Gears

Just stare at the machine. There is nothing wrong with that. Just live with it for a while. Watch it the way you watch a line when fishing and before long, as sure as you live, you’ll get a little nibble, a little fact asking in a timid, humble way if you’re interested in it. That’s the way the world keeps on happening. Be interested in it.

Robert Piersig, Zen and the Art of Motorcycle Maintenance

Image: An assortment of gears.
GEARS
- a very effective & efficient method of motion & power transfer
  - efficient up to 98%
  - strong (teeth)
  - compact

- Cost?
- Noise and vibration

I. Types of Gears
II. Tooth geometry, conjugate action
III. Gear trains ... Kinematics
IV. Design for strength (bending teeth, surface fatigue)
- AGMA design codes
- American Manufacturers Association
- Gears usually lubricated and operate under elastohydrodynamic lubrication to avoid metal to metal contact
- Of course not all gears are metal; plastic-gear design is specialty area... we won't consider.
"HEXICAL" TEETH MAKE SMOTHER ACTION

Helical Gear Drive
Straight Tooth Bevel Gear Drive

(Like rolling cones)

Bevel gear drive with straight teeth.

Nonparallel shafts... can be 90° or other...
SPRIMAL BEVEL
(bored equiv. of helical)

Center of

HYPOID (AXES OFFSET)
(IN AUTO DIFFERENTIALS)
Figure 14.4 Worm gear drive. (a) Cylindrical teeth; (b) double enveloping.

"Cross between screw & gear"

WORM DRIVES WHEEL
(WHEEL CANT DRIVE WORM)
Law of Gearing

- Angular velocity ratio is constant
- Pitch point velocity same on "gear" & "pinion" smaller of two gears

\[
\frac{\omega_g}{\omega_p} = \frac{r_p}{r_g}
\]
- Pitch point "P" is fixed
- Various tooth profiles are possible to achieve:
  \[ \frac{\omega_g}{\omega_p} = \text{Const.} \]

- Most commonly std is the involute shape.
- another is cycloid
Involute Curve

Construction of involute curve.
Some terms

Circular pitch, \( p \)
... distance along pitch circle between same pts on adjacent teeth

Module, \( m \) (in SI)

... pitch diameter, \( d \) (mm)

Number of teeth, \( N \)

Diametral Pitch, \( P \)

Number of Teeth \( \frac{N}{d} \) (in)
Figure 14.7 Standard diametral pitches compared with tooth size.

Full size is assumed.

Text Reference: Figure 14.7, page 622

Hamrock, Jacobson and Schmid

**Preferred Diametral pitches & modules**

<table>
<thead>
<tr>
<th>DIAMETRAL PITCHES</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coarse</td>
</tr>
<tr>
<td>2, 2(\frac{1}{4}), 2(\frac{1}{2}), 3, 4, 6, 8, 10, 12, 16</td>
</tr>
<tr>
<td>Fine</td>
</tr>
<tr>
<td>20, 24, 32, 40, 48, 64, 80, 96, 120, 150, 200</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>MODULES</th>
</tr>
</thead>
<tbody>
<tr>
<td>Preferred</td>
</tr>
<tr>
<td>1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 16, 20, 25, 32, 40, 50</td>
</tr>
<tr>
<td>Next choice</td>
</tr>
<tr>
<td>1.125, 1.375, 1.75, 2.25, 2.75, 3.5, 4.5, 5.5, 7, 9, 11, 14, 18, 22, 28, 36, 45</td>
</tr>
</tbody>
</table>
Tooth action

- Pitch point is fixed
- Pressure line, line of action followed as teeth interact along common normal
- Theoretically there is pure rolling between teeth at pitch point... elsewhere slide + roll action.
Pressure line, line of action along which contact occurs.

