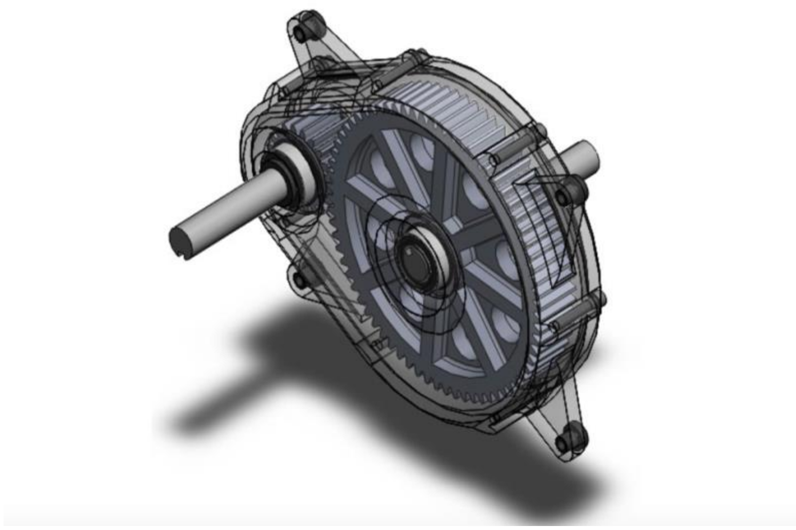
One - Stage Reduction Gearbox Technical Report



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Title and Abstract

Executive Summary

The designing of a single reduction gearbox was focused when conducting the implementation of the single reduction gear box project. Our group of ten members focused on designing a single reduction spur gearbox. Which includes a series of gears, shafts, bearings, keys, housing and mechanical keys and shafts that meet to the specification of our team. The objective of our team was to create a gearbox design that can handle 25 HP and an input speed of 1750 rpm and an output speed of 500 rpm.

The design of our gearbox was executed by the implementation of using Microsoft excel spreadsheets to create equations for each design component of the gearbox. The initial design of the gearbox was designed using solid works and the proper fixtures between the contact parts of housing and the connectors of bearing loads were executed through the simulation process.

Introduction

The function of a gearbox is to transfer a portion of energy from one device into another device by increasing its torque. When creating our gearbox design our group focused on many key aspects of the gearbox.

The gear attachments were carefully considered when designing our solid works as the pinion and gears were fixed onto the shafts. When designing our gear components, the necessary factors of safety of the bending and wearing are calculated and determined to ensure a proper rotation of the gears.

A major design component in our gearbox was the creation of the shafts. When designing our shafts, the size of the shaft and the arrangement of the shaft assemblies were determined to judge location of the bearings. The creation of the shaft key in the gearbox was very critical as the input shaft key had to fail the ultimate shear strength before the gears or the shafts began to yield. Our group calculated the yielding and shear failure safety factors of the shaft keys. This calculation process ensures that the failure will occur in the weakest key initially as this prevents the rest of the gearbox from overloads.

The gearbox housing was another major component towards the designing process of our gear box. As the gearbox housing provided the necessary support of the bearing to ensure that all components of the gearbox are sealed and are properly fixed in place. As in our designing creation of our gearbox there were two mounted points assembled in the gearbox.

By using the software of solid works our group was able to properly design and assemble part modeling of our gear box design. Initially our group designed drawings of the shaft in order to get an outlook of the gearbox components. When creating our design and drawing a judgment of the necessary tolerances and dimensions were heavily measured and considered when developing our gearbox. Each of our drawings displayed clear information on the sizing of the shafts and the gears. With proper dimensions and drawings presented our group was able to efficiently start on the design modelling of solid works and the simulation process by assembling the necessary parts into the gearbox design.

The delivered power will be in the range 25 HP, and the input speed used for our gearbox is 1750rpm and the output speed is 500 rpm. The input and output shafts designed in our gearbox were 100mm long and were displayed on the slots of the outside of the gearbox.

The gear pressure angle for our design specification was at an angle of 20°, in our gearbox design a helix angle was not used. The design life for the gears and bearings were 20,000 hours of life.

In our gear box design two keys were developed using solid works software the formation of the gear and pinions were developed with the use of a yielding safety factor of ny 2. The fatigue safety factors for both the gears and pinions are 1.2. The bending of our pinion is 1.2 also, yet for the gear the bending was calculated to be 1.5.

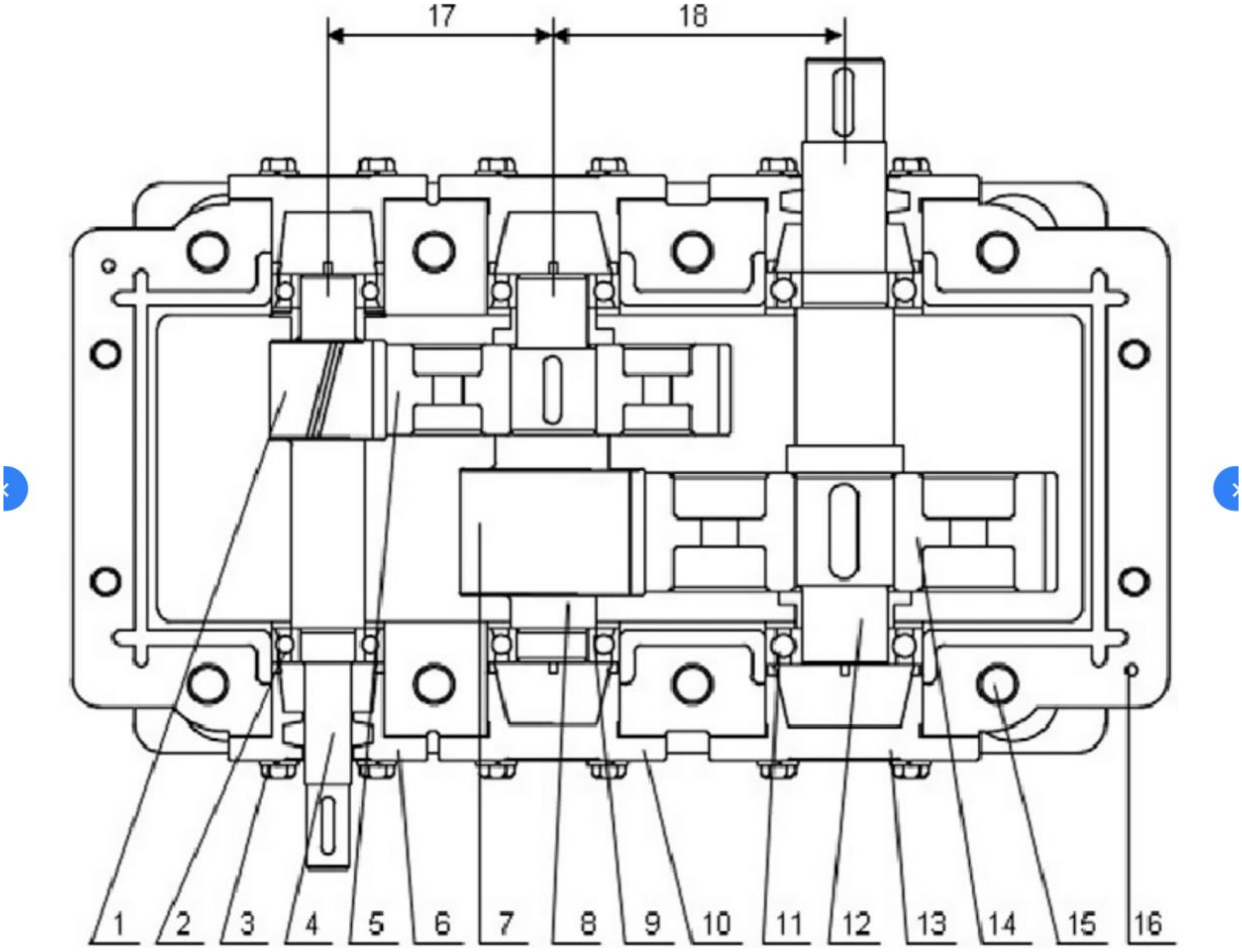
In our gearbox design our group went with a diametral pitch of 6, with the data provided in the section 14-4 of the Shigley's Mechanical Engineering Design eBook. The required measurements of the development of the gears and pinions used in the gearbox were collected from the figure and tables of 14-4 section.

In our gearbox design we were able to precisely mark the axial locations of the gears on the shaft. In the gearbox the motor shaft was required to couple with the input shaft for the proper assembly the input shaft measurements were taken to be extended over the housing component for a proper assembly connection with the motor shaft. For the output shafts of the gearbox the mates had to have a proper connection with the drive shaft in the propeller. With the correct tolerance and dimension used in the solid works software, our proper assembly of the gearbox was conducted. A design reliability of 0.999 was also used through the development of our gearbox design

When developing the gears component of our gearbox we ensured that the number of teeth in the gear stay at the limit of 75 to ensure proper rotational components can be established within the gearbox.

The diameters of our shafts used in the gearbox were 1 inch and the length of each shaft used were less than 10 inches to ensure that the shaft would fit into the gearbox. Each bearing used in the gearbox had a 95% reliability with a design life component of 20,000 hours.

The gears in the gearbox were at commercial quality of 99.9 reliability and the centre of distance marked between each gear had a separation distance of 10 inches in the gearbox from its center location.



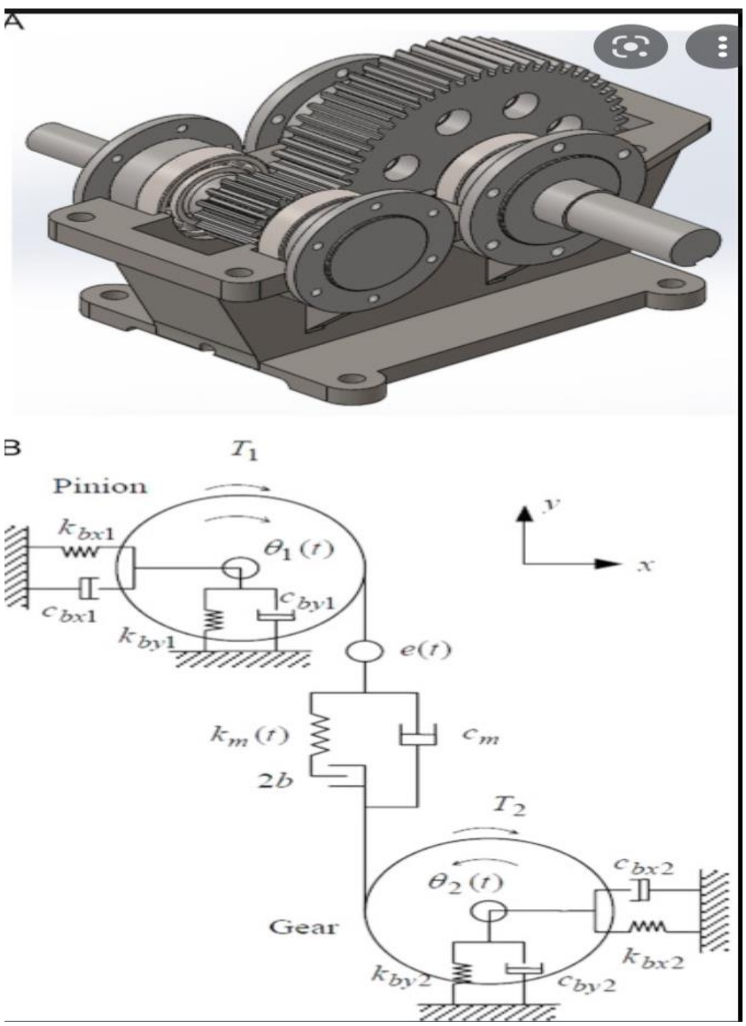
An example of gearbox layout

1. pinion; 2. bearing; 3. connecting bolt for bearing cover; 4. high speed shaft; 5. gear; 6. bearing cover; 7. pinion; 8. middle shaft; 9. bearing; 10. bearing cover; 11. bearing; 12. low speed shaft; 13. bearing cover; 14. gear; 15. housing connecting bolt; 16. guide pin; 17. high-speed stage center distance; 18. low-speed stage center distance.

