ME 308
MACHINE ELEMENTS II

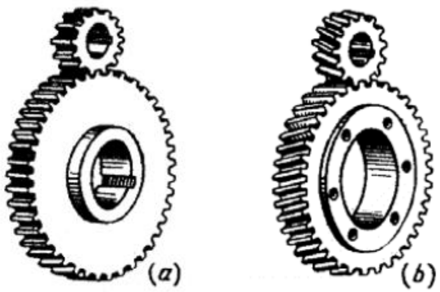
CHAPTER 8

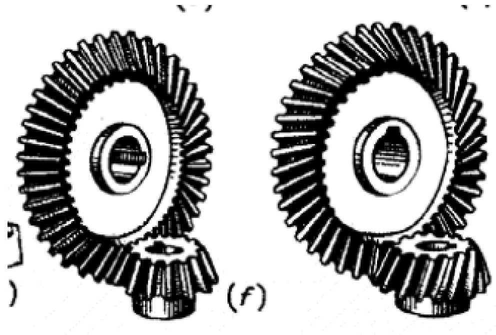
WORM GEARS

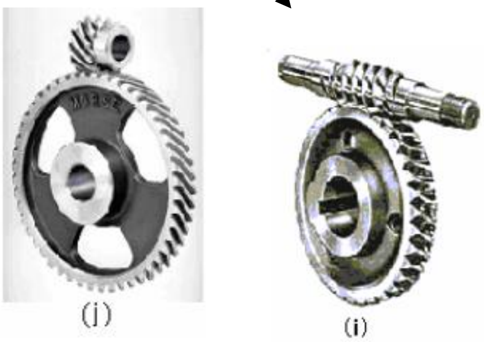
This chapter is about those gears which transmit power between
- non-parallel and
- non-intersecting shafts
- with relatively high speed reduction ratio

DIFFERENT TYPES OF GEARS AND SHAFT CONFIGURATIONS

Similar but not performing the same work







Parallel shafts are used to transmit motion via spur and helical gears

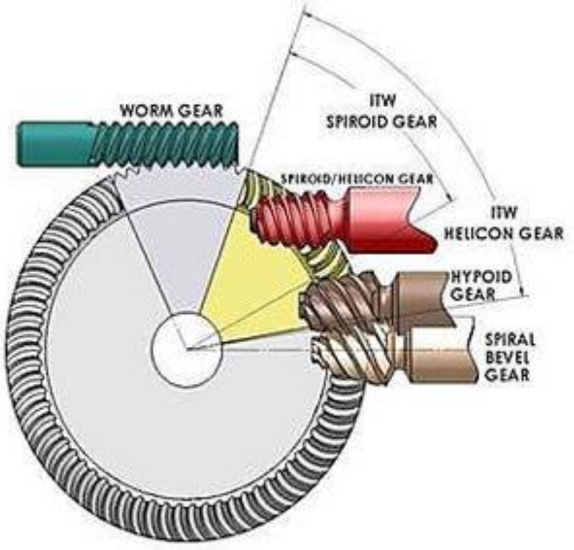
Straight and spiral bevel gears are used in non-parallel and intersecting shafts configuration with 90^{o} shaft angle usually

Non-intersecting non-parallel shafts are usually employed with crossed helical or worm gears

Regarding the figure below, there seems to be a transformation from spiral bevel to worm gearing if the pinion position is changed from center of the gear to top of the gear step-by-step.

The two cases between spiral bevel and worm are the hypoid and spiroid gearing.

The teeth of the gear shift from side to top of the ring too.



