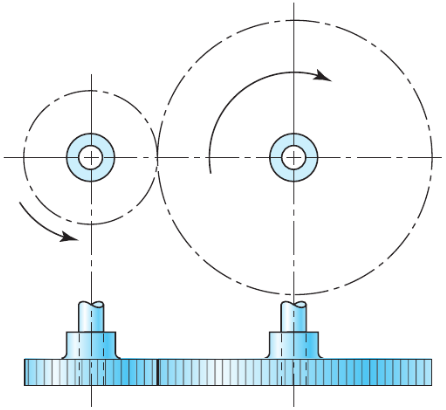
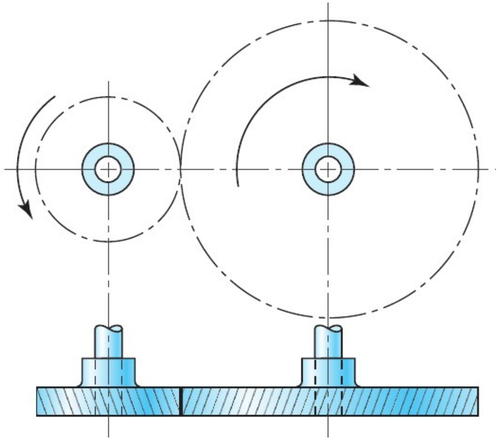
# Introduction

Gears are mechanical components with teeth that interlock with other gears to transmit motion and force.

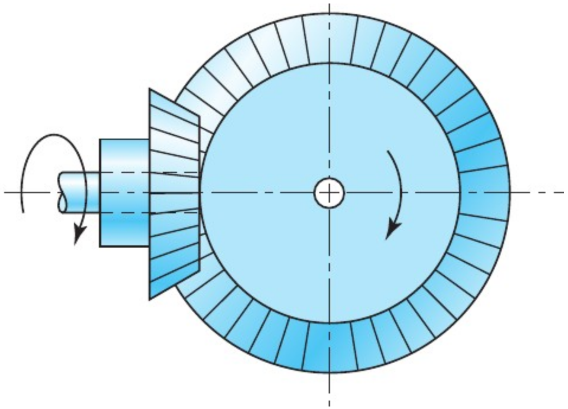
They are used to change the direction of motion, increase or decrease speed, and adjust torque in various machines



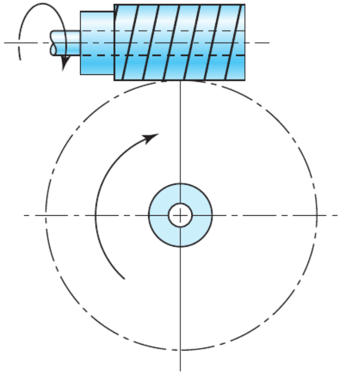
Spur



Helical



Bevel



Worm and Worm Wheel

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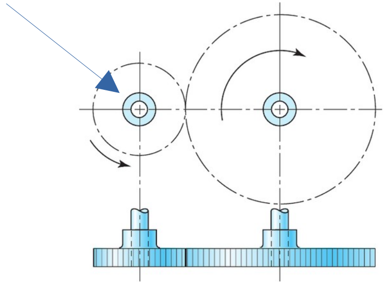
# Gears – Common Types