Synthesis and Decision on General Layout

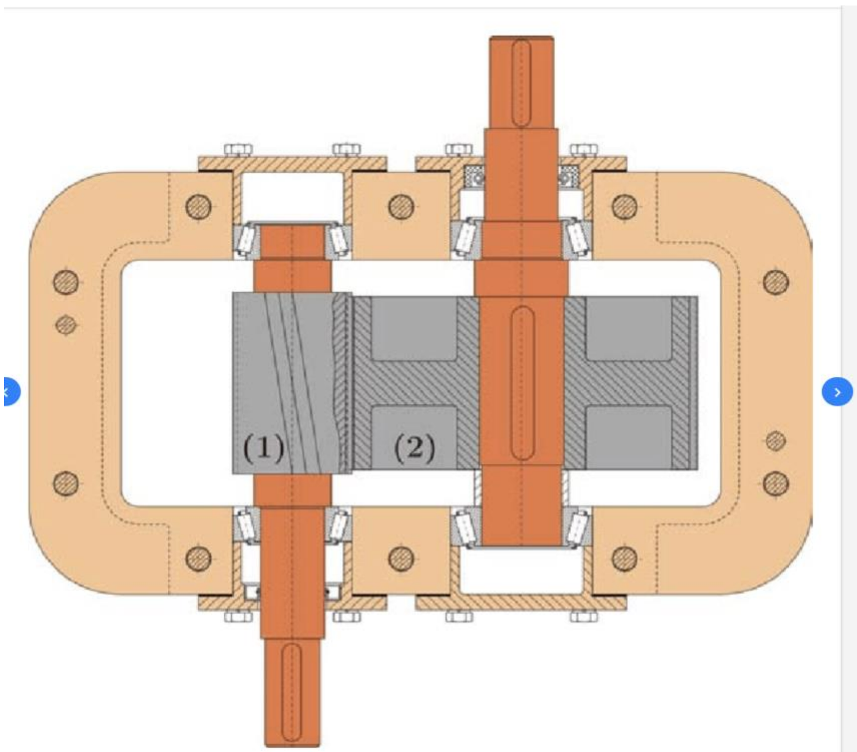
Single Stage Spur Gear

When designing a one-stage speed reduction gearbox there were two various design choices that were presented. The option of using spur gear or helical gears was a critical design and ultimately designing our gear box. The differences and benefits of using one gear over another required our team members to gain a proper understanding of the two gears. When using spear gears in our gearbox there teeth are in a straight manner and the teeth are parallel to the axis of the gear. The spur gears when compared to the helical gears are very loud gears and the transmission of the gears are not the smoothers. The spur gears tend to produce a higher efficient power and this was one of the reasons our group tended to lean towards developing a gearbox that contained the use of spur gears. In the gearbox design the spur gear when in use at the higher power loads will vibrate and cause the noise of the gearbox to be heard in a loud manner when higher speeds are produced.



Single Stage Helical Gear

The second option was using helical gears in our gearbox , our group initially gained a proper understanding of spur gears and their implication towards a gearbox. When studying the effects of helical gears instead of a gear box we were able to compare that in helical gears the teeth are arranged in an angle to the gears axis. The helical gears provided a more smoother and quieter operation compared to the spur gears in the gearbox design. Helical gears inside a gearbox have a higher load carrying capacity and thus creating a greater tooth strength for the gears teeth. The main difference in design layout between the two gears is that helical gears transmit power in between the parallel and the non parallel shafts. Spur gears only had the ability to transmit power to the parallel shafts. After analyzing and studying the design layout between the two gears our group tended to lean towards designing a single stage spur gearbox due to the gearbox's ability to produce a high efficient power.



The sketch of the single-stage helical gear unit.

Design Procedure and Sample Calculations

Gear Calculations:

Design Procedure - First Iteration Parameters:

Pinion Teeth Number (Np) = 20
Gear Teeth Number (Ng) = 70
Material used: Gray Cast Iron Class 40
Diametral Pitch (Pd) = 4
Pressure Angle = 20°
Input Power (P) = 25 HP
Input Speed (n) = 1750 rpm
Output Speed = 500 rpm
Life (N) = 20,000 hours
Reliability = 99.9%
Gear Ratio (mg) = 3.5

Pinion Pitch Diameter:

$$dp = Np/Pd = 5$$

Gear Pitch Diameter:

$$dg = Ng/Pd = 17.5$$

Pitch Line Speed:

$$V = (\pi * Dp * n)/12 = 2290.74$$

Transmitted Load:

$$Wt = (33{,}000 * P)/V = 360.14$$

Face Width Guidelines:

$$Fmin = 8/Pd = 2\ inch$$
$$Fnom = 12/Pd = 3\ inch$$
$$Fmax = 16/Pd = 4 \text{ inch}$$

Face Width/Pinion Diameter Ratio:

$$F/Dp = 0.6$$

Elastic Coefficient: 1554.116

$$vp = 0.211$$
$$Ep = 14.5\ Mpsi$$

Elastic Coefficient:

The elastic coefficient was calculated, using the values found in Table A-5.

$$Cp = (\ 1\ \pi * (1 - vp\text{^}2\ )/Ep + (1 - vg\text{^}2\ )/Eg\ )\ 0.5$$

Poisson's Ratio for Gray Cast Iron: v = 0.211

Modulus of Elasticity for Gray Cast Iron: E = 14.5 $Mpsi$

Lewis Form Factor:

Based on the assumed number of pinion teeth, 20, and gear teeth, 70, table 14-2 is used to find the Lewis Form Factor for the pinion and gear.

Yp = 0.309 Yg = 0.4246

Bending Geometry Factor:

Using figure 14-6, the bending geometry factor for the pinion and the gear can be found.

Jp = 0.320 Jg = 0.405

Surface Strength Geometry Factor:

$$I = (cos(\phi)sin(\phi) * mg)/(2mn * (mg + 1)$$

$$I = 0.125$$

Load Distribution Factor:

$$Km = 1 + Cmc(CpfCpm + CmaCe)$$

$$Cpm = 1$$

$$Cma = A + BF + CF\ 2$$

Using Table 14-9, A, B and C can be found.

A = 0.127 B = 0.0158 C = 0.0001093

Cma = 0.158

Ce = 1

Km = 1.212

Size Factor:

$$Ks = 1.192((F\surd Y)/(Pd))\ 0.053$$

The pinion and gear are assumed to have constant thickness

$$Kbp = 1\ Kbg = 1$$

Dynamic Factor:

$$Kv = ((A + \sqrt{V})/A)\ B = 1.497$$

$$B = 0.25(12 - Qv)^{2/3} = 0.731$$

Hardness Ratio Factor:

$$CH = 1 + A'(mg - 1)$$

$$A' = 8.98(10^{-3})(HBP/HBG) - 8.29(10^{-3}) \qquad 1.2 \leq (HBP/HBG) \leq 1.7$$

The pinion and gear material are assumed to both be Gray Cast Iron Class 40, which means they have the same Brinell hardness value. Since $(HBP/HBG) = 1$, A'=0

$$CH = 1$$

Temperature Factor:

The temperature factor can be found assuming that the oil temperature in the gearbox will have a maximum value of 120 degrees.

$$KT = 1$$

Reliability Factor:

As per the design requirements, the gearboxe's reliability should be 99.9%. The reliability factor can be found using table 14-10.

$$KR = 1.25$$

Bending Stress Cycle Factor:

From figure 14-14, the formula for the bending stress cycle factor, for cycles over , is:

$$YNp = 0.841$$

$$Cycles\ Ngear = 2.1\text{x}10^9$$

$$YNg = 0.876$$

$$Cycles\ YNp = 6.0\text{x}10^8$$

Contact Stress Cycle Factor:

From figure 14-15, the formula for the contact stress cycle factor, for cycles over , is:

$$ZNp = 0.741$$

$$ZNg = 0.795$$

Using the gear design factors, the bending and contact stress values can be found.

$$(\sigma bending)\text{p} = Wt(KoKvKmKbp\ Ksp\ )(Pd/(FJp)\ (\sigma bending) = 3155 \text{ psi}$$

$$(\sigma bending)\text{g} = Wt(KoKvKmKbg\ Ksg\ ))(Pd/(FJg) = 2512 \text{ psi}$$

$$(\sigma contact)p = Cp(WtKoKvKsp\ KmCf\ /dpFI)\ 0.5 = 31{,}232 \text{ psi}$$

$$(\sigma contact)g = Cp(WtKoKvKsg\ KmCf\ /dpFI)\ 0.5 = 31{,}354 \text{ psi}$$

With the selected material of Gray Cast Iron class 40 for the pinion and gear, the "fully corrected" bending ($S't$) and contact stress ($S'c$) can be found. Using table 14-4, figure 14-2, table 14-7 and figure 14-5.

$$(S't)p = St(YNp\ /KTKR) = 7{,}030 \text{ psi}$$

$$(S't)g = St(YNg\ /KTKR) = 5{,}376 \text{ psi}$$

$$(S'c)p = Sc(ZNp\ CH/KTKR) = 63{,}204 \text{ psi}$$

$$(S'c)g = Sc(ZNg\ CH/KTKR) = 59{,}152 \text{ psi}$$

The bending and contact safety factors can be calculated for the pinion and gear, using the "fully corrected" numbers and bending and contact stress numbers calculated.

$$(SF)P = (S't\ )p/(\sigma bending) = 2.77$$

$$(SF)G = (S't\ )g/(\sigma bending) = 3.63$$

$$(SH)P = (S'c)p/(\sigma contact) = 1.52$$

$$(SH)G = (S'c)g/(\sigma contact) = 1.62$$

Shaft Calculations:

Tensile strength(room temperature) (ksi) Sut = 123 ksi
Yield strength (ksi) Suy = 94 ksi
Corrected endurance limit (ksi) Se' = 61.5 ksi
a = 2
b = -0.217

Torque (eq 18-1):

T pinion = 900 in lb   T gear= 3150in lb

$$T = 63000 \; x \; H \; / \; n_p$$

Transmitted load:

$$W_{t\,Gear} = W_{t\,Pinion} = 600.24 \; lb$$
Wt = $33000 \; x \; H \; / \; N_p$

Total force:

$$W_{Gear} = W_{Pinion} = 638.76 \; lb$$
$$W = \frac{W_t}{cos(2\pi/180)}$$

Moment max:

$$M_{max\,Gear} = M_{max\,Pinion} = 1437.22 \; lb \; in$$

Shaft Length (in): 9

Gear Width (in): 3
Bearing Width (mm): 18

$$Mmax = \frac{T_f \; x \; Shaft \; Length}{2}$$

Stresses

Bending moment alternating (lbf in):

$$M_{a\,Gear} = M_{a\,Pinion}$$
$M_a = M_{max} \times (w_b/L_s) = 1437.22 \times (0.354/4.500) = 113.17$ lbf in

Bending moment mean (lbf in):

$$M_{m\,Gear} = M_{m\,Pinion}$$
$M_m = 0$ lbf in

Torque alternating (lbf in):

$T_a = 0$ lbf in

Torque mean (lbf in):

$$T_{m\,Pinion} = 900\ lbf\ in \qquad\qquad T_{m\,Gear} = 0\ lbf\ in$$

Marin Factors

Surface factor (machined) T-6-2:

a = 2
b = -0.217

Surface Factor Eq. 6-19:

$$k_{a\,Gear} = k_{a\,Pinion}$$
$$\underline{k_a = a \times S_{ut}{}^{b} = 2 \times 123^{-0.217} = 0.704}$$

Shaft Diameter (in):

d = 48mm = 1.8897637795 in

Size Factor (rotating shaft), Eq. 6-20:

$$k_{a\,Gear} = k_{a\,Pinion}$$
$$\underline{k_a = 0.879 \times d^{-0.107} = 0.879 \times 1.889^{-0.107} = 0.821}$$

Loading Factor, Eq. 6-26:

$$k_{c\,Gear} = k_{c\,Pinion}$$
$$\underline{k_c = 1}$$

Temperature Factor:

$$k_{d\,Gear} = k_{d\,Pinion}$$
$$\underline{k_d = 1}$$

Reliability Factor (0.999) T-6-5:

$$k_{e\,Gear} = k_{e\,Pinion}$$
$$\underline{k_e = 0.753}$$

Modified Endurance Limit, Eq.6-18:

$$S_{e\,Gear} = S_{e\,Pinion}$$
$$S_e = ka \times kb \times kc \times kd \times ke \times Se' = 26.767 \text{ ksi}$$

Notch Sensitivity, q

Notch radius, r (in) : 0.02

Bending

Neuber constant:

$$\sqrt{a}_{\,Gear} = \sqrt{a}_{\,Pinion}$$

√a=0.246 - 3.08(10^(-3))S_ut+1.51(10^(-5)) 〖S_ut〗 ^2-2.67(10^(-8)) 〖S_ut〗 ^3 = 0.0458

$$q_{Gear} = q_{Pinion}$$

q=1/(1+√a/√r) = 0.7554

Torsion

Neuber constant:

$$\sqrt{a}_{Gear} = \sqrt{a}_{Pinion}$$

√a=0.190 - 2.51(10^(-3))S_ut+1.35(10^(-5)) 〖S_ut〗 ^2-2.67(10^(-8)) 〖S_ut〗 ^3 = 0.0358

$$q_{Gear} = q_{Pinion}$$

q=1/(1+√a/√r) = 0.7979

Fatigue Stress Concentration Factor

$$Gear = Pinion$$

D/d = 1.5/d = 0.794

r/d = 0.011

Stress Conc. Factor (Fig.A15-9):

$$k_{t\ Gear} = k_{t\ Pinion}$$

$$k_t = 1.45$$

Stress Conc. Factor (shear) (Fig.A15-8):

$$k_{ts\ Gear} = k_{ts\ Pinion}$$

$$k_{ts} = 1.15$$

Fatigue Stress Concentration Factor (Eq.6-32):

$$k_{f\ Gear} = k_{f\ Pinion}$$

$$k_f = 1 + q(k_t - 1) = 1.340$$

Fatigue Stress Concentration Factor (shear) (Eq. 6-32):

$$k_{fs\ Gear} = k_{fs\ Pinion}$$

$$k_{fs} = 1 + qs(k_{ts} - 1) = 1.120$$

Fatigue Factor of Safety

DE Goodman Safety Factor (Eq.7-7):

$$A = \sqrt{4\,(kf\ x\ Ma)^2 + 3(kfs\ x\ Ta)^2}$$

$$B = \sqrt{4\,(kf\ x\ Mam + 3(kfs\ x\ Tm)^2}$$

$$nf = \frac{\pi d^3}{16}\left(\frac{A}{Sc} + \frac{B}{Sut}\right)^{-1}$$

$$n_{f\ Gear} = 116.956 \qquad n_{f\ Pinion} = 51.924$$

Alternating Stress(kpsi) (Eq. 7-4):

$$\sigma_a' = [(\frac{32\ x\ kf\ x\ M_a}{\pi\ x\ d^3})^2 + 3(\frac{16\ x\ kfs\ x\ T_a}{\pi\ x\ d^3})^2]^{1/2}$$