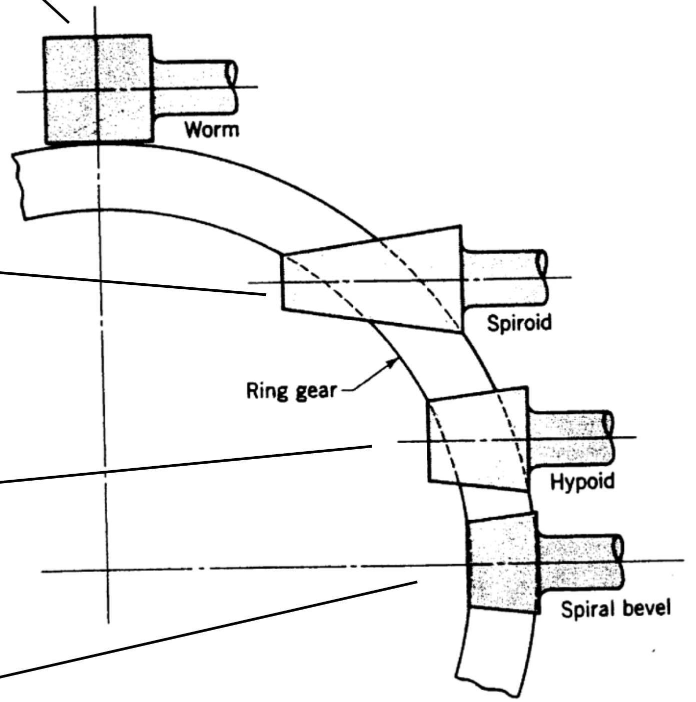
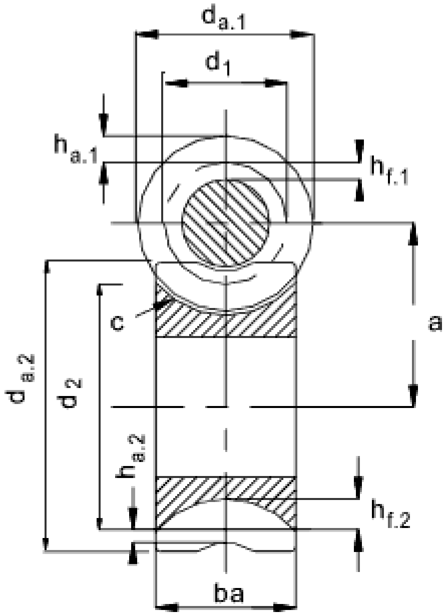
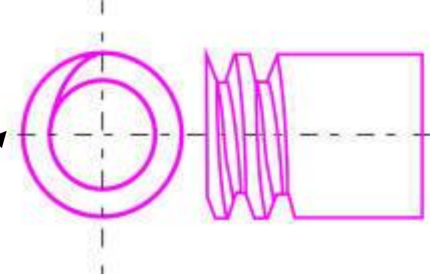


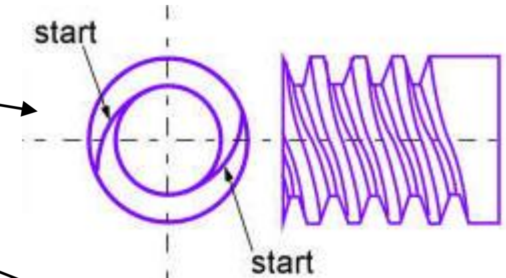
FIGURE 14-28 Comparison of intersecting- and offset-shaft bevel-type gearings. (*By permission, from Gear Handbook, McGraw-Hill, New York, 1962, p. 2–24.*)

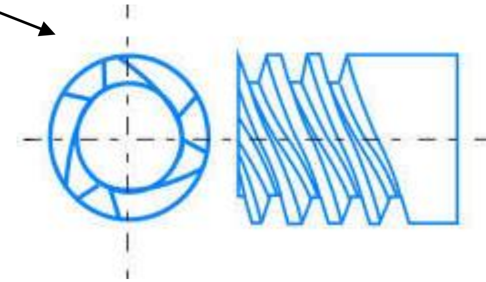




Worms could be:

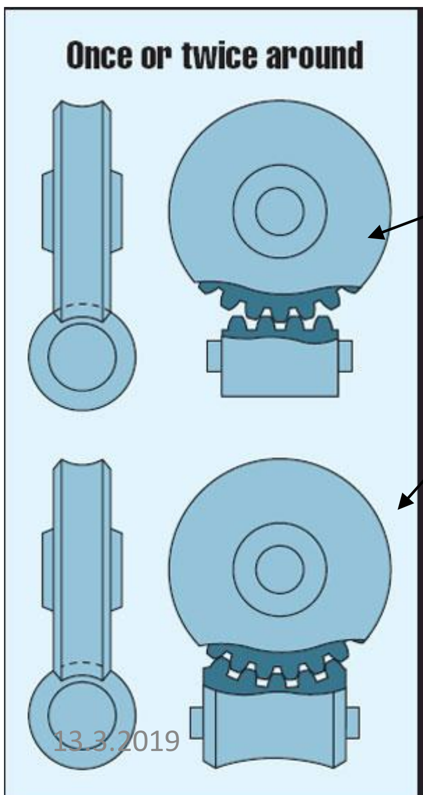
- Single tooth (start)
- Double teeth (start)
- Triple start or
- Quadropole (four) start

