HAMMER MILLS	–	1.75	2.00
HOISTS			
Heavy Duty	–	1.75	2.00
Medium Duty and Skip Type	–	1.25	1.50

Example 14–1 (1)

A stock spur gear is available having a diametral pitch of 8 teeth/in, a $1\frac{1}{2}$-in face, 16 teeth, and a pressure angle of 20° with full-depth teeth. The material is AISI 1020 steel in as-rolled condition. Use a design factor of $n_d = 3$ to rate the horsepower output of the gear corresponding to a speed of 1200 rev/m and moderate applications.

Solution

The term *moderate applications* seems to imply that the gear can be rated by using the yield strength as a criterion of failure. From Table A–20, we find $S_{ut} = 55$ kpsi and $S_y = 30$ kpsi. A design factor of 3 means that the allowable bending stress is 30/3 = 10 kpsi. The pitch diameter is $d = N/P = 16/8 = 2$ in, so the pitch-line velocity is.

$$V = \frac{\pi d n}{12} = \frac{\pi(2)1200}{12} = 628 \text{ ft/min}$$

The velocity factor from Equation (14–4*b*) is found to be.

$$K_v = \frac{1200 + V}{1200} = \frac{1200 + 628}{1200} = 1.52$$

Example 14–1 (2)

Table 14–2 gives the form factor as $Y = 0.296$ for 16 teeth. We now arrange and substitute in Equation (14–7) as follows:

$$W^t = \frac{FY\sigma_{\text{all}}}{K_v P} = \frac{1.5(0.296)10\,000}{1.52(8)} = 365\,\text{lbf}$$

The horsepower that can be transmitted is.

$$hp = \frac{W^t V}{33\,000} = \frac{365(628)}{33\,000} = 6.95\,\text{hp}$$

Answer

It is important to emphasize that this is a rough estimate, and that this approach must not be used for important applications. The example is intended to help you understand some of the fundamentals that will be involved in the AGMA approach.

Example 14–2 (1)

Estimate the horsepower rating of the gear in the previous example based on obtaining an infinite life in bending.

Solution

The rotating-beam endurance limit is estimated from Equation (6–10),

$$S'_e = 0.5S_{ut} = 0.5(55) = 27.5 \text{ kpsi}$$

To obtain the surface finish Marin factor k_a we refer to Table 6–3 for machined surface, finding $a = 2.00$ and $b = -0.217$. Then Eq. (6–18) gives the surface finish Marin factor k_a as.

$$k_a = aS_{ut}^b = 2.00(55)^{-0.217} = 0.838$$

The next step is to estimate the size factor k_b. From Table 13–1, the sum of the addendum and dedendum is.

$$l = \frac{1}{P} + \frac{1.25}{P} = \frac{1}{8} + \frac{1.25}{8} = 0.281 \text{ in}$$

Example 14–2 (2)

The tooth thickness t in Figure 14–1b is given in Sec. 14–1 [Equation (b)] as $t = (4lx)^{1/2}$ when $x = 3Y/(2P)$ from Equation (14–3). Therefore, since from Example 14–1 $Y = 0.296$ and $P = 8$,

$$x = \frac{3Y}{2P} = \frac{3(0.296)}{2(8)} = 0.0555 \text{ in}$$

then

$$t = (4lx)^{1/2} = [4(0.281)0.0555]^{1/2} = 0.250 \text{ in}$$

We have recognized the tooth as a cantilever beam of rectangular cross section, so the equivalent rotating-beam diameter must be obtained from Equation (6–24):

$$d_e = 0.808(hb)^{1/2} = 0.808(Ft)^{1/2} = 0.808[1.5(0.250)]^{1/2} = 0.495 \text{ in}$$

Then, Equation (6–19) gives k_b as.

$$k_b = \left(\frac{d_e}{0.30}\right)^{-0.107} = \left(\frac{0.495}{0.30}\right)^{-0.107} = 0.948$$

The load factor k_c from Equation (6–25) is unity. With no information given concerning temperature and reliability we will set $k_d = k_e = 1$.