$\sigma_{a\ Gear}' = 0.229\ kpsi$ $\sigma_{a\ Pinion}' = 228.866\ kpsi$

Mean Stress (kpsi) (Eq. 7-5):

$\sigma_{m\ Gear}' = 0.0\ kpsi$ $\sigma_{m\ Pinion}' = 16055.646\ kpsi$

Maximum Von Mises (kpsi) (Eq. 7-15):

$$\sigma_{max}' = [(\frac{32\ x\ kf\ x\ (M_m + M_a)}{\pi\ x\ d^3})^2 + 3(\frac{16\ x\ kfs\ x\ (T_m + T_a)}{\pi\ x\ d^3})^2]^{1/2}$$

$\sigma_{max\ Gear}' = 0.229\ kpsi$ $\sigma_{max\ Pinion}' = 1336.915$ kpsi

First Cycle Yielding (2.5 min) (Eq. 7-16):

$$nfy = \frac{Syt}{\sigma_{max}'}$$

$n_{fy\ Gear} = 410.721$ $n_{fy\ Pinion} = 70.311$

Key Calculations:

Horsepower (hp) =25
RPM_{Pinion} = 1750
RPM_{Gear} = 500
Shaft Diameter (in) = 1.772
S_{yt} of Shaft (ksi) = 94
S_{yt} of Hub (ksi) = 140
Desired Safety Factor = 2

Pinion Key Material:

Key Thickness (t) = 0.5
Key Height (H) = 0.375
S_{yt} of Pinion key (ksi) = 24

Gear Key Material:

Key Thickness (t) = 0.5
Key Height (H) = 0.375
S_{yt} of Gear key (ksi) = 32

Gear Torque:

$$T_g\ (lb\ in) = \frac{(HP)\ (63025)}{RPM_g} = 3151.3$$

Pinion Torque:

$$T_g\ (lb\ in) = \frac{(HP)\ (63025)}{RPM_p} = 900.4$$

Gear Force Fg (lb):

$$F_g = \frac{T_g}{(D_s/2)} = 3557.4$$

Pinion Force Fp (lb):

$$F_p = \frac{T_p}{(D_s/2)} = 1016.4$$

Ssy of Pinion Key (ksi):

$$S_{sy} = 0.577\ x\ S_{yt} = 13.8$$

Ssy of Gear Key (ksi):

$$S_{sy} = 0.577\ x\ S_{yt} = 18.5$$

Length for Shear (in):

$$L_{Shear} = \frac{F\ x\ n}{S_{yt}\ x\ 1000\ x\ t}$$

$L_{Shear\ Gear}$ = 0.771 $L_{Shear\ Pinion}$= 0.295

Key Length (in):

$$L_{Key} = \frac{2\ x\ F\ x\ n}{H\ x\ S_{yt}\ x\ 1000}$$

$L_{Key\ Gear}$ = 1.186 $L_{Key\ Pinion}$= 0.452

Shaft Length (in):

$$L_{Shaft} = \frac{2\ x\ F\ x\ n}{H\ x\ S_{yt}\ x\ 1000}$$

$L_{Shaft\ Gear}$= 0.404 $L_{Shaft\ Pinion}$= 0.115

Hub Length (in):

$$L_{Hub} = \frac{2\ x\ F\ x\ n}{H\ x\ S_{yt}\ x\ 1000}$$

$L_{Hub\ Gear}$=0.271 $L_{Hub\ Pinion}$= 0.077

Final Key Lengths:

MAX(Ls:Lh)

$L_{Final\ Gear}$= 1.186 $L_{Final\ Pinion}$= 0.452

Bearing Calculations:

Bore (mm) Table 11-2:

$Bore_{Pinion}$ = 25 30 35 40 45 $Bore_{Gear}$= 25 30 35 40 45

Radial Force (lbf):

$$F_{Radial} = \frac{W}{2}$$

$F_{Radial\ Pinion}$= 319.38 $F_{Radial\ Gear}$= 300.12

Axial Force (lbf):

$$F_{Axial\ Pinion} = F_{Axial\ Gear} = 0$$

Life (hours):

$$Life_{Hours\ Pinion} = Life_{Hours\ Gear} = 20000$$

RPM:

RPM_{Pinion}= 1750 RPM_{Gear}= 500

V (1 for internal rotation):

$$V_{Pinion} = V_{Gear} = 1$$

C10 (KN)Table 11-2:

$C_{10\ Pinion}$= 14, 19.5, 25.5, 30.7, 33.2 $C_{10\ Gear}$= 14, 19.5, 25.5, 30.7, 33.2

C0 (KN)Table 11-2:

$C_{0\ Pinion}$= 6.95, 10, 13.7, 16.6, 18.6 $C_{0\ Gear}$= 6.95, 10, 13.7, 16.6, 18.6

Fa/C0:

$$F_r/(C_o \times 0.224808942443 * 1000)$$

$\frac{F_a}{C_0} pinion$= 0.204, 0.142, 0.104, 0.086, 0.076 $\frac{F_a}{C_0} gear$= 0.192, 0.134, 0.097, 0.080, 0.072

e Table 11-1:

e_{Pinion}= 0.31, 0.28, 0.265, 0.26, 0.24 e_{Gear}= 0.31, 0.28, 0.265, 0.26, 0.24

Fa/V(Fr) (Eq 11-11a):

$$\frac{F_a}{V\ x\ F_r}$$

$$\frac{F_a}{V\ x\ F_r} pinion = \frac{F_a}{V\ x\ F_r} gear = 0$$

X1 Table 11-1:

$$X_{1\ Pinion} = X_{1\ Gear} = 1$$

Y1 Table 11-1:

$$Y_{1\ Pinion} = Y_{1\ Gear} = 1$$

Fe (lbf) eq 11-12:

$$(X1 \times V \times F_r) + (Y1\ \times F_r)$$

$F_{e\ Pinion}$= 319.382 $F_{e\ Gear}$= 300.121

a (3 for ball bearing):

$a_{\ Pinion}$= 3 a_{Gear}= 3

C required:

$(F_e \times$ 4.44822162825/1000)×(((Life×RPM×60)/(10^6))^(1/a))

$C_{\ Pinion}$= 18.19 $C_{\ Pinion}$= 11.26

Safety Factor:

$$C_{10}/C_{required}$$

SF_{Pinion}= 0.77, 1.07, 1.40, 1.69, 1.82 SF_{Gear}= 1.24, 1.73, 2.26, 2.73, 2.95

Discussion - Conclusions

The final product of our gear box design was conducted with the use of Microsoft excel, and solid works software. Calculating precise dimensions and tolerances was a requirement to get the necessary measurements towards building our final gear box design.

The design specifications for our designed gearbox were

- Power delivered 25HP
- Input speed: 1750 rpm

- Output speed: 500 rpm
- Gear pressure angle: normal 20 degrees
- The fatigue safety factors for pinion and gears are 1.2 and 1.2 and 1.5 for bending
- Design shafts with first cycle yielding safety factor ny ≥2.5.
- Design keys with yielding safety factor ny = 2
- The total length of each shaft is 7 in
- The diameter of each designed shaft is 1 in

The specifications of our gear box design were implemented to fit each individual component of the gear and shafts for each designed component to properly fit into the single gear box. Each of the designed components were worked on separately, although the dimensions of each part were dependent on one another. The dependency of the part in the gearbox was used efficiently and developed accurately to their required dimensions in order to deliver the required powers at the proper output and input speeds.

The first components developed in our gearbox were the pinion and gears. The design pinion had 18 teeth and the gears were designed to have 75 teeth. Using Shigley's Mechanical Engineering Design eBook from the tables and data present in section 14-4 and with the necessary calculation steps. The 40 gray cast iron material was used for the pinions and 30 gray cast iron material used for gears designed in our gear box design. When choosing our material, we focused on ensuring the pinion has a stronger material than the gear to ensure they can withstand the pressure of the gears in the gearbox.

With the completion of the design process of the gear and pinions, our focus shifted towards the designing of the shafts. The shafts had to have proper dimensions in length in order to meet the needs of the gears and bearings of the gearbox. The length of shafts as well as the diameter of the shafts was determined as the diameter if the shaft had to be able to prevent the shafts from bending due to the forces of the gears present.

When developing the keys in the gearbox the measurements of the gears and shaft length were carefully considered to accommodate the designing and implementation of the keys in the gearbox.

The final design component that was developed in our gearbox was the housing component. The housing component in our gear box provided the necessary support for each of the bearings and allowed the space between the shafts. Two housing components were mounted, and the dimensions were carefully conserved to allow the space for the assembly of the pinions and gears in the gearbox design.

The overall design of the gearbox project allowed our team members to gain a proper understanding of the topic covered in our lectures. The principles of the shafts, gears and bearings were each explored in this project. The necessary knowledge gained in class lectures were helpful when designing our gearbox. As the challenges of obtaining proper measurements and calculations were simplified by previous knowledge gained in class. The problems faced in this project allowed an opportunity for our members to gain an insightful understanding of the real-world problems we may face as future engineer's working in the job industry. The understanding of adapting and changing measurements and deriving equations to fit our needs were gained and selecting proper material based on cost and need were also learned through this project. With this project we as a team applied the concepts learned in class to create our very own design of a functionable gearbox design.

Supporting Information to Determine the Conclusion:

Gear Design:

WorkSheet 1: Gear Design															
	30 Cast Iron			40 Cast Iron			40 Cast Iron			Steel Grade 1 HB = 275			Steel Grade 2 HB = 275		
Initial Input Data	**Variable**														
Input Power:	H =	25	hp	H =	25	hp	H =	25	hp	H =	25	hp	H =	25	hp
Input Speed:	n_P =	1750	rpm	n_P =	1750	rpm	n_P =	1750	rpm	n_P =	1750	rpm	n_P =	1750	rpm
Diametral Pitch:	P_d =	6	teeth/in	P_d =	4	teeth/in	P_d =	6	teeth/in	P_d =	6	teeth/in	P_d =	6	teeth/in
Number of Pinion Teeth:	N_P =	20		N_P =	20		N_P =	20		N_P =	20		N_P =	20	
Desired Output Speed:	n_G =	500	rpm	n_G =	500	rpm	n_G =	500	rpm	n_G =	500	rpm	n_G =	500	rpm
Computed number of gear teeth:		70			70			70			70			70	
Actual Chosen No. of Gear Teeth:	N_G =	70		N_G =	70		N_G =	70		N_G =	70		N_G =	70	
Computed Data:															
Actual Output Speed:	n_G =	500	rpm	n_G =	500	rpm	n_G =	500	rpm	n_G =	500	rpm	n_G =	500	rpm
Gear Ratio:	m_G =	3.5		m_G =	3.5		m_G =	3.5		m_G =	3.5		m_G =	3.5	
Pitch Diameter - Pinion:	d_P =	3.33	in	d_P =	5.00	in	d_P =	3.33	in	d_P =	3.33	in	d_P =	3.33	in
Pitch Diameter - Gear:	d_G =	11.67	in	d_G =	17.50	in	d_G =	11.67	in	d_G =	11.67	in	d_G =	11.67	in
Center Distance:	C =	7.50	in	C =	11.25	in	C =	7.50	in	C =	7.50	in	C =	7.50	in
Pitch Line Speed (eq 13-34):	V =	1527.16	ft/min	V =	2290.74	ft/min	V =	1527.16	ft/min	V =	1527.16	ft/min	V =	1527.16	ft/min
Transmitted Load (eq 13-35):	W_t =	540.22	lb	W_t =	360.14	lb	W_t =	540.22	lb	W_t =	540.22	lb	W_t =	540.22	lb
Secondary Input Data:															
	Min	Nom	Max	Min	Nom	Max	Min	Nom	Max	Min	Nom	Max	Min	Nom	Max
Face Width Guidelines (in):	1.333	2.000	2.667	2.000	3.000	4.000	1.333	2.000	2.667	1.333	2.000	2.667	1.333	2.000	2.667
Face Width:	F =	2.000	in	F =	3.000	in	F =	2.000	in	F =	2.000	in	F =	2.000	in