Pressure Angle

Pinion
Driver

Dedendum circle
Base circle
Pitch circle
Addendum circle

Angle of approach
Angle of recess

Angle of approach
Addendum circle
Pitch circle
Base circle
Dedendum circle

Gear
Driven

$\theta$

Point of first contact

$A \rightarrow$ point of first contact

$B \rightarrow$ point of final contact
<table>
<thead>
<tr>
<th>TOOTH SYSTEM</th>
<th>PRESSURE ANGLE $\phi$, deg</th>
<th>ADDENDUM $a$</th>
<th>DEEDENDUM $b$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Full depth</td>
<td>20</td>
<td>$1/P_d$ or $1m$</td>
<td>$1.25/P_d$ or $1.25m$</td>
</tr>
<tr>
<td></td>
<td>$22\frac{1}{2}$</td>
<td>$1/P_d$ or $1m$</td>
<td>$1.35/P_d$ or $1.35m$</td>
</tr>
<tr>
<td></td>
<td>25</td>
<td>$1/P_d$ or $1m$</td>
<td>$1.25/P_d$ or $1.25m$</td>
</tr>
<tr>
<td></td>
<td>20</td>
<td>$0.8/P_d$ or $0.8m$</td>
<td>$1/P_d$ or $1m$</td>
</tr>
</tbody>
</table>

20° Pressure angle is std & most common.

Shorter teeth: see proportions.
Contact ratio, \( M_c \)
- Avg # of teeth in contact
- Should be greater than 1.2

Interference
- to be avoided ... when gear tooth & pinion tooth overlap. ... reason for clearance
Backlash
- amount by which tooth space exceeds thickness of mating gear
Backlash

Illustration of backlash in gears.
<table>
<thead>
<tr>
<th>Diametral pitch $P_d$, in.</th>
<th>Center distance in.</th>
<th>2</th>
<th>4</th>
<th>8</th>
<th>16</th>
<th>32</th>
</tr>
</thead>
<tbody>
<tr>
<td>18</td>
<td>0.005</td>
<td>0.006</td>
<td>0.007</td>
<td>0.008</td>
<td>0.010</td>
<td>0.012</td>
</tr>
<tr>
<td>12</td>
<td>0.006</td>
<td>0.007</td>
<td>0.007</td>
<td>0.010</td>
<td>0.012</td>
<td>0.014</td>
</tr>
<tr>
<td>8</td>
<td>0.006</td>
<td>0.007</td>
<td>0.007</td>
<td>0.010</td>
<td>0.012</td>
<td>0.014</td>
</tr>
<tr>
<td>5</td>
<td>0.007</td>
<td>0.007</td>
<td>0.007</td>
<td>0.010</td>
<td>0.012</td>
<td>0.014</td>
</tr>
<tr>
<td>3</td>
<td>0.006</td>
<td>0.007</td>
<td>0.007</td>
<td>0.010</td>
<td>0.012</td>
<td>0.014</td>
</tr>
<tr>
<td>2</td>
<td>0.007</td>
<td>0.007</td>
<td>0.007</td>
<td>0.010</td>
<td>0.012</td>
<td>0.014</td>
</tr>
<tr>
<td>1.25</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Recommended minimum backlash for coarse-pitch gears.
GEAR TRAINS

- External mesh
- Internal mesh
- Rack & pinion
- Simple gear train
- Compound gear train
- Planetary gear sets

From Law of Gearing

\[
\frac{W_g}{W_p} = \frac{r_p}{r_g} = \frac{d_p}{d_g} \quad \text{(number of teeth)}
\]

but \( d = \frac{N}{P} \) & $ P \]
\[
\left( \frac{W_g}{W_p} \right) = \frac{N_p}{N_g}
\]

\[ \text{diameter, pitch} \]
Externally Meshing Spur Gears

**PINION**

\[ N_1 \]

Gear 1

\[ N_1 \text{ teeth} \]

Gear 2

\[ N_2 \text{ teeth} \]

\[ W_1 \]

\[ W_2 \]

\[ \omega_1 \]

\[ \omega_2 \]

\[ + \text{ccw} \]

\[ - \text{cw} \]

Externally meshing spur gears.

\[ \frac{W_2}{W_1} = -\frac{N_1}{N_2} \]

\[ W_2 \neq W_2 \text{ in opposite directions} \]
INVOLOUTE RACK & PINION

RACK \( (r = \infty) \)

Base pitch

Circular pitch

PINION

\( V_p = r_p \omega_p \)