Convention

Smaller of the two gears is called pinion



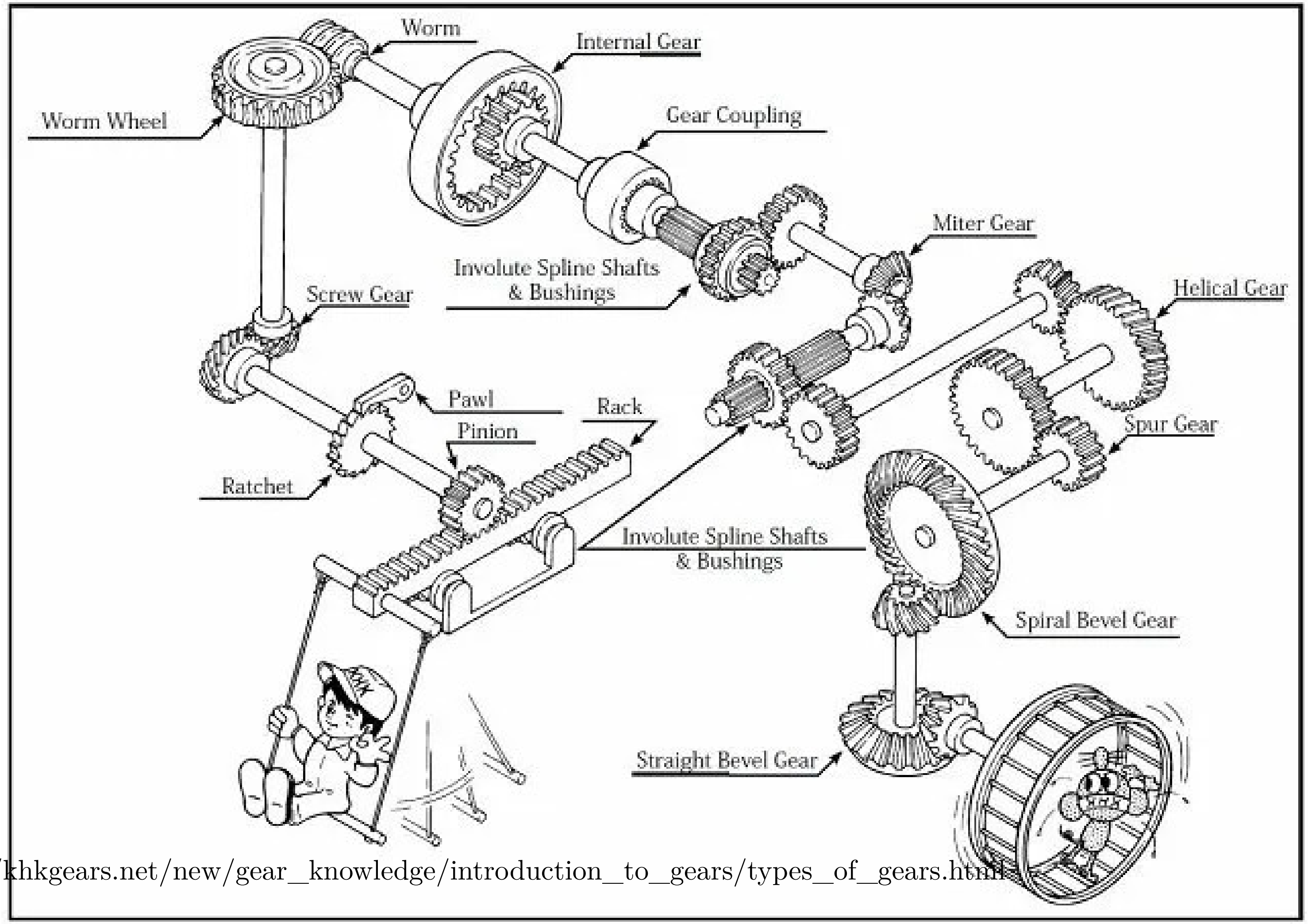


https://www.theengineeringchoice.com/what-is-gear/

# Gears – Common Types

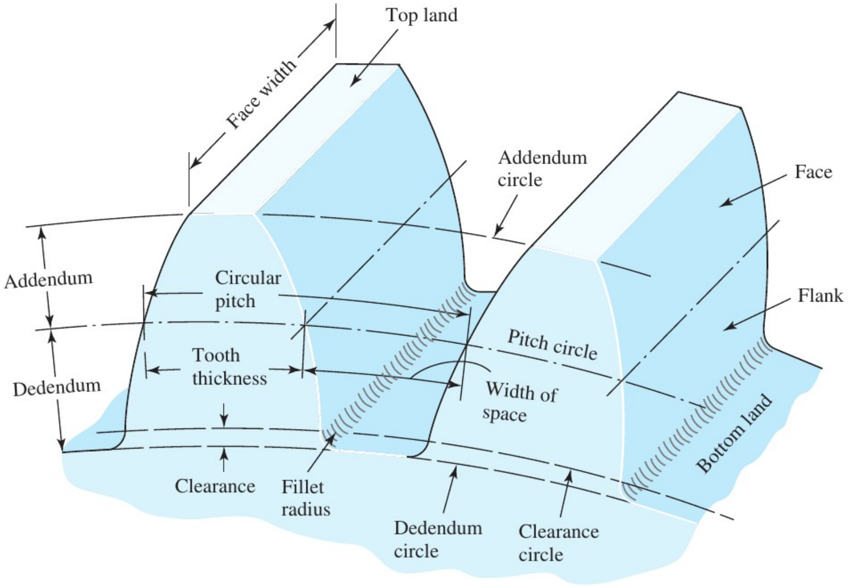
- **Spur gears** – have teeth parallel to the axis of rotation and are used to transmit motion/torque between parallel shafts
- **Helical gears** – have teeth inclined to the axis of rotation and are mostly used to transmit motion/torque between parallel shafts
- **Bevel gears** – have teeth formed on conical surfaces and are used mostly for transmitting motio between intersecting shafts, either perpendicular or at an angle
- **Worm and worm gear** – used to transmit motion between non-intersecting shafts and are used when the speed ratios are high, i.e. $> 3$
- Hypoid gears – similar to bevel gears except that shafts are offset and non-intersecting
- Mitre gears - bevel gears with ratio 1:1
- Herringbone gear - similar to double helical gears but with no gap

# Gears – Common Types




https://khkgears.net/new/gear_knowledge/introduction_to_gears/types_of_gears.html

# Nomenclature of the Spur-Gear Tooth




**Terminology**

- **Pitch circle**: theoretical circle on which all the calculations are based. The pitch circles of the mating gears are tangent to each other.
- **Pitch diameter** $d$ is the diameter of the pitch circle.
- **Module** $m$ is the ratio of the pitch diameter to the number of teeth, $N$. $m = \dfrac{d}{N}$
- **Addendum** is the radial distance between the top land and the pitch circle.
- **Dedendum** is the radial distance from the bottom land to the pitch circle.

- **Whole depth** $h_t$ is the sum of the addendum and the dedendum
- **Backlash** – amount by which the width of the tooth space exceeds the thickness of the engaging tooth measured at the pitch circle
- **Circular pitch** $p$ distance measure along the pitch circle from a point on one tooth to a corresponding point on adjacent tooth

https://khkgears.net/new/gear_knowledge/gear_technical_reference/calculation_gear_dimensions.html

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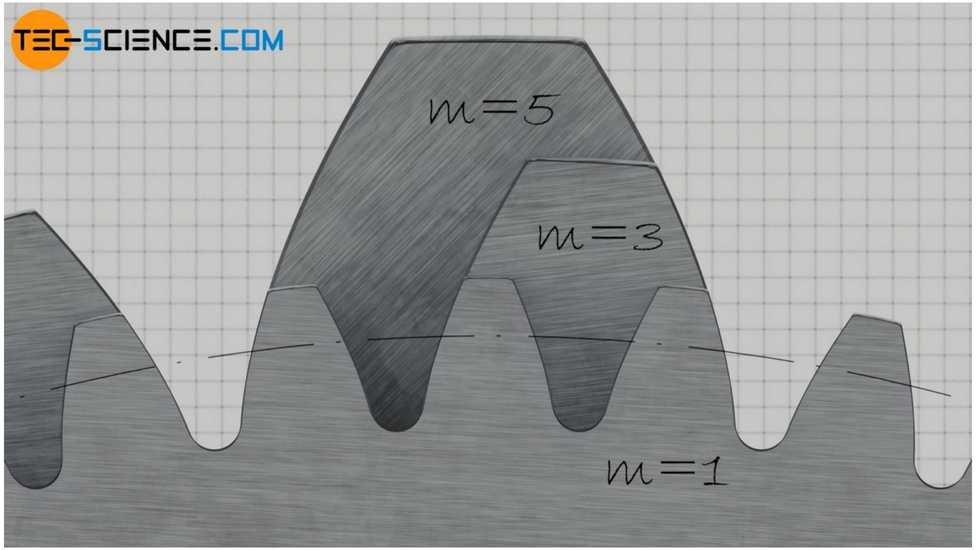
# Module

Examples of modules in different applications (as the load increases, the module increases

Wrist watches ~ 0.05 – 0.2

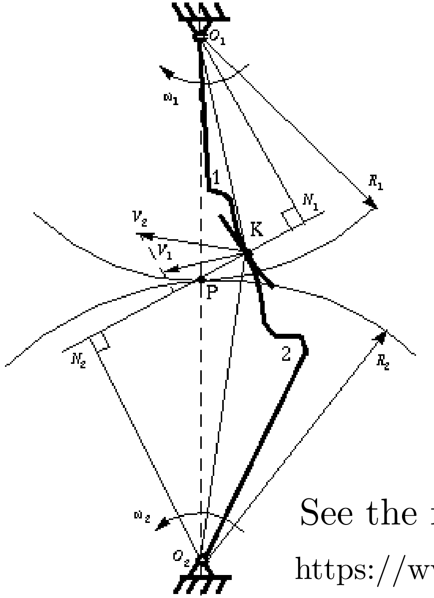
Printers, office copiers ~ 0.6 – 1.0

Transmissions for cars ~ 1.5 – 3.0



# Conjugate Action and Involute Profile

- When tooth profiles or cams are designed to produce a constant angular velocity ratio during meshing, they are said to have conjugate action
- Fundamental law of gear tooth action – to obtain a constant angular velocity ratio during meshing, the common normal at the point of contact between a pair of gear teeth must always pass through a fixed point on the line of centers (pitch point).




$$\left|\frac{\omega_1}{\omega_2}\right| = \frac{O_2P}{O_1P} = \frac{R_2}{R_1} \quad \text{(law of gearing)}$$

Involute profile satisfies the above requirement – most commonly used gear profile

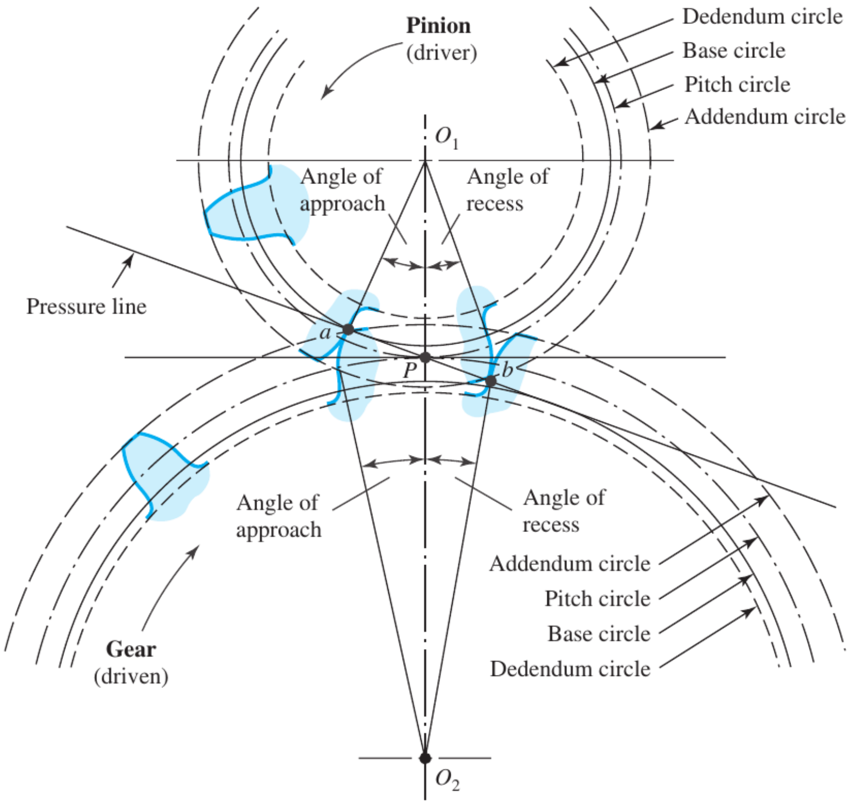
See video below for gears with cycloidal profile