Stress Analysis: Bending										HB =	275		HB =	275	
Safety Factor (1.2 for pinion, 1.5 for gear):	n_t	*1.50*	(given)	n_t	*1.50*	(given)	n_t	*1.50*	(given)	n_t	*1.50*	(given)	n_t	*1.50*	(given)
Pinion Bending Stress:	σ_{bP} =	8972	psi Eq. 14-15	σ_{bP} =	3155	psi Eq. 14-15	σ_{bP} =	9393	psi Eq. 14-15	σ_{bP} =	9393	psi Eq. 14-15	σ_{bP} =	8972	psi Eq. 14-15
Gear Bending Stress:	σ_{bG} =	7299	psi Eq. 14-15	σ_{bG} =	2512	psi Eq. 14-15	σ_{bG} =	7388	psi Eq. 14-15	σbG =	7388	psi Eq. 14-15	σ_{bG} =	7299	psi Eq. 14-15
Pinion: Required Bending stress #:	S_{tP} =	19,995	psi,	S_{tP} =	7,030	psi,	S_{tP} =	20,932	psi,	StP =	20,932	psi,	S_{tP} =	19,995	psi,
Gear: Required Bending stress #:	S_{tG} =	15,621	psi	S_{tG} =	5,376	psi	S_{tG} =	15,812	psi	S_{tG} =	15,812	psi	S_{tG} =	15,621	psi
For Given MATL St below:	See Fig. 14-2,3,4 for Steel or...			See Fig. 14-2,3,4 for Steel or...			See Fig. 14-2,3,4 for Steel or...			See Fig. 14-2,3,4 for Steel or...			See Fig. 14-2,3,4 for Steel or...		
Pinion: Bending stress number:	S_{tP} =	8,500	psi T-14-3,14-4	S_{tP} =	13,000	psi T-14-3,14-4	S_{tP} =	13,000	psi T-14-3,14-4	S_{tP} =	34,783	psi T-14-3,14-4	S_{tP} =	45,755	psi T-14-3,14-4
Gear: Bending stress number:	S_{tG} =	8,500	psi T-14-3,14-4	S_{tG} =	13,000	psi T-14-3,14-4	S_{tG} =	13,000	psi T-14-3,14-4	S_{tG} =	34,783	psi T-14-3,14-4	S_{tG} =	45,755	psi T-14-3,14-4
Pinion: Max Allowed Bending stress =	σ_{Pall} =	5,722	psi	σ_{Pall} =	8,751	psi	σ_{Pall} =	8,751	psi	σ_{Pall} =	23,413	psi	σ_{Pall} =	30,799	psi
Gear: Max Allowed Bending stress =	σ_{Gall} =	5,958	psi	σ_{Gall} =	9,112	psi	σ_{Gall} =	9,112	psi	σ_{Gall} =	24,380	psi	σ_{Gall} =	32,070	psi
Actual Safety Factor for Bending															
Pinion		0.64	>SF		2.77	>SF		0.93	>SF		2.49	>SF		3.43	>SF
Gear		0.82	>SF		3.63	>SF		1.23	>SF		3.30	>SF		4.39	>SF
Stress Analysis: Pitting															
Safety Factor:	n_t	*1.20*	(given)	n_t	*1.20*	(given)	n_t	*1.20*	(given)	n_t	*1.20*	(given)	n_t	*1.20*	(given)
Pinion Contact Stress:	σ_{cP} =	53,887	psi Eq. 14-16	σ_{cP} =	31,232	psi Eq. 14-16	σ_{cP} =	53,891	psi Eq. 14-16	σ_{cP} =	79,224	psi Eq. 14-16	σ_{cP} =	79,224	psi Eq. 14-16
Gear Contact Stress:	σ_{cG} =	54,097	psi Eq. 14-16	σ_{cG} =	31,354	psi Eq. 14-16	σ_{cG} =	54,101	psi Eq. 14-16	σ_{cG} =	79,532	psi Eq. 14-16	σ_{cG} =	79,532	psi Eq. 14-16
Pinion: Required Contact Stress # :	S_{cP} =	109,050		S_{cP} =	63,204		S_{cP} =	109,058		S_{cP} =	160,324		S_{cP} =	160,324	
Gear: Required Contact Stress # :	S_{cG} =	102,058		S_{cG} =	59,152		S_{cG} =	102,066		S_{cG} =	150,044		S_{cG} =	150,044	
For Given MATL Sc below:															
Pinion: Contact Stress #:	S_{cP} =	70,000	psi T-14-5,6,7	S_{cP} =	80,000	psi T-14-5,6,7	S_{cP} =	80,000	psi T-14-5,6,7	S_{cP} =	117,650	psi T-14-5,6,7	S_{cP} =	130,275	psi T-14-5,6,7
Gear: Contact Stress #:	S_{cG} =	70,000	psi T-14-5,6,7	S_{cG} =	80,000	psi T-14-5,6,7	S_{cG} =	80,000	psi T-14-5,6,7	S_{cG} =	117,650	psi T-14-5,6,7	S_{cG} =	130,275	psi T-14-5,6,7
Pinion: Max allowed Contact Stress =		41,508	psi		47,438	psi		47,438	psi		69,764	psi		77,250	psi
Gear: Max Alowed Contact Stress =		44,525	psi		50,886	psi		50,886	psi		74,834	psi		82,864	psi
Actual Safety Factor for Contact Stress															
Pinion		0.77	>SH		1.52	>SH		0.88	>SH		0.88	>SH		0.98	>SH
Gear		0.82	>SH		1.62	>SH		0.94	>SH		0.94	>SH		1.04	>SH
Note:	All teeth and Pd do not satisfy design specs for Cast Iron 30			Note:	Everything satisfied		Note:	Does not satisfy Bending or surface safety factor		Note:	Satisfies Bending but not surface safety factor		Note:	Satisfies Bending but not surface safety factor	

Pinion Shaft

WorkSheet 2: Pinion Shaft Design

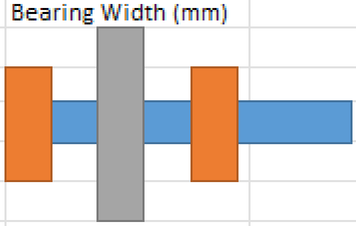
	AISI 1020 CD (T-A-20)		AISI 1040 CD (T-A-20)		AISI 1030 Q&T (T-A-21)			
					Temp (°F)	400	800	1200
Tensile strength(room temperature) (ksi)	S_{ut} =	68	S_{ut} =	85	S_{ut} =	123	106	85
Yield strength (ksi)	S_{yt} =	57	S_{yt} =	71	S_{yt} =	94	84	64
Corrected endurance limit (ksi)	S_e =	34	S_e =	42.5	S_e =	61.5	53	42.5

Calculation:					
Material Properties	**Variable**				
Tensile strength(room temperature) (ksi)	S_{ut} =	123		Note:	changeable parameter
Yield strength (ksi)	S_{yt} =	94			Selected Material
Corrected endurance limit (ksi)	S_e' =	61.5			
	a =	2			
	b =	-0.217			

Stresses												
Location	A (Bearing to Shoulder)		B (Shoulder to Gear)		C (keyseat)		D (Gear Snap ring Groove)		D (Outside Snap ring Groove)		E (outside key)	
Bending moment alternating (lbf in)	M_a =	101.85	M_a =	1006.05	M_a =	1437.22	M_a =	1006.05	M_a =	0	M_a =	0
Bending moment mean (lbf in)	M_m =	0	M_m =	0	M_m =	0	M_m =	0	M_m =	0	M_m =	0
Torque alternating (lbf in)	T_a =	0	T_a =	0	T_a =	0	T_a =	0	T_a =	0	T_a =	0
Torque mean (lbf in)	T_m =	900	T_m =	900	T_m =	900	T_m =	0	T_m =	900	T_m =	900
Marin Factors												
Surface factor (machined) T-6-2	a:	2	a:	2	a:	2	a:	2	a:	2	a:	2
	b:	-0.217	b:	-0.217	b:	-0.217	b:	-0.217	b:	-0.217	b:	-0.217
Surface Factor Eq. 6-19	K_a:	0.704	K_a:	0.704	K_a:	0.704	K_a:	0.704	K_a:	0.704	K_a:	0.704
Shaft Diameter (in)	d (48mm) :	1.890	d (48mm) :	1.890	d (45mm) :	1.772	d (44.5mm) :	1.752	d (44.5mm) :	1.752	d (40mm) :	1.575

Fatigue Stress Concentration Factor												
	D/d	1.089	D/d	1.089			a/t	0.644	a/t	0.644		
	r/d	0.015	r/d	0.015			r/t	0.115	r/t	0.115		
Stress Conc. Factor (Fig.A15-9)	K_t:	1.45	K_t:	1.45	K_t (T-7-1):	2.2	K_t (T-7-1):	5.5	K_t (T-7-1):	5.5	K_t (T-7-1):	2.2
Stress Conc. Factor (shear) (Fig.A15-8)	K_{ts}:	1.15	K_{ts}:	1.15	K_{ts} (T-7-1):	3	K_{ts} (T-7-1):	3.25	K_{ts} (T-7-1):	3.25	K_{ts} (T-7-1):	3
Fatigue Stress Concentration Factor (Eq.6-32)	K_f:	1.266	K_f:	1.266	K_f:	1.709	K_f:	3.274	K_f:	3.274	K_f:	1.709
Fatigue Stress Concentration Factor (shear) (Eq. 6-32)	K_{fs}:	1.099	K_{fs}:	1.099	K_{fs}:	2.317	K_{fs}:	2.298	K_{fs}:	2.298	K_{fs}:	2.317
Fatigue Factor of Saftey												
DE Goodman Safety Factor (Eq.7-7)	n_f	12.612	n_f	10.216	n_f	3.204	n_f	7.914	n_f	9.752	n_f	9.672
Alternating Stress(kpsi) (Eq. 7-4)	σ_a':	526.759	σ_a':	0.850	σ_a':	3.633	σ_a':	2.200	σ_a':	0.000	σ_a':	0.000
Mean Stress (kpsi) (Eq. 7-5)	σ_m':	14440.614	σ_m':	3.334	σ_m':	3.382	σ_m':	0.000	σ_m':	6.973	σ_m':	7.031
Maximum Von Mises (kpsi) (Eq. 7-15)	σ_{max}':	3375.468	σ_{max}':	4.185	σ_{max}':	7.015	σ_{max}':	2.200	σ_{max}':	6.973	σ_{max}':	7.031
First Cycle Yielding (2.5 min) (Eq. 7-16)	n_f	16.887	n_f	13.621	n_f	8.126	n_f	25.913	n_f	8.174	n_f	8.107

Calculations:					
	Torque (eq 18-1)	T	900	in lb	
	Transmitted load	W_t	540.22	lb	
	Total force	W	574.89	lb	
	Moment max	M_{max}	1293.50	lb in	
	Shaft Length (in)		9.00	Half	4.500
	Gear Width (in)		3.00	Widths	1.500
	Bearing Width (mm)		18.00	(in)	0.354

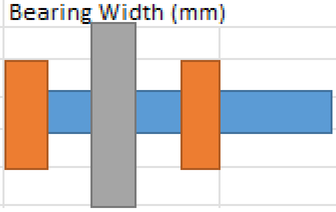


11.25	285.75				
1.7272	0.068				

Gear Shaft

Location	A (Bearing to Shoulder)		B (Shoulder to Gear)		C (keyseat)		D (Gear Snap Ring Groove)		E (outside key)	
Bending moment alternating (lbf in)	M_a =	101.85	M_a =	862.33	M_a =	1293.50	M_a =	862.33	M_a =	0.00
Bending moment mean (lbf in)	M_m =	0	M_m =	0	M_m =	0	M_m =	0	M_m =	0
Torque alternating (lbf in)	T_a =	0	T_a =	0	T_a =	0	T_a =	0	T_a =	0
Torque mean (lbf in)	T_m =	0	T_m =	0	T_m =	3150	T_m =	3150	T_m =	3150
Marin Factors										
Surface factor (machined) T-6-2	a:	2	a:	2	a:	2	a:	2	a:	2
	b:	-0.217	b:	-0.217	b:	-0.217	b:	-0.217	b:	-0.217
Surface Factor Eq. 6-18	K_a:	0.704	K_a:	0.704	K_a:	0.704	K_a:	0.704	K_a:	0.704
Shaft Diameter (in)	d (48mm) :	1.890	d (48mm) :	1.890	d (45mm) :	1.772	d (44.5mm) :	1.752	d (40mm) :	1.575
Size Factor (rotating shaft), Eq. 6-19	K_b:	0.821	K_b:	0.821	K_b:	0.827	K_b:	0.828	K_b:	0.837
Loading Factor, Eq. 6-25	K_c:	1	K_c:	1	K_c:	1	K_c:	1	K_c:	1
Temperature Factor	K_d:	1	K_d:	1	K_d:	1	K_d:	1	K_d:	1
Reliability Factor (0.999) T-6-4	K_e:	0.753	K_e:	0.753	K_e:	0.753	K_e:	0.753	K_e:	0.753
Modified Endurance Limit, Eq.6-17	S_e =	26.767	S_e =	26.767	S_e =	26.953	S_e =	26.985	S_e =	27.295
Notch Sensitivity, q										
Notch radius, r (in) :	0.020		0.020		0.020		0.010		0.020	
Bending										
Neuber constant, √a (Eq. 6-35) :	0.0458		0.0458		0.0458		0.0458		0.0458	
q (Eq. 6-33):	0.7554		0.7554		0.7554		0.6859		0.7554	
Torsion										
Neuber constant, √a (Eq. 6-36) :	0.0358		0.0358		0.0358		0.0358		0.0358	
qs (Eq. 6-33):	0.7979		0.7979		0.7979		0.7362		0.7979	
Fatigue Stress Concentration Factor										
	D/d	0.794	D/d	0.794			a/t	0.121		
	r/d	0.011	r/d	0.011			r/t	0.022		
Stress Conc. Factor (Fig.A15-9)	K_t:	1.45	K_t:	1.45	K_t (T-7-1):	2.2	K_t (T-7-1):	5.5	K_t (T-7-1):	2.2
Stress Conc. Factor (shear) (Fig.A15-8)	K_{ts}:	1.15	K_{ts}:	1.15	K_{ts} (T-7-1):	3	K_{ts} (T-7-1):	3.25	K_{ts} (T-7-1):	3
Fatigue Stress Concentration Factor (Eq.6-32)	K_f:	1.340	K_f:	1.340	K_f:	1.906	K_f:	4.087	K_f:	1.906
:igue Stress Concentration Factor (shear) (Eq. 6-32)	K_{fs}:	1.120	K_{fs}:	1.120	K_{fs}:	2.596	K_{fs}:	2.657	K_{fs}:	2.596
Fatigue Factor of Saftey										
DE Goodman Safety Factor (Eq.7-7)	n_f:	129.951	n_f:	15.349	n_f:	3.662	n_f:	2.786	n_f:	6.660
Alternating Stress(kpsi) (Eq. 7-4)	σ_a':	0.206	σ_a':	1.744	σ_a':	4.517	σ_a':	6.675	σ_a':	0.000
Mean Stress (kpsi) (Eq. 7-5)	σ_m':	0.000	σ_m':	0.000	σ_m':	5.595	σ_m':	13.727	σ_m':	18.468
Maximum Von Mises (kpsi) (Eq. 7-15)	σ_{max}':	0.206	σ_{max}':	1.744	σ_{max}':	10.112	σ_{max}':	20.402	σ_{max}':	18.468
First Cycle Yielding (2.5 min) (Eq. 7-16)	n_f:	456.357	n_f:	53.900	n_f:	9.296	n_f:	4.607	n_f:	5.090