\( V_p \)

pitch circle

NOTE STRAIGHT LINE TOOTH SHAPE ON RACK
Simple Gear Train

\[ \begin{align*}
N_4 &\rightarrow N_3 \\
N_3 &\rightarrow N_2 \\
N_2 &\rightarrow N_1 \\
N_2 &\rightarrow N_3 \\
N_3 &\rightarrow N_4 \\
N_4 &\rightarrow N_5
\end{align*} \]

\[ w_5 = \frac{N_3}{N_1} \]

\[ w_1 = \frac{N_1}{N_5} \]

Output:

\[ w_2 = \frac{N_1}{N_2} \]

\[ w_3 = \frac{N_2}{N_3} \]

\[ w_4 = \frac{N_3}{N_4} \]
Compound Gear Train

\[ \omega_2 = \frac{-N_1}{N_2} \]

\[ \omega_5 = -\frac{N_2}{N_5} \]

\[ \omega_1 = \frac{N_6}{N_7} \]

\[ \omega_7 = \frac{N_8}{N_7} \]

\[ \omega_8 = \omega_9 \]

\[ i = \frac{\omega_9}{\omega_8} \]
Therefore:

\[ w_8 = \left( -\frac{N_7}{N_8} \right) w_7 = \left( -\frac{N_3}{N_8} \right) w_6 \]

\[ = \left( -\frac{N_7}{N_8} \right) \left( -\frac{N_5}{N_6} \right) w_5 \]

\[ = \left( -\frac{N_7}{N_8} \right) \left( -\frac{N_5}{N_6} \right) w_4 \]

\[ = \cdots \text{ etc... so that:} \]

\[ w_8 = \left( -\frac{N_7}{N_8} \right) \left( -\frac{N_5}{N_6} \right) \left( -\frac{N_3}{N_4} \right) \left( -\frac{N_1}{N_2} \right) w_1 \]

\[ w_8 = \left( \frac{N_7 N_5 N_3 N_1}{N_8 N_6 N_4 N_2} \right) w_1 \]
Gears inside industrial mixer.

1. Blade
   - Rotates on shaft
2. Blade rotates within bowl?

[Courtesy of Hobart]
NOTE: \( N_{\text{ring}} = N_{\text{sun}} + 2N_{\text{planet}} \)

Planet axis

1. Planet rotates about axis
2. And it sees one additional rotation with each rotation of arm

Illustration of planetary gear.
(a) With three planets (typical); (b) with one planet (for analysis only).

PLANETARY OR EPICYCLIC GEARS

TWO DEGREES OF FREEDOM
Example:

1. Fixed ring
2. Sun rotates at 600 radians/sec
3. Find warm and W planet
Governing eqs to be used

\[ \frac{N_{\text{sun}}}{N_{\text{ring}}} = \frac{W_{\text{ring}} - \text{Warm}}{W_{\text{sun}} - \text{Warm}} \quad \ldots (1) \]

Also

\[ \frac{N_{\text{sun}}}{N_{\text{planet}}} = \frac{W_{\text{planet}} - \text{Warm}}{W_{\text{sun}} - \text{Warm}} \quad \ldots (2) \]

And

\[ \frac{N_{\text{planet}}}{N_{\text{ring}}} = \frac{W_{\text{ring}} - \text{Warm}}{W_{\text{planet}} - \text{Warm}} \quad \ldots (3) \]

\[ \uparrow \quad \text{Eqs applicable for values of Warm} \]

Gear ratios with arm fixed \ldots \text{ i.e. } \text{Warm} = 0
Start with warm \( \Phi \) eq (1)

\[-\frac{20}{80} = 0 - \text{Warm}\]

\[-\frac{20}{80} = \frac{600 - \text{Warm}}{600 - \text{Warm}}\]

\[
\text{Warm} = \frac{150}{1.25} = 120 \text{ radians/s}
\]

Now find \( \omega_{\text{planet}} \) from eq (2)

\[-\frac{20}{30} = \frac{\omega_{\text{planet}} - 120}{600 - 120}\]

\[
\omega_{\text{planet}} = -\frac{20}{30} (480) + 120
\]

\[
\omega_{\text{planet}} = -240 \text{ radians/s}
\]
INDUSTRIAL MIXER (Revisited)