See the following videos

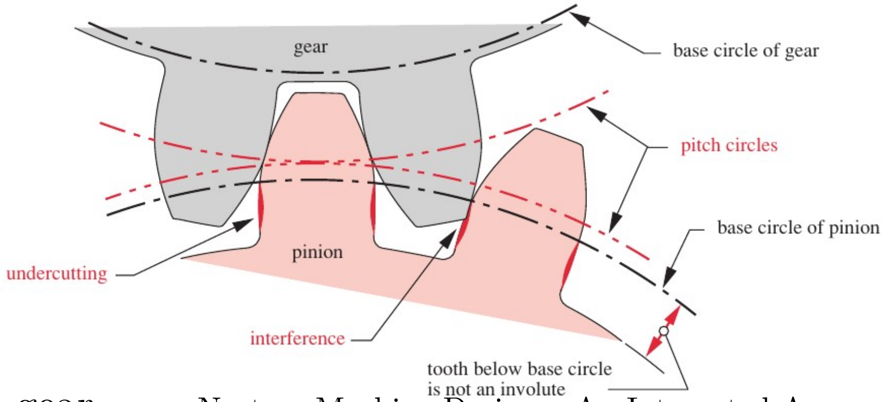
https://www.tec-science.com/mechanical-power-transmission/involute-gear/meshing-line-action-contact-pitch-circle-law/

https://www.tec-science.com/mechanical-power-transmission/cycloidal-gear/geometry-of-cycloidal-gears/

# Tooth Action




- Pressure line = line of action = generating line
- Pressure angle = 20° (most common) – angle between the line of action and the direction of velocity at the pitch point
- Contact ratio – average number of teeth in contact at any given instant ~ 1.2
- Interference – contact of tooth profiles which are not conjugate is called interference (tooth profiles are not involute below the base circle)




Base circle is used to generate the involute profile – basic to a gear.

$$r_b = r_p \cos\phi$$

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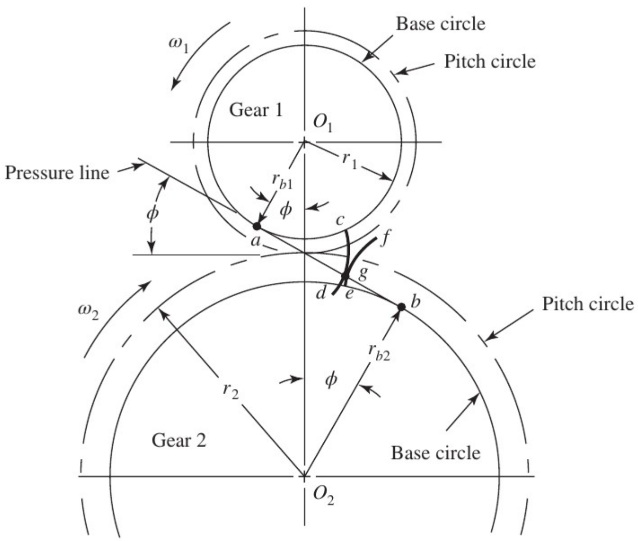
# Problem

A gearset consists of a 16-tooth pinion driving a 40-tooth gear. The module is 12 mm, and the addendum and dedendum are 1 m and 1.25 m, respectively. The gears are cut using a pressure angle of 20°.

(a) Compute the circular pitch, the center distance, and the radii of the base circles.

(b) In mounting these gears, the center distance was incorrectly made 6.35 mm larger. Compute the new values of the pressure angle and the pitch-circle diameters.



# Tooth Systems - American Gear Manufacturers Association (AGMA)

A tooth system is a standard that specifies the relationships involving addendum, dedendum, working depth, tooth thickness, and pressure angle.

**Table 13–1** **Standard and Commonly Used Tooth Systems for Spur Gears**

| Tooth System | Pressure Angle $\phi$, deg | Addendum $a$ | Dedendum $b$ |
|---|---|---|---|
| Full depth | 20 | $1/P$ or $m$ | $1.25/P$ or $1.25m$ |
| | | | $1.35/P$ or $1.35m$ |
| | $22\frac{1}{2}$ | $1/P$ or $m$ | $1.25/P$ or $1.25m$ |
| | | | $1.35/P$ or $1.35m$ |
| | 25 | $1/P$ or $m$ | $1.25/P$ or $1.25m$ |
| | | | $1.35/P$ or $1.35m$ |
| Stub | 20 | $0.8/P$ or $0.8m$ | $1/P$ or $m$ |

| **Module $m$ (mm/tooth)** | |
|---|---|
| Preferred | 1, 1.25, 1.5, 2, 2.5, 3, 4, 5, 6, 8, 10, 12, 16, 20, 25, 32, 40, 50 |
| Next Choice | 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 |

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# Calculation for Standard Spur Gears – Type 1

| Item | Symbol/Formula |
|---|---|
| Center distance | $a$ (**given**) |
| Speed ratio | $i$ ($i = N_g/N_p$) (**given**) |
| module | $m$ (**choose**) |
| Pressure angle | $\boldsymbol{\phi}$ (**choose**) |
| Number of teeth | $N_p$, $N_g$ (calculate) |
| Pitch diameter | dp = mNp,dg = mNg (calculate) |
| Base diameter | dbp = dp cos$\boldsymbol{\phi}$ dbg = dg cos$\boldsymbol{\phi}$ (calculate) |
| Addendum | $m$ |
| Tooth depth | 1.25 $m$ |
| Tip diameter | dp+2$m$, dg+2$m$ |
| Root diameter | dp-2.5m,dg-2.5m |

$$N_p = \frac{2a}{m(i+1)}$$

$$N_g = \frac{2ai}{m(i+1)}$$

https://khkgears.net/new/gear_knowledge/gear_technical_reference/calculation_gear_dimensions.html

# Calculation of Standard Spur Gears – Type 2 Problems

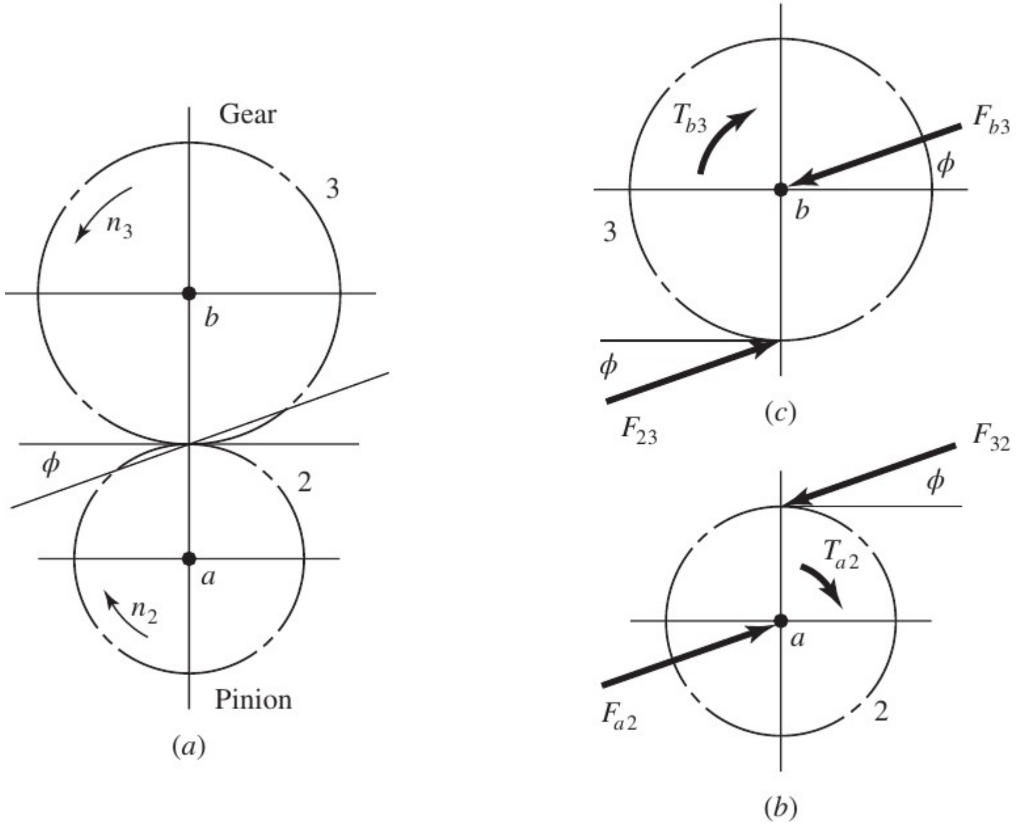
| Item | Symbol/Formula |
|---|---|
| Module | $m$ (**choose**) |
| Pressure angle | $\boldsymbol{\phi}$ (**choose**) |
| Number of teeth | $N_p$, $N_g$ (**given**) |
| Center distance | a=(dp+dg)/2 (calculate) |
| Pitch diameter | dp = mNp,dg = mNg (calculate) |
| Base diameter | dbp = dp cos$\boldsymbol{\phi}$ dbg = dg cos$\boldsymbol{\phi}$ (calculate) |
| Addendum | $m$ |
| Tooth depth | 1.25 $m$ |
| Tip diameter | dp+2$m$, dg+2$m$ |
| Root diameter | dp-2.5m,dg-2.5m |