Calculations:					
	Torque (18-1)	T	3150	in lb	
	Transmitted load	W_t	540.22	lb	
	total force	W	574.89	lb	
	Moment max	M_{max}	1293.50	lb in	
	Shaft Length (in)		9.00	Half	4.5
	Gear Width (in)		3.00	Widths	1.5
	Bearing Width (mm)		18.00	(in)	0.35433



Key

WorkSheet 4: Key Design

Input Values		
Horsepower (hp)	25	
RPM Pinion	1750	
RPM Gear	500	
Shaft Diameter (in)	1.772	
S_{yt} of Shaft (ksi)	94	
S_{yt} of Hub (ksi)	140	
Desired Safety Factor	2	
	Pinion Key Material	AISI 1006 HR
Table 7-6	Key Thickness (t)	0.500
	Key Hieght (H)	0.375
Table A-20	S_{yt} of Pinion key (ksi)	24.0
	Gear Key Material	AISI 1018 CD
Table 7-6	Key Thickness (t)	0.500
	Key Hieght (H)	0.375
Table A-20	S_{yt} of Gear key (ksi)	32.0
	Gear Torque T_G (lb*in)	3151.3
	Pinion Torque T_P (lb*in)	900.4
	Gear Force F_G (lb)	3557.4
	Pinion Force F_G (lb)	1016.4
Eq 5-21	S_{sy} of Pinion Key (ksi)	13.8
	S_{sy} of Gear Key (ksi)	18.5
	Pinion	
Eq 8-53	Length for Shear (in)	0.294
Length For Bearing Stress (Eq 8-55)	Key Length (in)	0.452
	Shaft Length (in)	0.115
	Hub Length (in)	0.077
	Gear	
Eq 8-53	Length for Shear (in)	0.771
Length For Bearing Stress (Eq 8-55)	Key Length (in)	1.186
	Shaft Length (in)	0.404
	Hub Length (in)	0.271
	Final Length Pinion Key (in)	
	0.452	
	Final Length Gear Key (in)	
	1.186	

Bearing

	Bearing A						Bearing B				
Bore (mm) Table 11-2	25	30	35	40	45		25	30	35	40	45
Radial Force (lbf)	287.44						287.44				
Axial Force (lbf)	0						0				
Life (hours)	20000						20000				
RPM	1750						1750				
V (1 for internal rotation)	1						1				
C_{10} (KN)Table 11-2	14	19.5	25.5	30.7	33.2		14	19.5	25.5	30.7	33.2
C_0 (KN)Table 11-2	6.95	10	13.7	16.6	18.6		6.95	10	13.7	16.6	18.6
F_a/C_0	0.184	0.128	0.093	0.077	0.069		0.184	0.128	0.093	0.077	0.069
e Table 11-1	0.31	0.28	0.265	0.26	0.24		0.31	0.28	0.265	0.26	0.24
$F_a/V(F_r)$ (Eq 11-11a)	0						0				
X_1 Table 11-1	1	1	1	1	1		1	1	1	1	1
Y_1 Table 11-1	0	0	0	0	0		0	0	0	0	0
F_e (lbf) eq 11-12	287.44	287.44	287.44	287.44	287.44		287.44	287.44	287.44	287.44	287.44
a (3 for ball bearing)	3	3	3	3	3		3	3	3	3	3
$C_{required}$	16.37	16.37	16.37	16.37	16.37		16.37	16.37	16.37	16.37	16.37
Safety Factor	0.86	1.19	1.56	1.87	2.03		0.86	1.19	1.56	1.87	2.03

Gear Shaft Bearing											
	Bearing A						Bearing B				
Bore (mm) Table 11-2	25	30	35	40	45		25	30	35	40	45
Radial Force (lbf)	270.11						270.11				
Axial Force (lbf)	0						0				
Life (hours)	20000						20000				
RPM	500						500				
V (1 for internal rotation)	1						1				
C_{10} (KN)Table 11-2	14	19.5	25.5	30.7	33.2		14	19.5	25.5	30.7	33.2
C_0 (KN)Table 11-2	6.95	10	13.7	16.6	18.6		6.95	10	13.7	16.6	18.6
F_a/C_0	0.173	0.120	0.088	0.072	0.065		0.173	0.120	0.088	0.072	0.065
e Table 11-1	0.31	0.28	0.265	0.26	0.24		0.31	0.28	0.265	0.26	0.24
$F_a/V(F_r)$	0						0				
X_1 Table 11-1	1	1	1	1	1		1	1	1	1	1
Y_1 Table 11-1	0	0	0	0	0		0	0	0	0	0
F_e (lbf) eq 11-12	270.11	270.11	270.11	270.11	270.11		270.11	270.11	270.11	270.11	270.11
a (3 for ball bearing)	3	3	3	3	3		3	3	3	3	3
$C_{required}$	10.13	10.13	10.13	10.13	10.13		10.13	10.13	10.13	10.13	10.13
Safety Factor	1.38	1.92	2.52	3.03	3.28		1.38	1.92	2.52	3.03	3.28

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Appendices

Appendix A: List of Symbols

C_f	Surface condition factor
C_H	Hardness-ratio factor (gear only)
C_P	Gear elastic coefficient
C_{10}	Bearing catalogue load rating
d_P, d_G	Pitch diameter of pinion and gear, respectively
E	Young's Modulus
F	Face width
H	Input power
I	Surface-strength geometry factor
J_P, J_G	Bending-strength geometry factor for pinion and gear, respectively
K_B	Rim thickness factor,
k_a, k_b, k_c, k_d, k_e	Marin factors for fatigue loading
K_m	Load distribution factor,
K_o	Overload factor,
$(K_s)_P$, $(K_s)_G$	Size factor for pinion and gear, respectively
K_R	Gear design reliability factor
K_T	Gear design temperature factor
K_v	Dynamic Factor,
N_P, N_G	Number of teeth of pinion and gear, respectively
m_G	Gear ratio
n	Input velocity
N	Life cycles
P_d	Diametral pitch
R	Reliability
S_c	Contact strength number for cast iron
S_e	Corrected endurance limit
$S_e^{'}$	Uncorrected endurance limit
$(S_H)_P$, $(S_H)_G$	Safety factor for contact stress

$(S_F)_P, (S_F)_G$	Safety factor for bending
S_t	Unadjusted bending stress number
S_{ut}	Ultimate tensile strength
S_{yt}	Tensile yield strength
T	Torque
$(Y_N)_P$, $(Y_N)_G$	Stress cycle factor for bending stress
V	Pitch line velocity
v_P, v_G	Poisson's ratio for pinion and gear
W^t	Transmitted load
$(Z_N)_P$, $(Z_N)_G$	Stress cycle factor for contact stress
ϕ	Pressure angle for spur gear
ϕ_t	Transverse pressure angle
ϕ_n	Normal pressure angle
ψ	Helix angle
$(\sigma)_P$, $(\sigma)_G$	Bending stress on pinion and gear, respectively
$(\sigma_C)_P$, $(\sigma_C)_G$	Tooth wear (contact stress) for pinion and gear

Appendix B: Tables

Table 6–2

Parameters for Marin Surface Modification Factor, Eq. (6–19)

Surface Finish	Factor a S_{ut}, kpsi	Factor a S_{ut}, MPa	Exponent b
Ground	1.34	1.58	−0.085
Machined or cold-drawn	2.70	4.51	−0.265
Hot-rolled	14.4	57.7	−0.718
As-forged	39.9	272.	−0.995

Table 6–5

Reliability Factors k_e Corresponding to 8 Percent Standard Deviation of the Endurance Limit

Reliability, %	Transformation Variate z_a	Reliability Factor k_e
50	0	1.000
90	1.288	0.897
95	1.645	0.868
99	2.326	0.814
99.9	3.091	0.753
99.99	3.719	0.702
99.999	4.265	0.659
99.9999	4.753	0.620

Table 7–6

Inch Dimensions for Some Standard Square- and Rectangular-Key Applications

Source: Joseph E. Shigley, "Unthreaded Fasteners," Chap. 24 in Joseph E. Shigley, Charles R. Mischke, and Thomas H. Brown, Jr. (eds.), *Standard Handbook of Machine Design,* 3rd ed., McGraw-Hill, New York, 2004.

Shaft Diameter Over	Shaft Diameter To (Incl.)	Key Size w	Key Size h	Keyway Depth
$\frac{5}{16}$	$\frac{7}{16}$	$\frac{3}{32}$	$\frac{3}{32}$	$\frac{3}{64}$
$\frac{7}{16}$	$\frac{9}{16}$	$\frac{1}{8}$	$\frac{3}{32}$	$\frac{3}{64}$
		$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{16}$
$\frac{9}{16}$	$\frac{7}{8}$	$\frac{3}{16}$	$\frac{1}{8}$	$\frac{1}{16}$
		$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{32}$
$\frac{7}{8}$	$1\frac{1}{4}$	$\frac{1}{4}$	$\frac{3}{16}$	$\frac{3}{32}$
		$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{8}$
$1\frac{1}{4}$	$1\frac{3}{8}$	$\frac{5}{16}$	$\frac{1}{4}$	$\frac{1}{8}$
		$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{32}$
$1\frac{3}{8}$	$1\frac{3}{4}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{8}$
		$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{16}$
$1\frac{3}{4}$	$2\frac{1}{4}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{3}{16}$
		$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{4}$
$2\frac{1}{4}$	$2\frac{3}{4}$	$\frac{5}{8}$	$\frac{7}{16}$	$\frac{7}{32}$
		$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{16}$
$2\frac{3}{4}$	$3\frac{1}{4}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{4}$
		$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{8}$

Table 11–1

Equivalent Radial Load Factors for Ball Bearings

		$F_a/(VF_r) \le e$		$F_a/(VF_r) > e$	
F_a/C_0	e	X_1	Y_1	X_2	Y_2
0.014*	0.19	1.00	0	0.56	2.30
0.021	0.21	1.00	0	0.56	2.15
0.028	0.22	1.00	0	0.56	1.99
0.042	0.24	1.00	0	0.56	1.85
0.056	0.26	1.00	0	0.56	1.71
0.070	0.27	1.00	0	0.56	1.63
0.084	0.28	1.00	0	0.56	1.55
0.110	0.30	1.00	0	0.56	1.45
0.17	0.34	1.00	0	0.56	1.31
0.28	0.38	1.00	0	0.56	1.15
0.42	0.42	1.00	0	0.56	1.04
0.56	0.44	1.00	0	0.56	1.00

*Use 0.014 if $F_a/C_0 < 0.014$.

Table 11–2

Dimensions and Load Ratings for Single-Row 02-Series Deep-Groove and Angular-Contact Ball Bearings

Bore, mm	OD, mm	Width, mm	Fillet Radius, mm	Shoulder Diameter, mm		Load Ratings, kN			
						Deep Groove		Angular Contact	
				d_S	d_H	C_{10}	C_0	C_{10}	C_0
10	30	9	0.6	12.5	27	5.07	2.24	4.94	2.12
12	32	10	0.6	14.5	28	6.89	3.10	7.02	3.05
15	35	11	0.6	17.5	31	7.80	3.55	8.06	3.65
17	40	12	0.6	19.5	34	9.56	4.50	9.95	4.75
20	47	14	1.0	25	41	12.7	6.20	13.3	6.55
25	52	15	1.0	30	47	14.0	6.95	14.8	7.65
30	62	16	1.0	35	55	19.5	10.0	20.3	11.0
35	72	17	1.0	41	65	25.5	13.7	27.0	15.0
40	80	18	1.0	46	72	30.7	16.6	31.9	18.6
45	85	19	1.0	52	77	33.2	18.6	35.8	21.2
50	90	20	1.0	56	82	35.1	19.6	37.7	22.8
55	100	21	1.5	63	90	43.6	25.0	46.2	28.5
60	110	22	1.5	70	99	47.5	28.0	55.9	35.5
65	120	23	1.5	74	109	55.9	34.0	63.7	41.5
70	125	24	1.5	79	114	61.8	37.5	68.9	45.5
75	130	25	1.5	86	119	66.3	40.5	71.5	49.0
80	140	26	2.0	93	127	70.2	45.0	80.6	55.0
85	150	28	2.0	99	136	83.2	53.0	90.4	63.0
90	160	30	2.0	104	146	95.6	62.0	106	73.5
95	170	32	2.0	110	156	108	69.5	121	85.0

Table 14–2

Values of the Lewis Form Factor Y (These Values Are for a Normal Pressure Angle of 20°, Full-Depth Teeth, and a Diametral Pitch of Unity in the Plane of Rotation)

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 14–3

Repeatedly Applied Bending Strength S_t at 10^7 Cycles and 0.99 Reliability for Steel Gears

Source: ANSI/AGMA 2001-D04.