Driving torque to sun gear

Ring gear is fixed

Blade attaches here... spins around planet axis and moves around bowl
Pressure line \( \phi \)

Tooth Forces

\[ F_{32}^t = W_t \]

Bearing Forces

\[ W_t = \frac{HV}{33000 \text{ in } \text{rpm}} \]

\[ V = \frac{\pi \text{\pi} \text{dn}^{12}}{12} \]

\[ H = \text{horsepower} \]

\[ W_t = H(60)(10^3) \]

\[ \frac{\text{kN}}{\text{mm} \text{rpm}} \]
LOADING OF TEETH

Contact Stress

bending with Stress concentration
Size/Strength of Gear Teeth

Chapter 14

- We will cover methods in 14-1 & 14-2

- We will not use AGMA Stress formulas which cover the same mechanics as 14-1 & 14-2 but include more correction factors

- In practice use AGMA formulas
BENDING STRESSES

0.3
--- = f
P

For 20° fall depth (inches)

Like a beam subjected to W_t

Stress at a

\[ \sigma = \frac{Mc}{I} = \frac{W_t l (t/2)}{F t^3} \]

= \( \frac{6 W_t E}{F t^2} \)

F is far width
Lewis Form Factor, \( y \) (dimensionless)

\[
y = \frac{2 \times p}{3}
\]

\( y \) depends on tooth shape

- e.g. pressure angle
- whether full-depth or stub teeth.

Developed by Wilfred Lewis in 1892

**In terms of \( y \)**

\[
\sigma = \frac{W_{tp}}{FY}
\]
Table 14-2

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

Y for 20° Full-depth
(Note that N < 12 is not used)

0.245 ≤ Y ≤ 0.485
Nominal max stress ("static")

... include velocity factor $K_v$ ... acts for impacts & other dynamic effects

\[
K_v = \frac{1200}{1200 + V}
\]

$V$ ft/min → American units

\[
K_v = \frac{6.1}{6.1 + V}
\]

$V$ m/s → SI
Bending stress calc becomes

American: \[ \sigma = \frac{W_t + P}{K_v F_Y} \]

SI: \[ \sigma = \frac{W_t}{K_v F_{mY}} \]

Use Goodman "Approach"... Stress fluctuates between 0 and \( \sigma \)

\[ \sigma_2 = \sigma_m = \frac{\sigma}{2} \]
- Obtain straight $S_i$

Via $S_i = k_b k_i k_b k_i$  \[ S_i = 1.33 \]

- Two parts

Instead use $k_a$ in

$\frac{k} {k_a} = k_a (\frac{1} {k_a})$
$k_a \rightarrow$ surface finish

$k_b \rightarrow$ size obtain equiv. diameter

$k_c = 1$ Loading (bending)

$k_d = 1$ Temperature

$k_e = \frac{1.33}{Kf}$

And:

$\frac{52}{60} = n_a$  \[\text{For BENDING}\]

\[\text{INFINITE LIFE IN FATIGUE}\]
Surface Durability

- Based on Hertzian stresses
- Compute $\sigma_e$ (MAX comp. compression)

$$\sigma_e = \frac{1}{\sqrt{n \left( \frac{1 - \frac{1}{E_p^2}}{E_p^2} + \frac{1 - \frac{1}{E_1^2}}{E_1^2} \right)}}$$

"Equivalent Modulus" in AGMA Standards

$$\frac{We}{CvF \cos \phi \left( \frac{1}{r_1} + \frac{1}{r_2} \right)}$$

\[ \vdash \text{use radii @ pitch line} \]

\[ \vdash Cv = Kv \text{ (from before) } \]
\[ \eta_c = \frac{5c}{10c_1} \quad \text{Comp. Strength} \]

Factor of safety for contact fatigue

Above is for 10^8 cycles

\[ 5c \quad \text{is reduced for large } \# \text{, increased for smaller } \]

\[ 5c = 0.4H_B - 10 \text{ kpsi} \quad \text{or} \quad 2.76H_B - 70 \text{ MPa} \]