# Force Analysis – Spur Gears

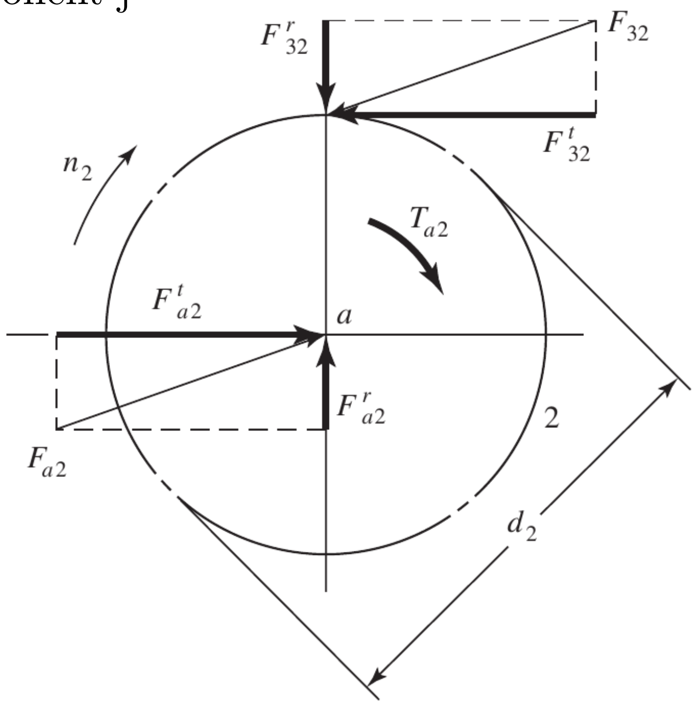
Input gear: 2 followed by 3,4,5, ...

Shaft: a, b, c, ...

Nomenclature: $F_{ij}$ : Force exerted by component i on component j



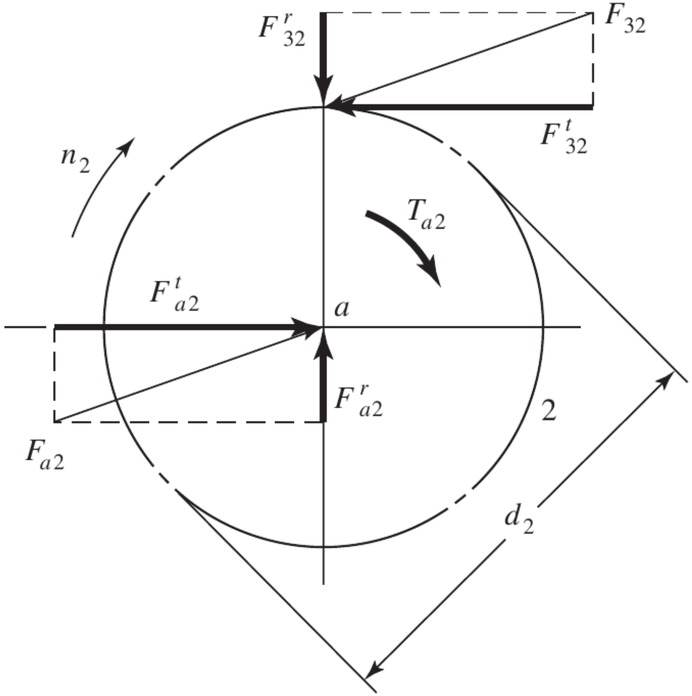

FBD for a gear pair




Force Resolution - Pinion

# Force Analysis – Spur Gears - Continued

б




Transmitted load $W_t = F^t_{32}$ (this is the useful component)

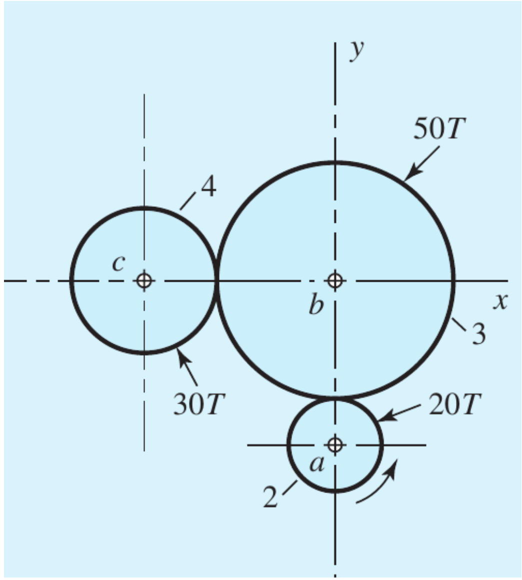
Torque transmitted: $T = W_t d_2/2$

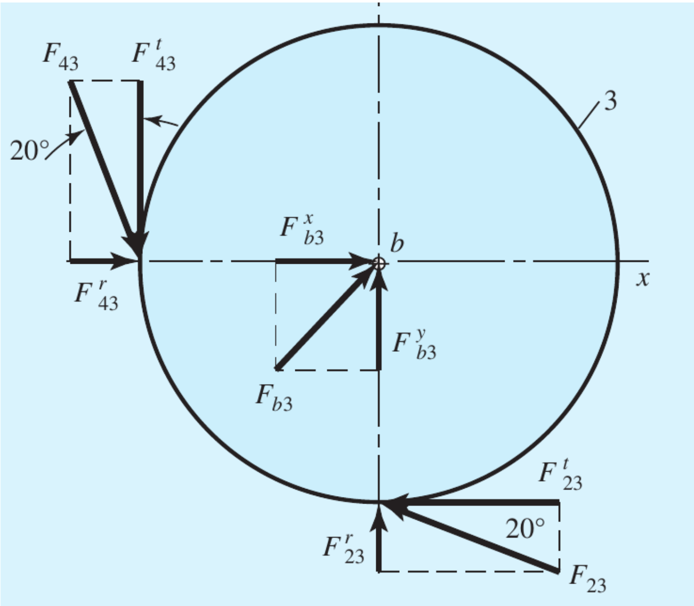
Power transmitted $P = T$
(efficiency of spur gears is ~ 98%)

Slip line velocity V = $(d/2)$

# Force Analysis – Spur Gears - Problem

Pinion 2 in Figure runs at 1750 rev/min and transmits 2.5 kW to idler gear 3. The teeth are cut on the 20° full-depth system and have a module of m = 2.5 mm. Draw a free-body diagram of gear 3 and show all the forces that act upon it.