Material Designation	Heat Treatment	Minimum Surface Hardness[1]	Allowable Bending Stress Number S_t,[2] psi		
			Grade 1	Grade 2	Grade 3
Steel[3]	Through-hardened	See Fig. 14–2	See Fig. 14–2	See Fig. 14–2	—
	Flame[4] or induction hardened[4] with type A pattern[5]	See Table 8*	45 000	55 000	—
	Flame[4] or induction hardened[4] with type B pattern[5]	See Table 8*	22 000	22 000	—
	Carburized and hardened	See Table 9*	55 000	65 000 or 70 000[6]	75 000
	Nitrided[4,7] (through-hardened steels)	83.5 HR15N	See Fig. 14–3	See Fig. 14–3	—
Nitralloy 135M, Nitralloy N, and 2.5% chrome (no aluminum)	Nitrided[4,7]	87.5 HR15N	See Fig. 14–4	See Fig. 14–4	See Fig. 14–4

Table 14–4

Repeatedly Applied Bending Strength S_t for Iron and Bronze Gears at 10^7 Cycles and 0.99 Reliability

Source: ANSI/AGMA 2001-D04.

Material	Material Designation[1]	Heat Treatment	Typical Minimum Surface Hardness[2]	Allowable Bending Stress Number, S_{t},[3] psi
ASTM A48 gray cast iron	Class 20	As cast	—	5000
	Class 30	As cast	174 HB	8500
	Class 40	As cast	201 HB	13 000
ASTM A536 ductile (nodular) Iron	Grade 60–40–18	Annealed	140 HB	22 000–33 000
	Grade 80–55–06	Quenched and tempered	179 HB	22 000–33 000
	Grade 100–70–03	Quenched and tempered	229 HB	27 000–40 000
	Grade 120–90–02	Quenched and tempered	269 HB	31 000–44 000
Bronze		Sand cast	Minimum tensile strength 40 000 psi	5700
	ASTM B–148 Alloy 954	Heat treated	Minimum tensile strength 90 000 psi	23 600

Table 14–5

Nominal Temperature Used in Nitriding and Hardnesses Obtained

Source: Darle W. Dudley, *Handbook of Practical Gear Design*, rev. ed., McGraw-Hill, New York, 1984.

Steel	Temperature Before Nitriding, °F	Nitriding, °F	Hardness, Rockwell C Scale Case	Hardness, Rockwell C Scale Core
Nitralloy 135*	1150	975	62–65	30–35
Nitralloy 135M	1150	975	62–65	32–36
Nitralloy N	1000	975	62–65	40–44
AISI 4340	1100	975	48–53	27–35
AISI 4140	1100	975	49–54	27–35
31 Cr Mo V 9	1100	975	58–62	27–33

*Nitralloy is a trademark of the Nitralloy Corp., New York.

Table 14–6

Repeatedly Applied Contact Strength S_c at 10^7 Cycles and 0.99 Reliability for Steel Gears

Source: ANSI/AGMA 2001-D04.

Material Designation	Heat Treatment	Minimum Surface Hardness[1]	Allowable Contact Stress Number,[2] S_c, psi		
			Grade 1	Grade 2	Grade 3
Steel[3]	Through hardened[4]	See Fig. 14–5	See Fig. 14–5	See Fig. 14–5	—
	Flame[5] or induction hardened[5]	50 HRC	170 000	190 000	—
		54 HRC	175 000	195 000	—
	Carburized and hardened[5]	See Table 9*	180 000	225 000	275 000
	Nitrided[5] (through hardened steels)	83.5 HR15N	150 000	163 000	175 000
		84.5 HR15N	155 000	168 000	180 000
2.5% chrome (no aluminum)	Nitrided[5]	87.5 HR15N	155 000	172 000	189 000
Nitralloy 135M	Nitrided[5]	90.0 HR15N	170 000	183 000	195 000
Nitralloy N	Nitrided[5]	90.0 HR15N	172 000	188 000	205 000
2.5% chrome (no aluminum)	Nitrided[5]	90.0 HR15N	176 000	196 000	216 000

Table 14–7

Repeatedly Applied Contact Strength S_c 10^7 Cycles and 0.99 Reliability for Iron and Bronze Gears

Source: ANSI/AGMA 2001-D04.

Material	Material Designation[1]	Heat Treatment	Typical Minimum Surface Hardness[2]	Allowable Contact Stress Number,[3] S_c, psi
ASTM A48 gray cast iron	Class 20	As cast	—	50 000–60 000
	Class 30	As cast	174 HB	65 000–75 000
	Class 40	As cast	201 HB	75 000–85 000
ASTM A536 ductile (nodular) iron	Grade 60–40–18	Annealed	140 HB	77 000–92 000
	Grade 80–55–06	Quenched and tempered	179 HB	77 000–92 000
	Grade 100–70–03	Quenched and tempered	229 HB	92 000–112 000
	Grade 120–90–02	Quenched and tempered	269 HB	103 000–126 000
Bronze	—	Sand cast	Minimum tensile strength 40 000 psi	30 000
	ASTM B-148 Alloy 954	Heat treated	Minimum tensile strength 90 000 psi	65 000

Table 14-8 **Elastic Coefficient C_p (Z_E), $\sqrt{\text{psi}}$ ($\sqrt{\text{MPa}}$)**

Pinion Material	Pinion Modulus of Elasticity $E_p$psi (MPa)*	Gear Material and Modulus of Elasticity $E_{G,}$ lbf/in				
		Steel 30 × 10^6 (2 × 10^5)	Malleable Iron 25 × 10^6 (1.7 × 10^5)	Nodular Iron 24 × 10^6 (1.7 × 10^5)	Cast Iron 22 × 10^6 (1.5 × 10^5)	Alumin Bronz 17.5 × 1 (1.2 × 1
Steel	30 × 10^6 (2 × 10^5)	2300 (191)	2180 (181)	2160 (179)	2100 (174)	1950 (162
Malleable iron	25 × 10^6 (1.7 × 10^5)	2180 (181)	2090 (174)	2070 (172)	2020 (168)	1900 (158
Nodular iron	24 × 10^6 (1.7 × 10^5)	2160 (179)	2070 (172)	2050 (170)	2000 (166)	1880 (156
Cast iron	22 × 10^6 (1.5 × 10^5)	2100 (174)	2020 (168)	2000 (166)	1960 (163)	1850 (154
Aluminum bronze	17.5 × 10^6 (1.2 × 10^5)	1950 (162)	1900 (158)	1880 (156)	1850 (154)	1750 (145
Tin bronze	16 × 10^6 (1.1 × 10^5)	1900 (158)	1850 (154)	1830 (152)	1800 (149)	1700 (141

Poisson's ratio = 0.30.

Table 14–10

Reliability Factors K_R (Y_Z)

Source: ANSI/AGMA 2001-D04.

Reliability	K_R (Y_Z)
0.9999	1.50
0.999	1.25
0.99	1.00
0.90	0.85
0.50	0.70

Table A–5 †Often preferred.

Physical Constants of Materials

Material	Modulus of Elasticity E		Modulus of Rigidity G		Poisson's Ratio ν	Unit Weight w		
	Mpsi	GPa	Mpsi	GPa		lbf/in^3	lbf/ft^3	kN/m^3
Aluminum (all alloys)	10.4	71.7	3.9	26.9	0.333	0.098	169	26.6
Beryllium copper	18.0	124.0	7.0	48.3	0.285	0.297	513	80.6
Brass	15.4	106.0	5.82	40.1	0.324	0.309	534	83.8
Carbon steel	30.0	207.0	11.5	79.3	0.292	0.282	487	76.5
Cast iron (gray)	14.5	100.0	6.0	41.4	0.211	0.260	450	70.6
Copper	17.2	119.0	6.49	44.7	0.326	0.322	556	87.3
Douglas fir	1.6	11.0	0.6	4.1	0.33	0.016	28	4.3
Glass	6.7	46.2	2.7	18.6	0.245	0.094	162	25.4
Inconel	31.0	214.0	11.0	75.8	0.290	0.307	530	83.3
Lead	5.3	36.5	1.9	13.1	0.425	0.411	710	111.5
Magnesium	6.5	44.8	2.4	16.5	0.350	0.065	112	17.6
Molybdenum	48.0	331.0	17.0	117.0	0.307	0.368	636	100.0
Monel metal	26.0	179.0	9.5	65.5	0.320	0.319	551	86.6
Nickel silver	18.5	127.0	7.0	48.3	0.322	0.316	546	85.8
Nickel steel	30.0	207.0	11.5	79.3	0.291	0.280	484	76.0
Phosphor bronze	16.1	111.0	6.0	41.4	0.349	0.295	510	80.1
Stainless steel (18-8)	27.6	190.0	10.6	73.1	0.305	0.280	484	76.0
Titanium alloys	16.5	114.0	6.2	42.4	0.340	0.160	276	43.4

Table 14–9

Empirical Constants A, B, and C for Eq. (14–34), Face Width F in Inches*

Source: ANSI/AGMA 2001-D04.

Condition	A	B	C
Open gearing	0.247	0.0167	$-0.765(10^{-4})$
Commercial, enclosed units	0.127	0.0158	$-0.930(10^{-4})$
Precision, enclosed units	0.0675	0.0128	$-0.926(10^{-4})$
Extraprecision enclosed gear units	0.00360	0.0102	$-0.822(10^{-4})$

*See ANSI/AGMA 2101-D04, pp. 20–22, for SI formulation.

Table A–20

Deterministic ASTM Minimum Tensile and Yield Strengths for Some Hot-Rolled (HR) and Cold-Drawn (CD) Steels [The strengths listed are estimated ASTM minimum values in the size range 18 to 32 mm ($\frac{3}{4}$ to $1\frac{1}{4}$ in). These strengths are suitable for use with the design factor defined in Sec. 1–10, provided the materials conform to ASTM A6 or A568 requirements or are required in the purchase specifications. Remember that a numbering system is not a specification.] *Source:* 1986 SAE Handbook, p. 2.15.

1 UNS No.	2 SAE and/or AISI No.	3 Processing	4 Tensile Strength, MPa (kpsi)	5 Yield Strength, MPa (kpsi)	6 Elongation in 2 in, %	7 Reduction in Area, %	8 Brinell Hardness
G10060	1006	HR	300 (43)	170 (24)	30	55	86
		CD	330 (48)	280 (41)	20	45	95
G10100	1010	HR	320 (47)	180 (26)	28	50	95
		CD	370 (53)	300 (44)	20	40	105
G10150	1015	HR	340 (50)	190 (27.5)	28	50	101
		CD	390 (56)	320 (47)	18	40	111
G10180	1018	HR	400 (58)	220 (32)	25	50	116
		CD	440 (64)	370 (54)	15	40	126
G10200	1020	HR	380 (55)	210 (30)	25	50	111
		CD	470 (68)	390 (57)	15	40	131
G10300	1030	HR	470 (68)	260 (37.5)	20	42	137
		CD	520 (76)	440 (64)	12	35	149
G10350	1035	HR	500 (72)	270 (39.5)	18	40	143
		CD	550 (80)	460 (67)	12	35	163
G10400	1040	HR	520 (76)	290 (42)	18	40	149
		CD	590 (85)	490 (71)	12	35	170
G10450	1045	HR	570 (82)	310 (45)	16	40	163
		CD	630 (91)	530 (77)	12	35	179
G10500	1050	HR	620 (90)	340 (49.5)	15	35	179
		CD	690 (100)	580 (84)	10	30	197
G10600	1060	HR	680 (98)	370 (54)	12	30	201
G10800	1080	HR	770 (112)	420 (61.5)	10	25	229
G10950	1095	HR	830 (120)	460 (66)	10	25	248