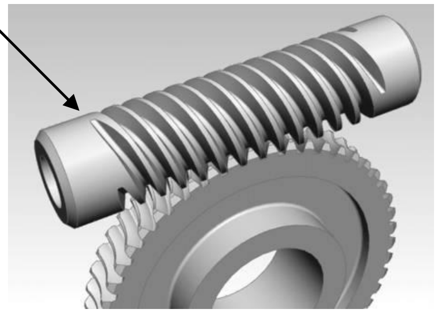




Similarly worm gearing could be

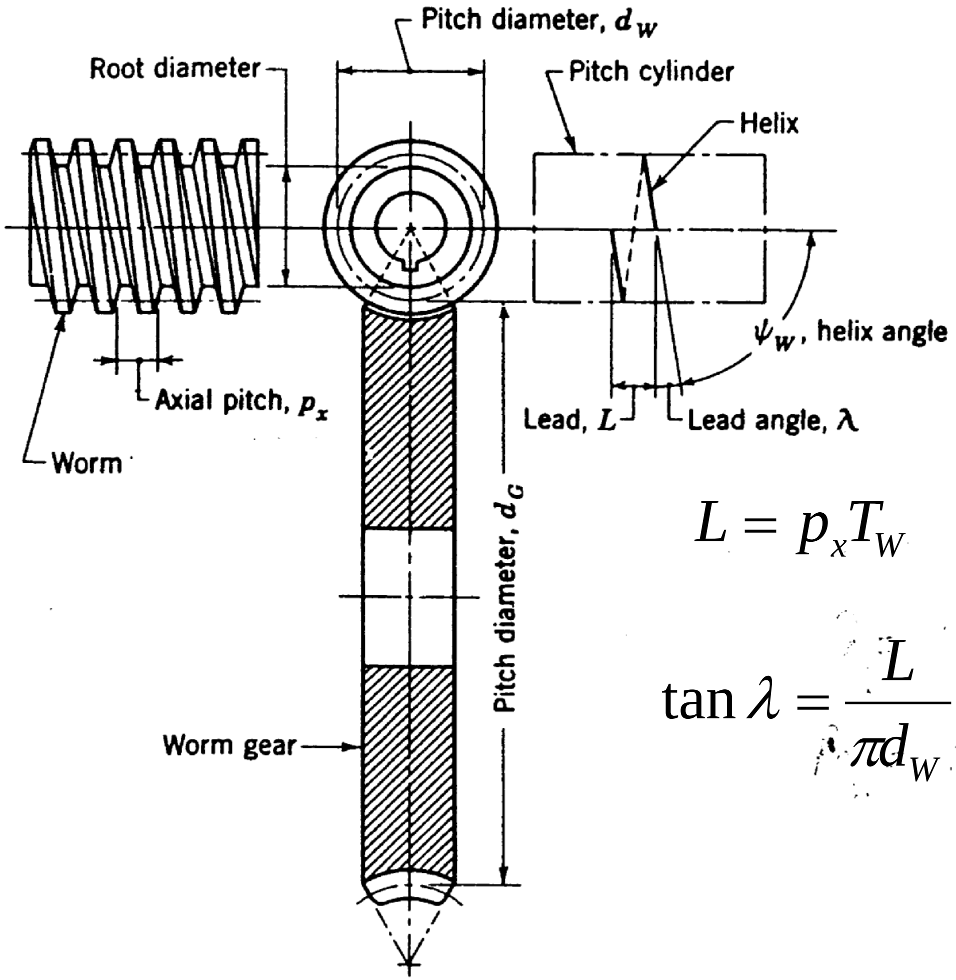
- single enveloping and
- double envoloping





Geometry of worm gearing

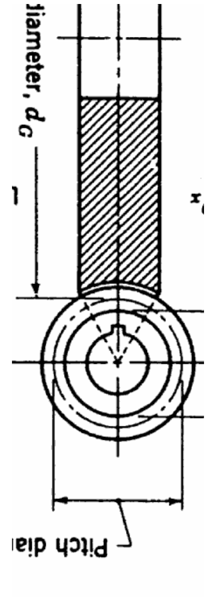
$$\frac{C^{0.875}}{3} \le d_W \le \frac{C^{0.875}}{1.7}$$



$$L = p_x T_W$$

$$\tan\lambda = \frac{L}{\pi d_W}$$

FIGURE 14-13 Nomenclature of a single-enveloping worm gearset.



$$C = \frac{d_W + d_G}{2}$$

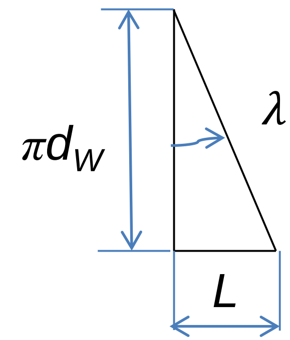


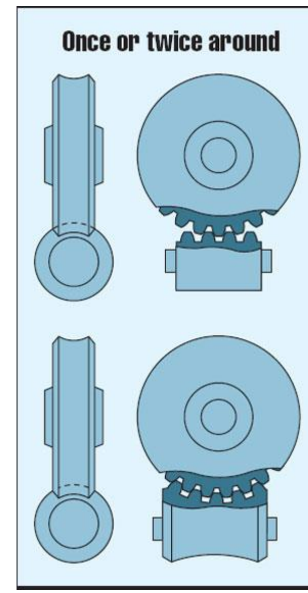
Table 14-2 RECOMMENDED PRESSURE ANGLES AND TOOTH DEPTHS FOR WORM GEARING