Example 14–2 (3)

In general, a gear tooth is subjected only to one-way bending. Exceptions include idler gears and gears used in reversing mechanisms. We will account for one-way bending by establishing a miscellaneous-effects Marin factor k_f .

For one-way bending the steady and alternating stress components are $\sigma_a = \sigma_m = \sigma/2$ where σ is the largest repeatedly applied bending stress as given in Equation (14–7). If a material exhibited a Goodman failure locus,

$$\frac{\sigma_a}{S'_e} + \frac{\sigma_m}{S_{ut}} = 1$$

Since σ_a and σ_m are equal for one-way bending, we substitute σ_a for σ_m and solve the preceding equation for σ_a, giving.

$$\sigma_a = \frac{S'_e S_{ut}}{S'_e + S_{ut}}$$

Now replace σ_a with $\sigma/2$, and in the denominator replace S'_e with $0.5S_{ut}$ to obtain.

$$\sigma = \frac{2S'_e S_{ut}}{0.5S_{ut} + S_{ut}} = \frac{2S'_e}{0.5 + 1} = 1.33S'_e$$

Example 14–2 (4)

Now defining a miscellaneous Marin factor $k_f = \sigma/S'_e = 1.33S'_e/S'_e = 1.33$. Similarly, if we were to use a Gerber fatigue locus,

$$\frac{\sigma_a}{S'_e} + \left(\frac{\sigma_m}{S_{ut}}\right)^2 = 1$$

Setting $\sigma_a = \sigma_m$ and solving the quadratic in σ_a gives.

$$\sigma_a = \frac{S_{ut}^2}{2S'_e}\left(-1 + \sqrt{1 + \frac{4S'^2_e}{S_{ut}^2}}\right)$$

Setting $\sigma_a = \sigma/2$, $S_{ut} = S'_e/0.5$ gives.

$$\sigma = \frac{S'_e}{0.5^2}\left[-1 + \sqrt{1 + 4(0.5)^2}\right] = 1.66S'_e$$

and $k_f = \sigma/S'_e = 1.66$. Since a Gerber locus runs in and among fatigue data and Goodman does not, we will use $k_f = 1.66$. The Marin equation for the fully corrected endurance strength is.

$$S_e = k_a k_b k_c k_d k_e k_f S'_e$$

$$= 0.838(0.948)(1)(1)(1)1.66(27.5) = 36.3 \text{ kpsi}$$

Example 14–2 (5)

For stress, we will first determine the fatigue stress-concentration factor K_f. For a 20° full-depth tooth the radius of the root fillet is denoted r_f, with a typically proportioned value of

$$r_f = \frac{0.300}{P} = \frac{0.300}{8} = 0.0375 \text{ in}$$

From Figure A–15–6

$$\frac{r}{d} = \frac{r_f}{t} = \frac{0.0375}{0.250} = 0.15$$

Since $D/d = \infty$, we approximate with $D/d = 3$, giving $K_t = 1.68$. From Figure 6–26, $q = 0.62$. From Equation (6–32),

$$K_f = 1 + (0.62)(1.68 - 1) = 1.42$$

For a design factor of $n_d = 3$, as used in Example 14–1, applied to the load or strength, the maximum bending stress is.

$$\sigma_{\text{max}} = K_f \sigma_{\text{all}} = \frac{S_e}{n_d}$$

$$\sigma_{\text{all}} = \frac{S_e}{K_f n_d} = \frac{36.3}{1.42(3)} = 8.52 \text{ kpsi}$$

Example 14–2 (6)

The transmitted load W^t is.

$$W^t = \frac{FY\sigma_{\text{all}}}{K_v P} = \frac{1.5(0.296)8520}{1.52(8)} = 311 \text{ lbf}$$

and the power is, with V = 628 ft/min from Example 14–1,

$$hp = \frac{W^t V}{33\,000} = \frac{311(628)}{33\,000} = 5.9 \text{ hp}$$

Again, it should be emphasized that these results should be accepted *only* as preliminary estimates to alert you to the nature of bending in gear teeth.