(b)



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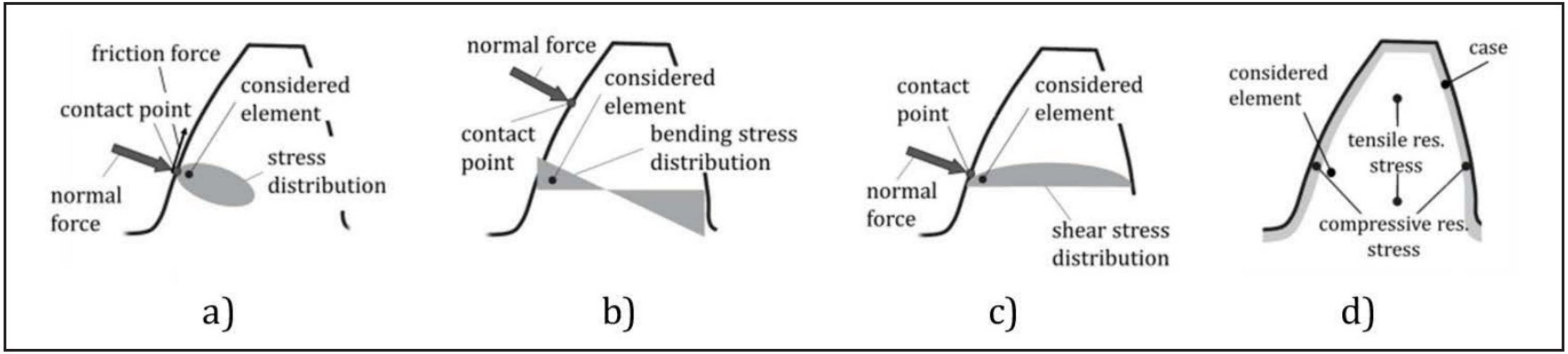
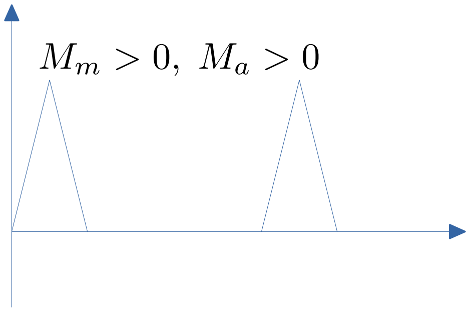
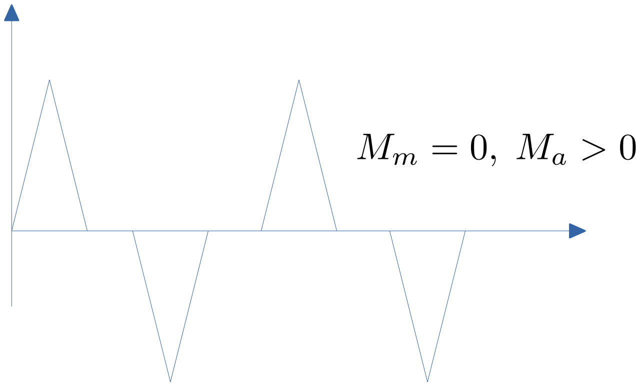
# Gear Tooth: Types of Stresses




Contact Stress Bending Stress Shear Stress Residual Stress

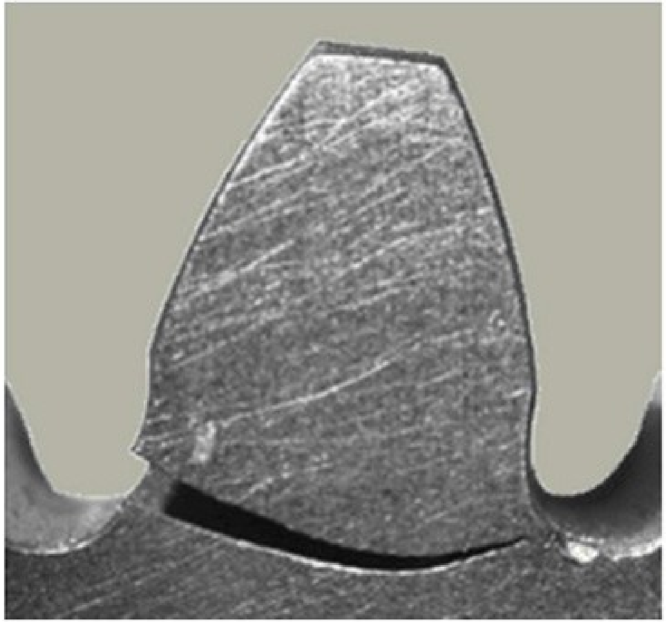



Repeated moment on a nonidler tooth

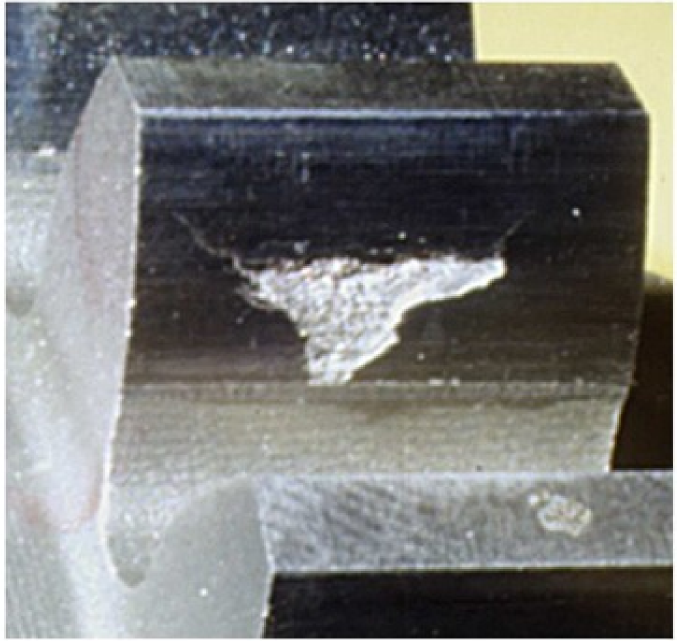



Reversed moment on an idler tooth

# Gear Tooth: Common Failure Modes



Tooth Root Breakage
(Bending Fatigue Fracture)



Pitting
(Surface Fatigue Fracture)

Pitting – the fracture and separation of small pieces of materials from the surface – form of surface fatigue

https://www.geartechnology.com/articles/22175-tooth-flank-fracture-basic-principles-and-calculation-model-for-a-sub-surface-initiated-fatigue-failure-mode-of-case-hardened-gears

# Gear Material

**Steel**: The most widely used material for gears.

- Low Carbon Steel: Offers good machinability and is often used for gears that do not require high strength.
- Medium Carbon Steel: Provides a balance of strength and toughness, suitable for many gear applications.
- High Carbon Steel: Known for its high strength and wear resistance, used in heavy-duty gears.
- Alloy Steel: Enhanced with elements like chromium, nickel, and molybdenum to improve strength, toughness, and wear resistance.

**Cast Iron**: Good wear resistance and damping properties, used for gears that operate under heavy loads and low speeds.

**Non-Ferrous Alloys**: Includes materials like bronze and brass, which offer good corrosion resistance and are often used in worm gears and other applications where lubrication is critical.

**Powder Metallurgy Materials**: Used for producing gears with complex shapes and fine details, offering good strength and wear resistance.

**Plastics**: Commonly used in applications where noise reduction and lightweight are important. Thermoplastics like nylon and acetal.

Each material has its own set of properties that make it suitable for different gear applications.
The choice of material depends on factors like load, speed, operating environment, and cost considerations.