Appendix C: Figures

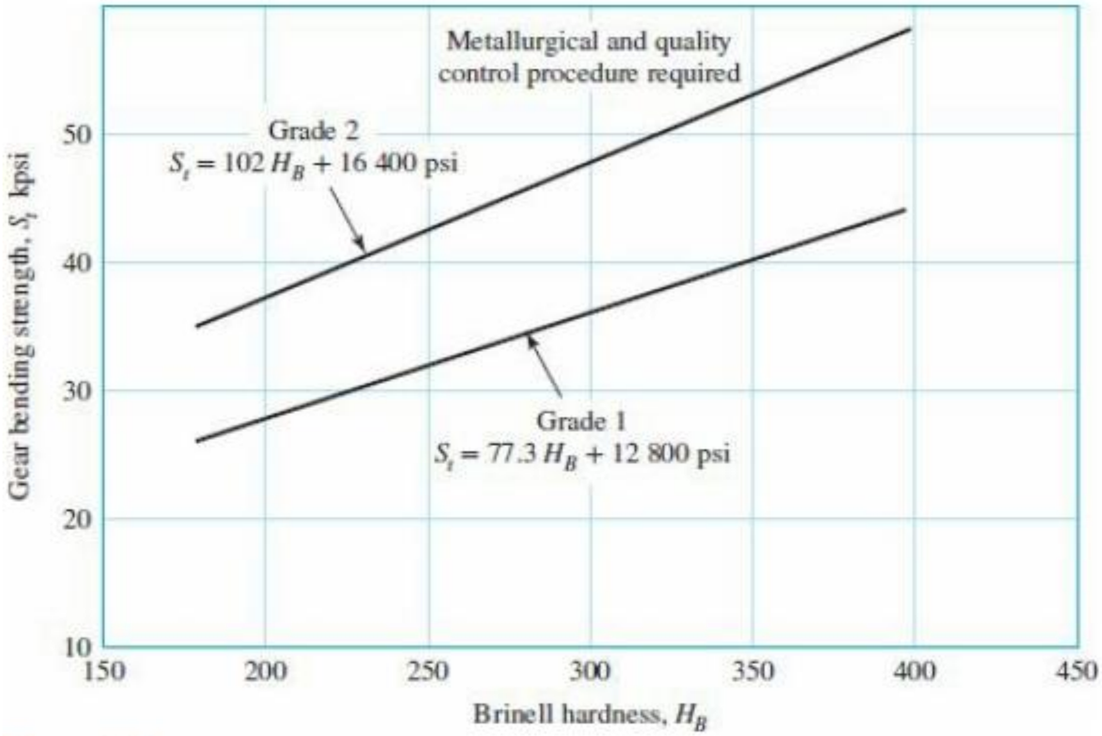


Figure 14-2

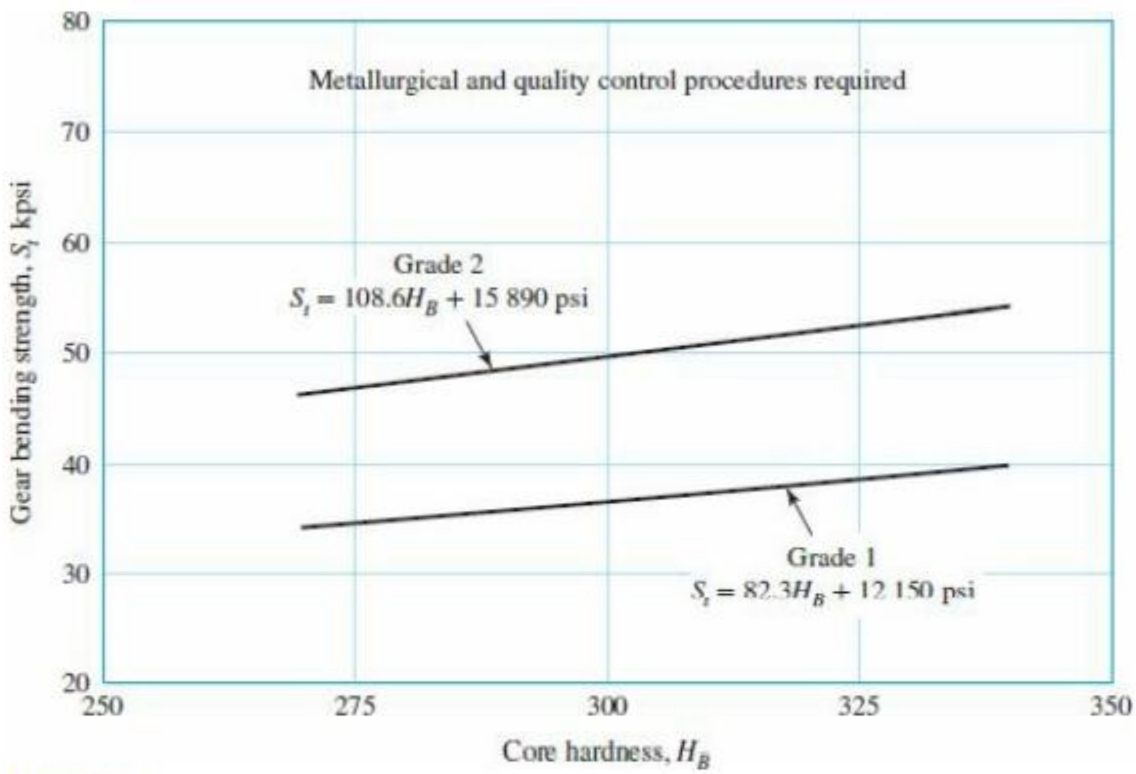


Figure 14-3

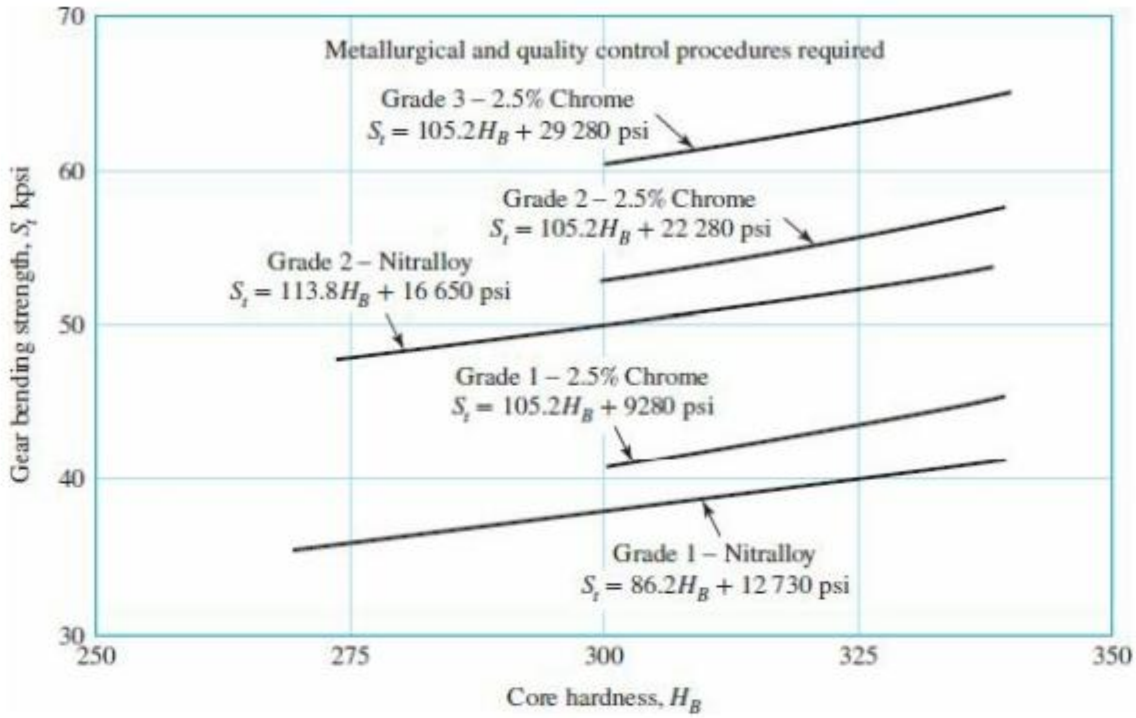

Metallurgical and quality control procedures required
Grade 3 – 2.5% Chrome
$S_t = 105.2H_B + 29\,280$ psi
Grade 2 – 2.5% Chrome
$S_t = 105.2H_B + 22\,280$ psi
Grade 2 – Nitralloy
$S_t = 113.8H_B + 16\,650$ psi
Grade 1 – 2.5% Chrome
$S_t = 105.2H_B + 9280$ psi
Grade 1 – Nitralloy
$S_t = 86.2H_B + 12\,730$ psi
Gear bending strength, S_t kpsi
Core hardness, H_B
70
60
50
40
30
250
275
300
325
350

Figure 14-4

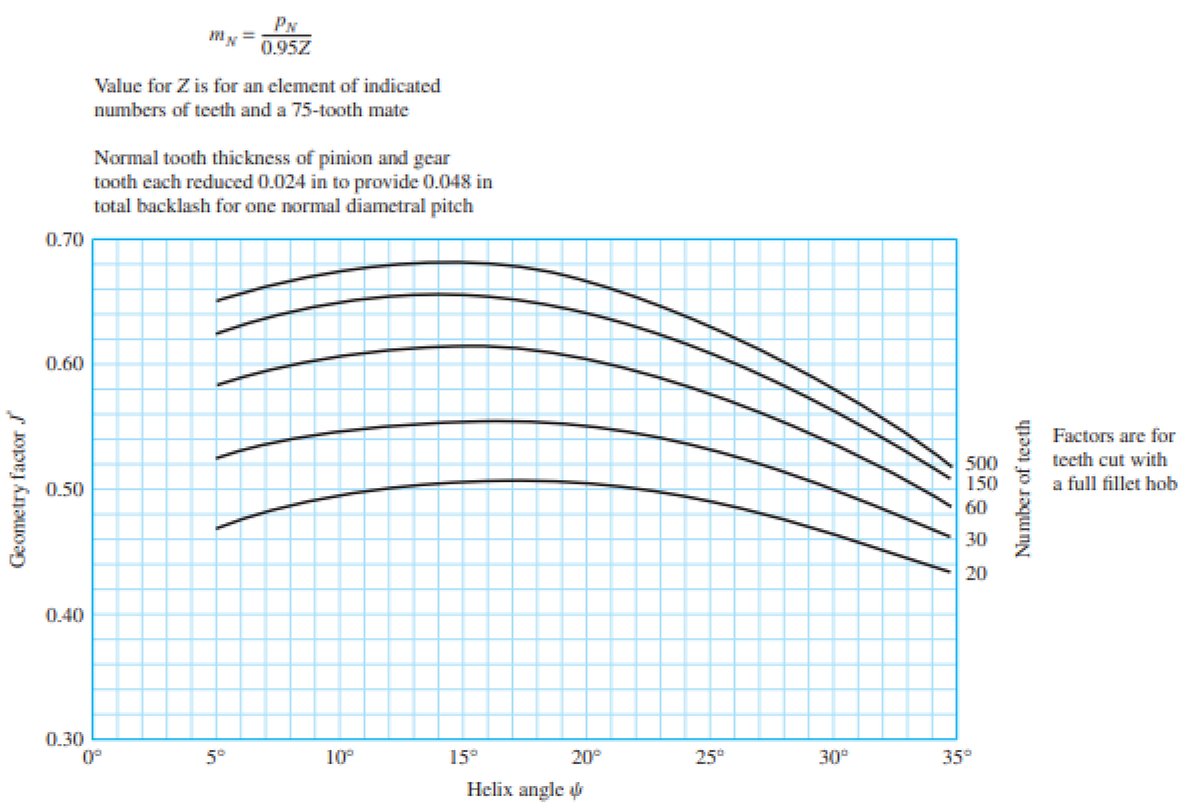

$m_N = \frac{p_N}{0.95Z}$
Value for Z is for an element of indicated numbers of teeth and a 75-tooth mate
Normal tooth thickness of pinion and gear tooth each reduced 0.024 in to provide 0.048 in total backlash for one normal diametral pitch
Geometry factor J'
0.70
0.60
0.50
0.40
0.30
0°
5°
10°
15°
20°
25°
30°
35°
Helix angle ψ
500
150
60
30
20
Number of teeth
Factors are for teeth cut with a full fillet hob

Figure 14–14

Repeatedly applied bending strength stress-cycle factor Y_N *(ANSI/AGMA 2001-D04.)*

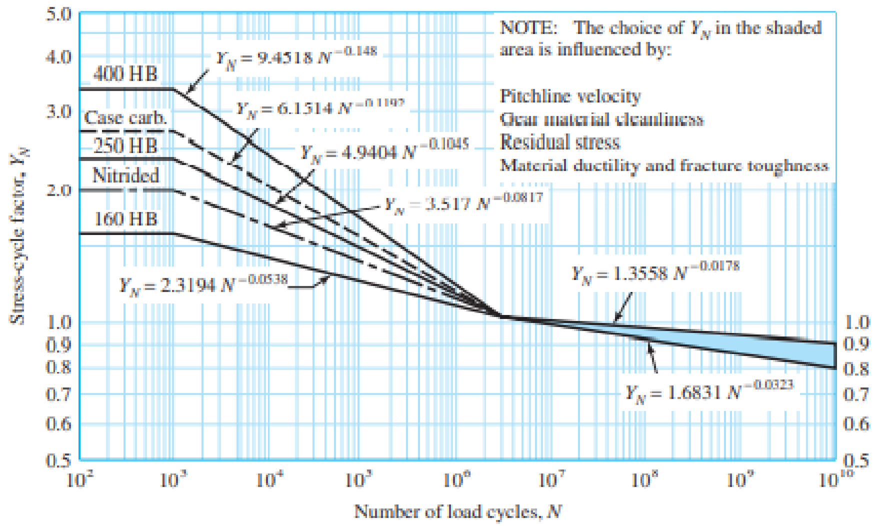
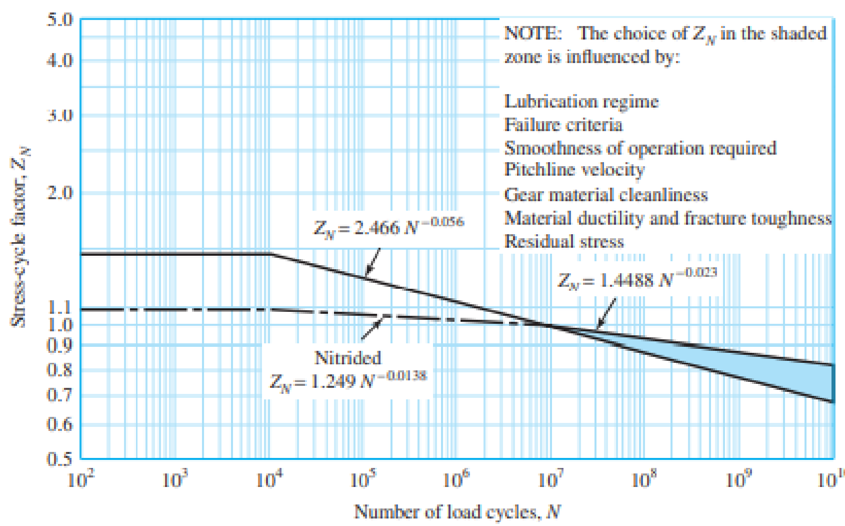