For Steels: \[ 5a = 0.45H_B \text{ kpsi} \quad 5a = 3.10H_B \text{ MPa} \]
Radius of curvature of gear teeth @ pitch line

\[ r_1 = \frac{d_p \sin \phi}{2} \]

\[ r_2 = \frac{d_g \sin \phi}{2} \]

Pinion

Gear.
(Slight)

**Example Variation**

on 14-1, 14-2, 14-3

following methodology
develop on preceding 9 pages
of notes. (To be used for
design purposes in this
course)

---

**GEAR**: Spar Gear

\[
P = 8 \text{ teeth/in} \\
F = 1.5 \text{ in.}
\]

Table A-20

\[
N = 16 \text{ teeth} \\
S_{ut} = 55 \text{kpsi} \\
\phi = 20^\circ \\
S_{yt} = 30 \text{kpsi} \\
A151 1020 Steel (rolled) \\
N = 1200 \text{ rev/min}
\]
I

Determine HP capacity based on "static failure" (No stress concentration or modifying factors)

... use factor $n_s = 3$

... bending only

i.e. $n_s = \frac{S_y}{f} = 3$

$W_t = \frac{K_v F Y \sigma}{P}$

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

$= \frac{1200}{1200 + 628}$

$K_v = 0.656$

$V = \frac{\pi (N/P)(1200)}{12} = \frac{628 ft/min}{12} = 628 ft/min$
$$\sigma = \frac{5\gamma}{N^3} = \frac{30}{3} = 10 \text{ksi}$$

From Table 14-2 \( Y = 0.296 \)

(for \( P = 0 \) \( N = 16 \))

\[ W_t = (0.656)(1.5)(0.296)(10^4) \]

\[ W_t = 364 \text{ lbs} \]

HP is \( H = \frac{W_t V}{33,000} \)

\[ H = 6.93 \text{ hp} \]
II Determine hip capacity for infinite life (Bending)

\[ \sigma = \frac{S_e}{\pi d} = \frac{S_e}{3} \]

Need to determine \( S_e \)

\[ S_e = k_a k_b k_c k_d k_e S_e' \]

\[ S_e' = 0.504 S_{ut} \]

\[ = (0.504 \times 55) = 27.7 \text{ ksi} \]

\[ k_a = a S_{ut} \]

\[ a = 14.4 \quad \text{Table} \]

\[ b = -0.718 > 7-4 \]

\[ k_a = 0.811 \]
\[ k_b = \left( \frac{d_e}{0.3} \right)^{-0.1133} = \left( \frac{0.495}{0.3} \right)^{0.1133} = 0.999 \]

\[ \text{To find } d_e = 0.808(b_b)^{1/2} \]

(Equidiam = 0.808(Ft)^{1/2} eq. 7-19)

\[ \text{need } t; \text{ (tooth thickness) } \]

\[ t = (42x)^{1/2} \]

\[ \frac{0.3}{P} = f_f \]

\[ \frac{1}{P} + \frac{1.25}{P} = 0.281 \text{ in} \]

\[ x = \frac{3Y}{2P} = \frac{3(0.296)}{2(8)} \]

\[ x = 0.0555 \text{ in} \]

\[ t = (4x)^{1/2} = 0.25 \text{ in} \]

\[ d_e = 0.808[(1.5)(0.25)]^{1/2} \]

\[ d_e = 0.495 \text{ in} \]
\[ k_e = 1 \text{ (bending)} \]
\[ k_d = 1 \text{ (No temperature effect)} \]
\[ k_e = 1.33 \left( \frac{1}{k_f} \right) \]
\[ k_f = 1 + 0.78 \left( K_e - 1 \right) \]
\[ q = 0.78 \quad \left( \text{From Fig. 5-16} \right) \]
\[ K_e \text{ from Fig A-15-6} \]
\[ K_f = 1 + (0.78)(1.68 - 1) \]
\[ = 1.53 \]
\[ K_e = 1.33 \left( \frac{1}{1.53} \right) = 0.87 \]