# AGMA Bending Stress Equation

Assumptions

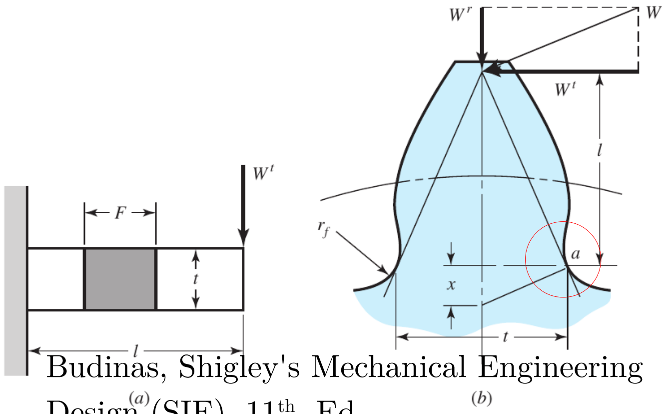
- The contact ratio is between 1 and 2.
- There is no interference between the tips and the root fillets of the mating teeth and no undercutting of the
- teeth above the theoretical start of the active profile.
- No teeth are pointed.
- There is nonzero backlash.
- The root fillets are standard, assumed smooth, and produced by a generating process.
- The friction force effects are neglected.
- Valid for external tooth only

$$\sigma_b = \frac{F_t}{FmJ}\frac{K_a K_m}{K_v} K_s K_B K_I$$

$F_t$ - tangential load, $F$ – face width, $m$ - module

Based on Lewis's formula – Lewis developed his formula in 1892 and is based on considering the loaded tooth as a cantilever beam

Lewis's formula neglects the stress concentration effect at the root

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# AGMA Bending Stress Equation

$$\sigma_b = \frac{F_t}{FmJ} \frac{K_a K_m}{K_v} K_s K_B K_I$$

- Bending strength geometry factor $J$ – vary with number of teeth on the pinion and the gear (AGMA standard) (different for pinion and gear)
- Dynamic factor $K_v$ – attempts to account for the internally generated vibration loads from tooth-tooth impacts induced by nonconjugate meshing of the gear teeth. Depends on the accuracy to which the gears are machined (estimate it from the standards)
- Load distribution factor $K_m$ – takes into account the fact that the transmitted load will be distributed unevenly across the face of the tooth
- Application factor $K_a$ – takes into account the non-uniform nature of the transmitted load (shock loading)
- Size factor $K_S$ – similar to the size factor used in fatigue calculations (larger components have lower strength). As per AGMA recommendations normally $K_S = 1$.
- Rim thickness factor $K_B$ – applicable in situations in which the gear is made with rim and spokes rather than a solid disk. For solid gears $K_B = 1$
- Idler factor $K_I$ - takes into account the fact that an idler gear is subject to more load cycles per unit time as compared to a non-idler gear. $K_I = 1.42$ for idler and $K_I = 1.0$ for non-idler

# AGMA Bending Stress Equation

**Table 12-8 AGMA Bending Geometry Factor J for 20°, Full-Depth Teeth with Tip Loading**

Pinion teeth

| Gear teeth | 12 | | 14 | | 17 | | 21 | | 26 | | 35 | | 55 | | 135 | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | P | G | P | G | P | G | P | G | P | G | P | G | P | G | P | G |
| 12 | U | U | | | | | | | | | | | | | | |
| 14 | U | U | U | U | | | | | | | | | | | | |
| 17 | U | U | U | U | U | U | | | | | | | | | | |
| 21 | U | U | U | U | U | U | 0.24 | 0.24 | | | | | | | | |
| 26 | U | U | U | U | U | U | 0.24 | 0.25 | 0.25 | 0.25 | | | | | | |
| 35 | U | U | U | U | U | U | 0.24 | 0.26 | 0.25 | 0.26 | 0.26 | 0.26 | | | | |
| 55 | U | U | U | U | U | U | 0.24 | 0.28 | 0.25 | 0.28 | 0.26 | 0.28 | 0.28 | 0.28 | | |
| 135 | U | U | U | U | U | U | 0.24 | 0.29 | 0.25 | 0.29 | 0.26 | 0.29 | 0.28 | 0.29 | 0.29 | 0.29 |

**Table 12-9 AGMA Bending Geometry Factor J for 20°, Full-Depth Teeth with HPSTC Loading**

Pinion teeth

| Gear teeth | 12 | | 14 | | 17 | | 21 | | 26 | | 35 | | 55 | | 135 | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | P | G | P | G | P | G | P | G | P | G | P | G | P | G | P | G |
| 12 | U | U | | | | | | | | | | | | | | |
| 14 | U | U | U | U | | | | | | | | | | | | |
| 17 | U | U | U | U | U | U | | | | | | | | | | |
| 21 | U | U | U | U | U | U | 0.33 | 0.33 | | | | | | | | |
| 26 | U | U | U | U | U | U | 0.33 | 0.35 | 0.35 | 0.35 | | | | | | |
| 35 | U | U | U | U | U | U | 0.34 | 0.37 | 0.36 | 0.38 | 0.39 | 0.39 | | | | |
| 55 | U | U | U | U | U | U | 0.34 | 0.40 | 0.37 | 0.41 | 0.40 | 0.42 | 0.43 | 0.43 | | |
| 135 | U | U | U | U | U | U | 0.35 | 0.43 | 0.38 | 0.44 | 0.41 | 0.45 | 0.45 | 0.47 | 0.49 | 0.49 |

HPSTC - highest point of single-tooth contact

For contact ratio > 1 and gear with good Accuracy, it lies somewhere below the tip

U - undercutting

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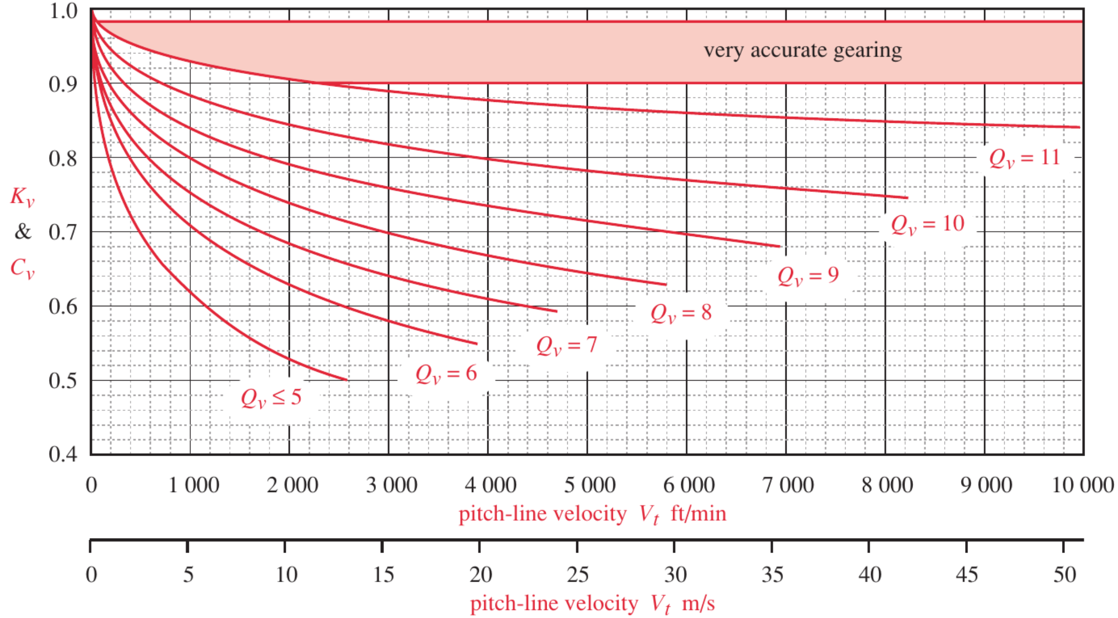
# AGMA Bending Stress Equation

$$\sigma_b = \frac{F_t}{FmJ}\frac{K_a K_m}{K_v} K_s K_B K_I$$

**Table 12-16**
Load Distribution Factors $K_m$

| Face Width in (mm) | $K_m$ |
|---|---|
| <2 (50) | 1.6 |
| 6 (150) | 1.7 |
| 9 (250) | 1.8 |
| ≥20 (500) | 2.0 |

## Dynamic factor $K_v$




The higher the quality number $Q_v$, the closer the actual geometry to what is specified

Depends on the gear manufacturing process and the specified tolerances

Tighter the tolerances, higher is the quality number

**Table 12-17 Application Factors $K_a$**

| Driving Machine | Driven Machine: Uniform | Driven Machine: Moderate Shock | Driven Machine: Heavy Shock |
|---|---|---|---|
| Uniform (Electric motor, turbine) | 1.00 | 1.25 | 1.75 or higher |
| Light Shock (Multicylinder engine) | 1.25 | 1.50 | 2.00 or higher |
| Medium Shock (Single-cylinder engine) | 1.50 | 1.75 | 2.25 or higher |

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# Gear Quality Number

The quality number indicates the the accuracy of tooth shape and placement.
The four main parameters accounted for are:

The **tooth lead or tooth alignment** criterion applies to spur and helical-type gearing, and measures the variation between the specified lead (or helix angle) and the lead of the produced gear.