Figure 14–15

Pitting resistance stress-cycle factor Z_N. *(ANSI/AGMA 2001-D04.)*



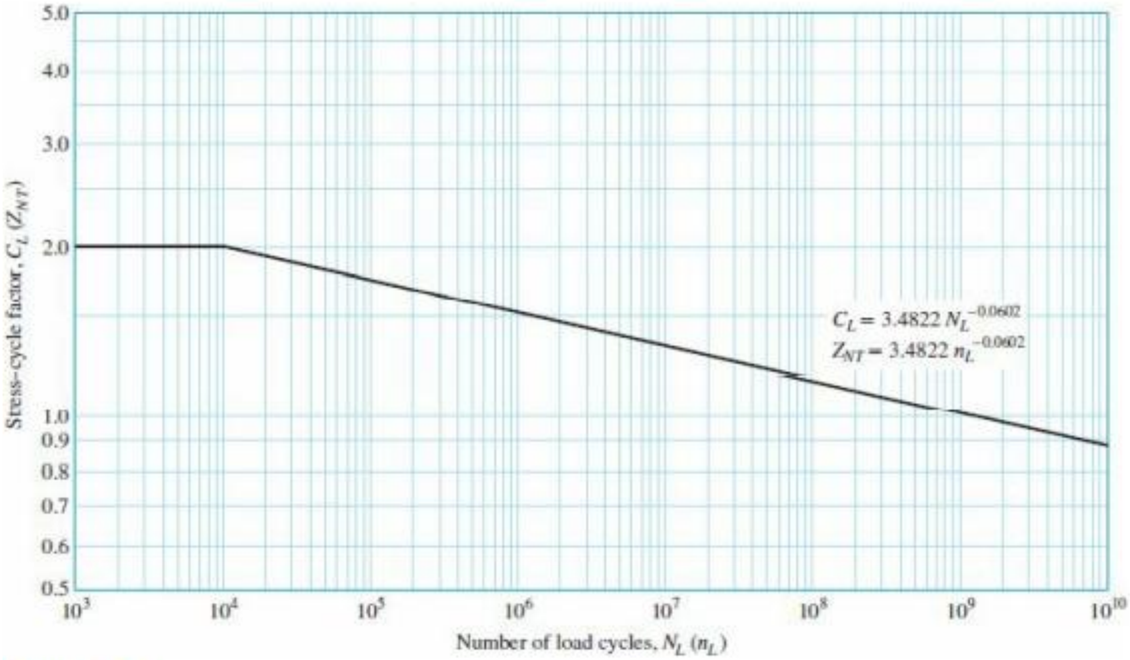

5.0
4.0
3.0
2.0
1.0
0.9
0.8
0.7
0.6
0.5
Stress-cycle factor, C_L (Z_{NT})
$C_L = 3.4822\, N_L^{-0.0602}$
$Z_{NT} = 3.4822\, n_L^{-0.0602}$
10^3
10^4
10^5
10^6
10^7
10^8
10^9
10^{10}
Number of load cycles, N_L (n_L)

Figure 15-8

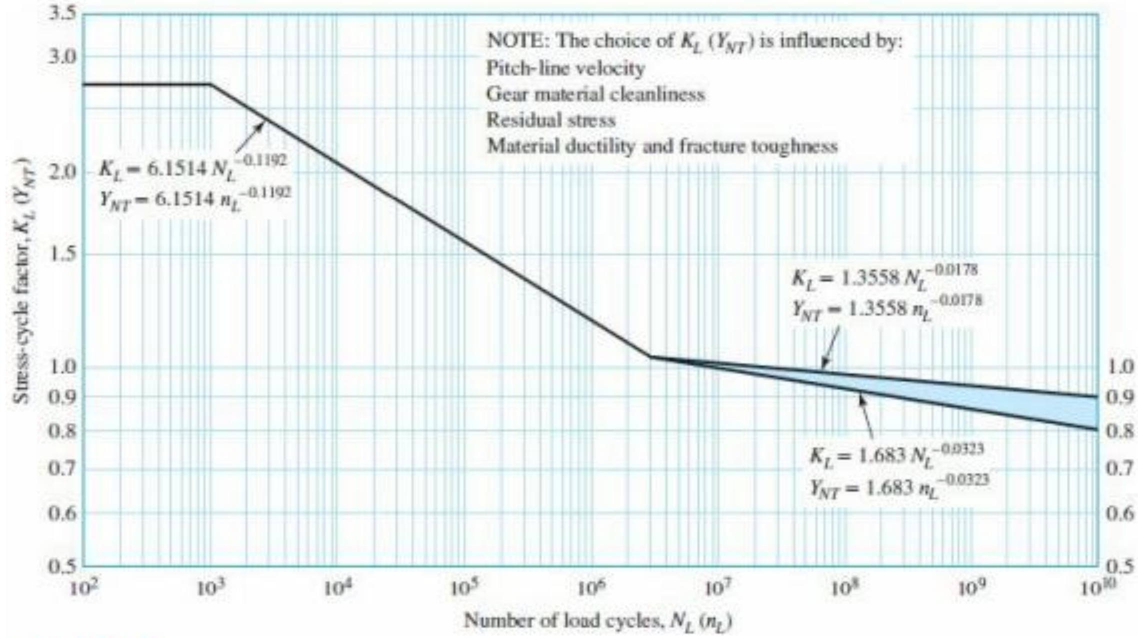


Figure 15-9

Table of Overload Factors, K_o

	Driven Machine		
Power source	Uniform	Moderate shock	Heavy shock
Uniform	1.00	1.25	1.75
Light shock	1.25	1.50	2.00
Medium shock	1.50	1.75	2.25

Appendix C: Equations

$$k_a = aS_{ut}^b \qquad (6\text{–}19)$$

$$k_b = \begin{cases} (d/0.3)^{-0.107} = 0.879d^{-0.107} & 0.11 \le d \le 2 \text{ in} \\ 0.91d^{-0.157} & 2 < d \le 10 \text{ in} \\ (d/7.62)^{-0.107} = 1.24d^{-0.107} & 2.79 \le d \le 51 \text{ mm} \\ 1.51d^{-0.157} & 51 < d \le 254 \text{ mm} \end{cases} \qquad (6\text{–}20)$$

$$k_c = \begin{cases} 1 & \text{bending} \\ 0.85 & \text{axial} \\ 0.59 & \text{torsion} \end{cases} \qquad (6\text{–}26)$$

$$K_f = 1 + q(K_t - 1) \quad \text{or} \quad K_{fs} = 1 + q_{\text{shear}}(K_{ts} - 1) \qquad (6\text{–}32)$$

$$q = \frac{1}{1 + \dfrac{\sqrt{a}}{\sqrt{r}}} \qquad (6\text{–}34)$$

$$\sqrt{a} = 0.246 - 3.08(10^{-3})S_{ut} + 1.51(10^{-5})S_{ut}^2 - 2.67(10^{-8})S_{ut}^3 \qquad 50 \le S_{ut} \le 250 \text{ kpsi}$$

$$\sqrt{a} = 1.24 - 2.25(10^{-3})S_{ut} + 1.60(10^{-6})S_{ut}^2 - 4.11(10^{-10})S_{ut}^3 \qquad 340 \le S_{ut} \le 1700 \text{ MPa} \qquad (6\text{-}35)$$

$$\sqrt{a} = 0.190 - 2.51(10^{-3})S_{ut} + 1.35(10^{-5})S_{ut}^2 - 2.67(10^{-8})S_{ut}^3 \qquad 50 \le S_{ut} \le 220 \text{ kpsi}$$

$$\sqrt{a} = 0.958 - 1.83(10^{-3})S_{ut} + 1.43(10^{-6})S_{ut}^2 - 4.11(10^{-10})S_{ut}^3 \qquad 340 \le S_{ut} \le 1500 \text{ MPa} \qquad (6\text{-}36)$$

$$\sigma'_a = (\sigma_a^2 + 3\tau_a^2)^{1/2} = \left[\left(\frac{32K_f M_a}{\pi d^3}\right)^2 + 3\left(\frac{16K_{fs} T_a}{\pi d^3}\right)^2\right]^{1/2} \qquad (7\text{-}4)$$

$$\sigma'_m = (\sigma_m^2 + 3\tau_m^2)^{1/2} = \left[\left(\frac{32K_f M_m}{\pi d^3}\right)^2 + 3\left(\frac{16K_{fs} T_m}{\pi d^3}\right)^2\right]^{1/2} \qquad (7\text{-}5)$$

$$n = \frac{\pi d^3}{16}\left(\frac{A}{S_e} + \frac{B}{S_{ut}}\right)^{-1} \qquad (7\text{-}7)$$

$$\begin{aligned} \sigma'_{\max} &= [(\sigma_m + \sigma_a)^2 + 3(\tau_m + \tau_a)^2]^{1/2} \\ &= \left[\left(\frac{32K_f(M_m + M_a)}{\pi d^3}\right)^2 + 3\left(\frac{16K_{fs}(T_m + T_a)}{\pi d^3}\right)^2\right]^{1/2} \end{aligned} \qquad (7\text{-}15)$$

$$n_y = \frac{S_y}{\sigma'_{\max}} \qquad (7\text{-}16)$$

$$\tau = \frac{F}{A} \qquad (8\text{–}53)$$

$$\sigma = -\frac{F}{A} \qquad (8\text{–}55)$$

$$F_e = X_i V F_r + Y_i F_a \qquad (11\text{-}12)$$

$$C_p = \left[\frac{1}{\pi\left(\frac{1-\nu_P^2}{E_P} + \frac{1-\nu_G^2}{E_G}\right)}\right]^{1/2} \qquad (14\text{–}13)$$

$$\sigma = \begin{cases} W^t K_o K_v K_s \dfrac{P_d}{F} \dfrac{K_m K_B}{J} & \text{(U.S. customary units)} \\ W^t K_o K_v K_s \dfrac{1}{b m_t} \dfrac{K_H K_B}{Y_J} & \text{(SI units)} \end{cases} \qquad (14\text{–}15)$$

$$\sigma_c = \begin{cases} C_p \sqrt{W^t K_o K_v K_s \dfrac{K_m}{d_P F} \dfrac{C_f}{I}} & \text{(U.S. customary units)} \\ Z_E \sqrt{W^t K_o K_v K_s \dfrac{K_H}{d_{w1} b} \dfrac{Z_R}{Z_I}} & \text{(SI units)} \end{cases} \qquad (14\text{–}16)$$

$$K_v = \begin{cases} \left(\dfrac{A + \sqrt{V}}{A}\right)^B & V \text{ in ft/min} \\ \left(\dfrac{A + \sqrt{200V}}{A}\right)^B & V \text{ in m/s} \end{cases} \qquad (14\text{–}27)$$

$$A = 50 + 56(1 - B)$$
$$B = 0.25(12 - Q_v)^{2/3} \qquad (14\text{–}28)$$

$$K_m = C_{mf} = 1 + C_{mc}(C_{pf} C_{pm} + C_{ma} C_e) \qquad (14\text{–}30)$$

$$C_{pf} = \begin{cases} \dfrac{F}{10d_P} - 0.025 & F \le 1 \text{ in} \\ \dfrac{F}{10d_P} - 0.0375 + 0.0125F & 1 < F \le 17 \text{ in} \\ \dfrac{F}{10d_P} - 0.1109 + 0.0207F - 0.000\,228F^2 & 17 < F \le 40 \text{ in} \end{cases} \qquad (14\text{–}32)$$

$$C_{ma} = A + BF + CF^2 \qquad \text{(see Table 14–9 for values of } A, B, \text{ and } C) \qquad (14\text{–}34)$$

$$S_{sy} = 0.577 S_y \tag{5-21}$$

$$S_e = k_a k_b k_c k_d k_e k_f S'_e \tag{6–18}$$

$$C_H = 1.0 + A'(m_G - 1.0) \tag{14–36}$$

where

$$A' = 8.98(10^{-3})\left(\frac{H_{BP}}{H_{BG}}\right) - 8.29(10^{-3}) \qquad 1.2 \le \frac{H_{BP}}{H_{BG}} \le 1.7$$

$$K_B = \begin{cases} 1.6 \ln \dfrac{2.242}{m_B} & m_B < 1.2 \\ 1 & m_B \ge 1.2 \end{cases} \tag{14–40}$$

$$H = T_i \omega_i = T_o \omega_o \tag{18-1}$$

$$K_s = \frac{1}{k_b} = 1.192\left(\frac{F\sqrt{Y}}{P}\right)^{0.0535} \tag{a}$$

$$V = \pi d n / 12 \tag{13-34}$$

$$W_t = 33\,000 \frac{H}{V} \tag{13-35}$$