Now
\[ S_e = (0.811)(0.945)(1.0)(0.87)(27.7) \]
\[ S_e = 18.4 \text{ kpsi} \]

\[ \sigma = \frac{S_e}{n} = \frac{18.4}{3} = 6.1 \text{ kpsi} \]

\[ W_t = F_Y \left( \frac{H}{P} \right)(6180) = 224 \text{ lb} \]

\[ H = \left( \frac{224}{364} \right)(6.93) = 4.26 \text{ hp} \]

\[ \text{Recall for static failure was 10 kpsi,} \]
\[ \geq 60\% \text{ of static failure} \]
III. Determine factor of safety against surface fatigue (10^6 cycles)

\[ n = \frac{S_c}{P_c} \]

for \( W_t = 224 \text{ lb} \) and \( H = 4.26 \text{ hp} \)

(both AISI 1020 steel)

\[ C_p = \sqrt{\frac{1}{\pi \left(1 - \frac{1}{2}\right) + \left(1 - \frac{1}{2}\right)}} \]

\[ E_p = 30 \text{ Mpsi} = E_5 \]

\[ \frac{1}{\rho} = \frac{1}{\rho_5} = 0.3 \]

\[ C_p = \frac{2095}{\sqrt{2}} \]

\[ r_1 = \frac{25 + 28}{2} = 0.342^\circ; r_2 = 1.069 \text{ in} \]

- 16 tooth pinion mesh with 50 tooth

\[ n = \frac{S_c}{P_c} \]
\[ \sigma_c = -2095 \left[ \frac{\text{Wk}}{C_V F \cos \phi} \left( \frac{1}{r_1} + \frac{1}{r_2} \right) \right] \]

\[ = -2095 \left[ \frac{224 \left( \frac{1}{.342} + \frac{1}{1.069} \right)}{.656(1.5) \cos 20^\circ} \right] \]

Same as

\[ K_V \]

Max.

Contact Stress

\[ \sigma_c = -63,756 \text{ psi} \]

\[ S_c = 0.4 H_B - 10 \text{ kips} \]

Note that

\[ H_B = \frac{S_{ut}}{0.45} = \frac{55}{0.45} = 122 \]

\[ S_c = 38,500 \text{ kips} = (0.4(122) - 10) \text{ kips} \]
\[ n = \frac{S_c}{1000} = \frac{38,800}{63,758} = 0.61 \]

But 222 lb = \( W_t \) includes some factor of safety (bending fatigue ... \( n_s = 3 \) 

If we recalculate \( \sigma_c \) with

\[ \frac{224}{3} = 74.716 = W_t \]

\[ \sigma_c' = 36,809 \]

\[ n = \frac{38,800}{36,809} = 1.05 \]

Still much less than 3.
### Fatigue (Design of Mechanical Elements)

**Aluminum Bending Strength, \( S_e \)**

<table>
<thead>
<tr>
<th>AGMA CLASS</th>
<th>COMMERCIAL DESIGNATION</th>
<th>HEAT TREATMENT</th>
<th>MINIMUM HARDNESS AT SURFACE</th>
<th>CORE</th>
<th>( S_e ) psi</th>
</tr>
</thead>
</table>

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**Surface Fatigue Strength, \( S_e \)**

<table>
<thead>
<tr>
<th>AGMA CLASS</th>
<th>COMMERCIAL DESIGNATION</th>
<th>HEAT TREATMENT</th>
<th>MINIMUM HARDNESS AT SURFACE</th>
<th>( S_e ) psi</th>
</tr>
</thead>
<tbody>
<tr>
<td>A-1 through A-5</td>
<td>Through-hardened and tempered</td>
<td>180 BHN and less 240 BHN 300 BHN 360 BHN 400 BHN</td>
<td></td>
<td>85–95 000 (590–660) 105–115 000 (720–790) 120–135 000 (830–930) 145–160 000 (1000–1150) 155–170 000 (1100–1250)</td>
</tr>
</tbody>
</table>

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Some Standard Gear Steels

For gears, we use much stronger steels.

Note high hardness for high surface strength.