**Involute profile variation** is the difference between the specified profile and the measured profile of the tooth.

**Pitch variation or spacing variation** is the difference between the specified tooth location and the actual tooth location around the circumference of the gear.

**Radial runout** refers to the disparity in radial position of teeth on a gear – the variation in tooth distances from the center of rotation.

https://www.machinedesign.com/automation-iiot/article/21834626/gear-quality-what-its-all-about

# AGMA Pitting Resistance Equation

$$\sigma_c = C_p \sqrt{\frac{F_t}{FId}\frac{C_a C_m}{C_v} C_s C_f}$$

Feature of contact stresses

$$\sigma \propto \sqrt{F}$$

$F_t$ – tangential force

$d$ – pitch circle diameter of the smaller of the two gears in mesh

$F$ – face width

$I$ – surface geometry factor accounts for the radii of curvature of the gear teeth and the pressure angle. Calculated for a gear pair in a mesh

$C_p$ – elastic coefficient that accounts for the differences in the gear and pinion material constants

$C_f$ - surface finish factor that accounts for the unusually rough finishes on the gear teeth.

$C_f = 1$ for gears made by conventional methods.

$C_a = K_a$ - Application factor

$C_m = K_m$ - Load distribution factor

$C_v = K_v$ - Dynamic factor

$C_s = K_s$ – Size factor

# AGMA Pitting Resistance Equation

**Table 12-18 AGMA Elastic Coefficient $C_p$ in Units of $[psi]^{0.5}$ ( $[MPa]^{0.5}$ )*†**

| Pinion Material | $E_p$ psi (MPa) | Gear Material: Steel | Malleable Iron | Nodular Iron | Cast Iron | Aluminum Bronze | Tin Bronze |
|---|---|---|---|---|---|---|---|
| Steel | 30*E*6 (2*E*5) | 2 300 (191) | 2 180 (181) | 2 160 (179) | 2 100 (174) | 1 950 (162) | 1 900 (158) |
| Malleable Iron | 25*E*6 (1.7*E*5) | 2 180 (181) | 2 090 (174) | 2 070 (172) | 2 020 (168) | 1 900 (158) | 1 850 (154) |
| Nodular Iron | 24*E*6 (1.7*E*5) | 2 160 (179) | 2 070 (172) | 2 050 (170) | 2 000 (166) | 1 880 (156) | 1 830 (152) |
| Cast Iron | 22*E*6 (1.5*E*5) | 2 100 (174) | 2 020 (168) | 2 000 (166) | 1 960 (163) | 1 850 (154) | 1 800 (149) |
| Aluminum Bronze | 17.5*E*6 (1.2*E*5) | 1 950 (162) | 1 900 (158) | 1 880 (156) | 1 850 (154) | 1 750 (145) | 1 700 (141) |
| Tin Bronze | 16*E*6 (1.1*E*5) | 1 900 (158) | 1 850 (154) | 1 830 (152) | 1 800 (149) | 1 700 (141) | 1 650 (137) |

$$C_p = \sqrt{\frac{1}{\pi\left(\left(\frac{1-\nu_p^2}{E_p}\right)+\left(\frac{1-\nu_g^2}{E_g}\right)\right)}}$$

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# AGMA Bending-Fatigue Strength Calculation for Gear Materials

$\sigma_b$ – Tooth bending stress (pinion/gear)

$S'_{fb}$– Uncorrected (published) bending-fatigue strength for pinion/gear material

(Strength data is stated for $10^7$ cycles of repeated stress and 99% reliability)

$S_{fb}$– Corrected bending-fatigue strength for pinion/gear material

$$S_{fb} = \frac{K_L}{K_T K_R} S'_{fb}$$

$K_L$ – Life factor. As test data is for $10^7$ cycles, a shorter or longer life cycle requires modification of the bending-fatigue strength

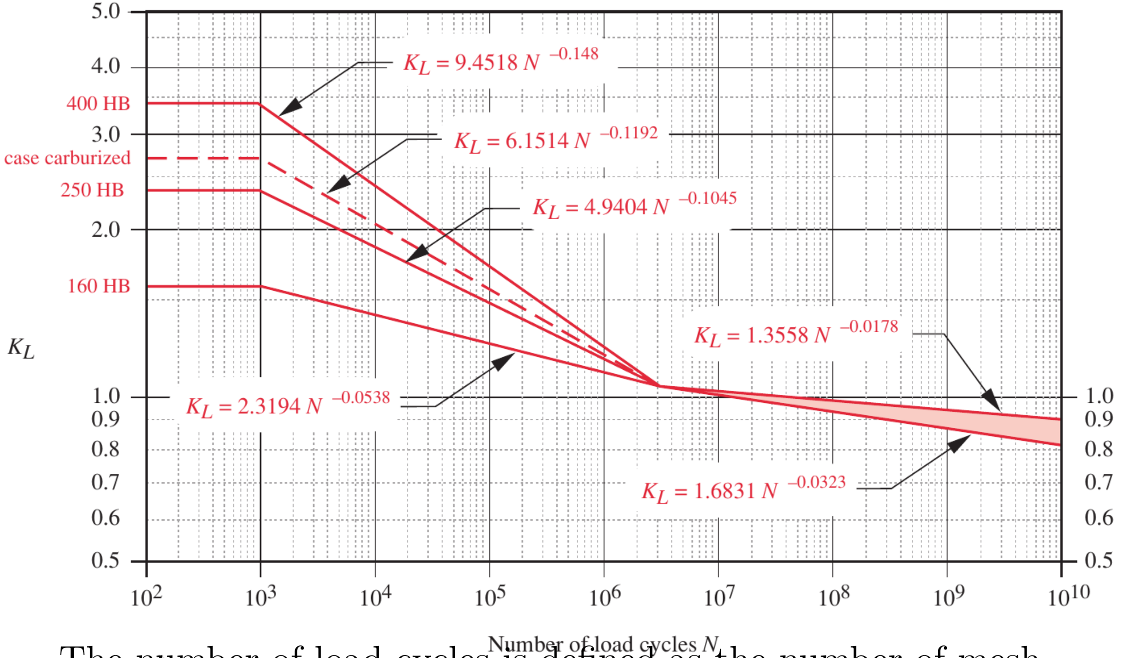
$K_T$ – Temperature factor, $K_T$ =1 if the gear temperature is less than 120$^{\circ}$

$K_R$ – Takes into account reliability other than 99%

Calculate the the following FOS

$$N_{b_{pinion}} = \frac{S_{fb_{pinion}}}{\sigma_{b_{pinion}}} \qquad N_{b_{gear}} = \frac{S_{fb_{gear}}}{\sigma_{b_{gear}}}$$

# AGMA Bending-Fatigue Strength Calculation for Gear Materials




The number of load cycles is defined as the number of mesh contacts, under load, of the gear tooth being analyzed

**Table 12-19**
AGMA Factor $K_R$

| Reliability % | $K_R$ |
|---|---|
| 90 | 0.85 |
| 99 | 1.00 |
| 99.9 | 1.25 |
| 99.99 | 1.50 |

**Table 12-20** AGMA Bending-Fatigue Strengths $S_{fb}'$ for a Selection of Gear Materials*

| Material | AGMA Class | Material Designation | Heat Treatment | Minimum Surface Hardness | Bending-Fatigue Strength psi x $10^3$ | Bending-Fatigue Strength MPa |
|---|---|---|---|---|---|---|
| **Steel** | A1–A5 | | Through hardened | ≤ 180 HB | 25–33 | 170–230 |
| | | | Through hardened | 240 HB | 31–41 | 210–280 |
| | | | Through hardened | 300 HB | 36–47 | 250–325 |
| | | | Through hardened | 360 HB | 40–52 | 280–360 |
| | | | Through hardened | 400 HB | 42–56 | 290–390 |
| | | | Flame or induction hardened | Type A pattern 50–54 HRC | 45–55 | 310–380 |
| | | | Flame or induction hardened | Type B pattern | 22 | 150 |
| | | | Carburized and case hardened | 55–64 HRC | 55–75 | 380–520 |
| | | AISI 4140 | Nitrided | 84.6 HR15N† | 34–45 | 230–310 |
| | | AISI 4340 | Nitrided | 83.5 HR15N | 36–47 | 250–325 |
| | | Nitralloy 135M | Nitrided | 90.0 HR15N | 38–48 | 260–330 |
| | | Nitralloy N | Nitrided | 90.0 HR15N | 40–50 | 280–345 |
| | | 2.5% Chrome | Nitrided | 87.5–90.0 15N | 55–65 | 380–450 |
| **Cast iron** | 20 | Class 20 | As cast | | 5 | 35 |
| | 30 | Class 30 | As cast | 175 HB | 8 | 69 |
| | 40 | Class 40 | As cast | 200 HB | 13 | 90 |
| **Nodular (ductile) iron** | A-7-a | 60-40-18 | Annealed | 140 HB | 22–33 | 150–230 |
| | A-7-c | 80-55-06 | Quenched and tempered | 180 HB | 22–33 | 150–230 |
| | A-7-d | 100-70-03 | Quenched and tempered | 230 HB | 27–40 | 180–280 |
| | A-7-e | 120-90-02 | Quenched and tempered | 230 HB | 27–40 | 180–280 |
| **Malleable iron (pearlitic)** | A-8-c | 45007 | | 165 HB | 10 | 70 |
| | A-8-e | 50005 | | 180 HB | 13 | 90 |
| | A-8-f | 53007 | | 195 HB | 16 | 110 |
| | A-8-i | 80002 | | 240 HB | 21 | 145 |
| **Bronze** | Bronze 2 | AGMA 2C | Sand cast | 40 ksi min tensile strength | 5.7 | 40 |
| | Al/Br 3 | ASTM B-148 78 alloy 954 | Heat treated | 90 ksi min tensile strength | 23.6 | 160 |

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# AGMA Surface-Fatigue Strength Calculation for Gear Materials

$\sigma_c$ – Tooth surface stress (pinion)

$S'_{fc}$– Uncorrected surface-fatigue strength for pinion material

(Strength data is stated for $10^7$ cycles of repeated stress and 99% reliability)

$S_{fc}$– Corrected surface-fatigue strength for pinion material

$$S_{fc} = \frac{C_L C_H}{C_T C_R} S'_{fc}$$

$C_L$ – surface life factor. As test data is for $10^7$ cycles, a shorter or longer life cycle requires modification of the surface-fatigue strength

$C_T$ – Temperature factor, $C_T = K_T$

$C_R$ – Takes into account reliability other than 99%, $C_T = K_T$

$C_H$ – function of the gear ratio and the relative hardness of pinion and gears, $C_H = 1$ if made from the same material

Calculate the the following FOS $N_{c_{pinion}} = \left( \frac{S_{fc_{pinion}}}{\sigma_{c_{pinion}}} \right)^2$ $\sigma \propto \sqrt{F}$

# AGMA Surface-Fatigue Strength Calculation for Gear Materials

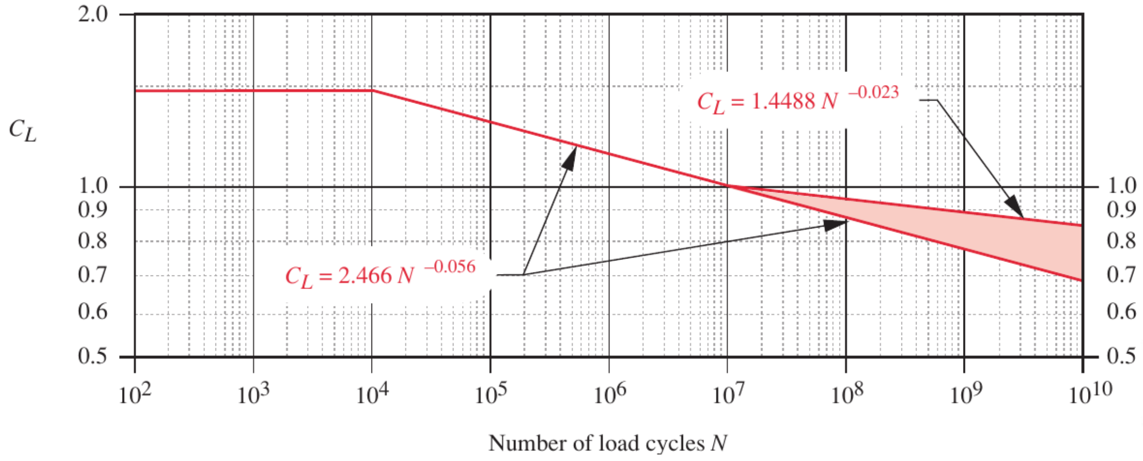


**Table 12-21 AGMA Surface-Fatigue Strengths $S_{fc}'$ for a Selection of Gear Materials***

| Material | AGMA Class | Material Designation | Heat Treatment | Minimum Surface Hardness | Surface-Fatigue Strength psi x $10^3$ | Surface-Fatigue Strength MPa |
|---|---|---|---|---|---|---|
| **Steel** | A1–A5 | | Through hardened | ≤ 180 HB | 85–95 | 590–660 |
| | | | Through hardened | 240 HB | 105–115 | 720–790 |
| | | | Through hardened | 300 HB | 120–135 | 830–930 |
| | | | Through hardened | 360 HB | 145–160 | 1000–1100 |
| | | | Through hardened | 400 HB | 155–170 | 1100–1200 |
| | | | Flame or induction hardened | 50 HRC | 170–190 | 1200–1300 |
| | | | Flame or induction hardened | 54 HRC | 175–195 | 1200–1300 |
| | | | Carburized and case hardened | 55–64 HRC | 180–225 | 1250–1300 |
| | | AISI 4140 | Nitrided | 84.6 HR15N† | 155–180 | 1100–1250 |
| | | AISI 4340 | Nitrided | 83.5 HR15N | 150–175 | 1050–1200 |
| | | Nitralloy 135M | Nitrided | 90.0 HR15N | 170–195 | 1170–1350 |
| | | Nitralloy N | Nitrided | 90.0 HR15N | 195–205 | 1340–1410 |
| | | 2.5% Chrome | Nitrided | 87.5 HR15N | 155–172 | 1100–1200 |
| | | 2.5% Chrome | Nitrided | 90.0 HR15N | 192–216 | 1300–1500 |
| **Cast iron** | 20 | Class 20 | As cast | | 50–60 | 340–410 |
| | 30 | Class 30 | As cast | 175 HB | 65–70 | 450–520 |
| | 40 | Class 40 | As cast | 200 HB | 75–85 | 520–590 |
| **Nodular (ductile) iron** | A-7-a | 60-40-18 | Annealed | 140 HB | 77–92 | 530–630 |
| | A-7-c | 80-55-06 | Quenched and tempered | 180 HB | 77–92 | 530–630 |
| | A-7-d | 100-70-03 | Quenched and tempered | 230 HB | 92–112 | 630–770 |
| | A-7-e | 120-90-02 | Quenched and tempered | 230 HB | 103–126 | 710–870 |
| **Malleable iron (pearlitic)** | A-8-c | 45007 | | 165 HB | 72 | 500 |
| | A-8-e | 50005 | | 180 HB | 78 | 540 |
| | A-8-f | 53007 | | 195 HB | 83 | 570 |
| | A-8-i | 80002 | | 240 HB | 94 | 650 |
| **Bronze** | Bronze 2 | AGMA 2C | Sand cast | 40 ksi min tensile strength | 30 | 450 |
| | Al/Br 3 | ASTM B-148 78 alloy 954 | Heat–treated | 90 ksi min tensile strength | 65 | 450 |

END