Lead angle λ, degrees	Pressure angle ϕ_n, degrees	Addendum a	Dedendum b_G
0–15	$14\frac{1}{2}$	$0.3683p_x$	$0.3683p_x$
15–30	20	$0.3683p_x$	$0.3683p_x$
30–35	25	$0.2865p_x$	$0.3314p_x$
35–40	25	$0.2546p_x$	$0.2947p_x$
40–45	30	$0.2228p_x$	$0.2578p_x$

A) Kinematics

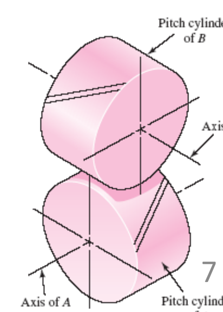
1) Shafts do not intersect and non-parallel shaft angle is 90º (usually).
2) Worm may have 1,2,3,4,5,6 teeth (single tooth, double teeth, triple, quadraple, etc.) (T_w).
3) Worm gearsets are:

- Either single envoloping (gear envelopes the worm - line contact occurs)
- Or double enveloping (both gear envelopes worm and worm envelopes gear - area contact occurs between the teeth)



- <u>Right angle, non-crossing shafts</u>
- <u>High level of speed reduction ratio</u> (1:20 - 1:50)
- <u>High friction and wear due to sliding between worm and gear teeth</u>
- <u>Self locking may (and does) happen</u>
- <u>Driver is always the worm and driven is the gear</u>

As in crossed helical gears both worm and the gear have the same hand of helix

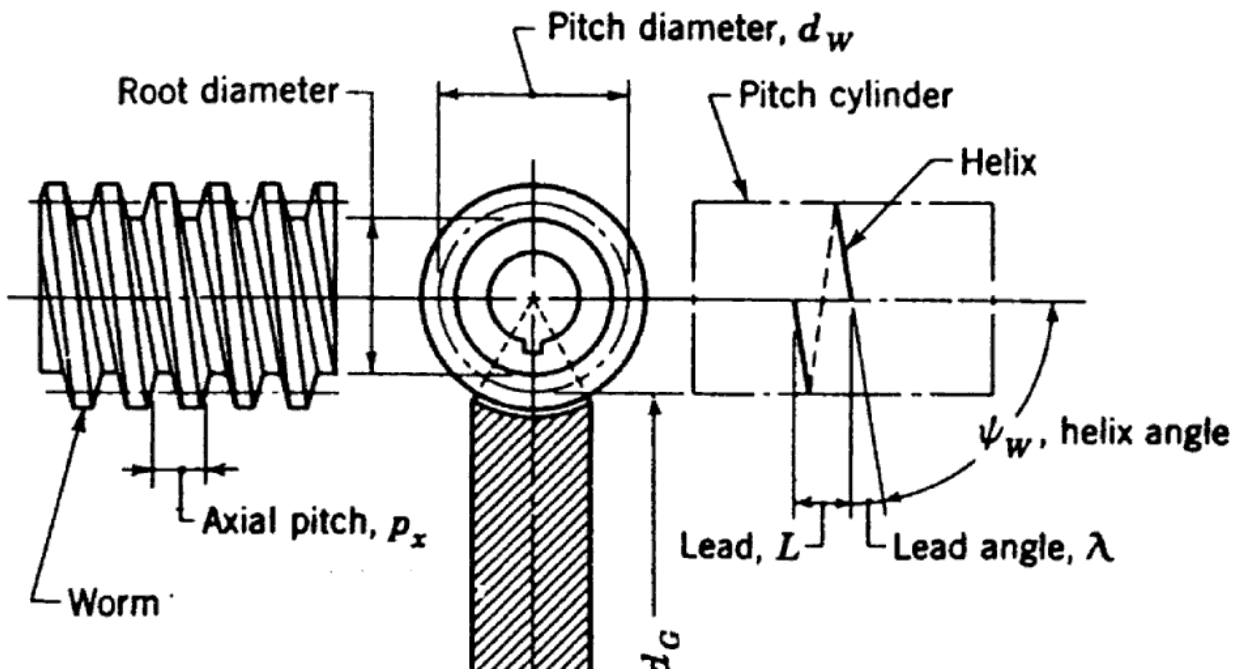


4) As in crossed helical gears both worm and the gear have the same hand but quite different helix angles ψ_W is very large but ψ_G is small.

The lead angle λ of the worm is generally specified which is the complement of ψ_W for 90^o shaft angle

$$\lambda = 90^o - \psi_W$$

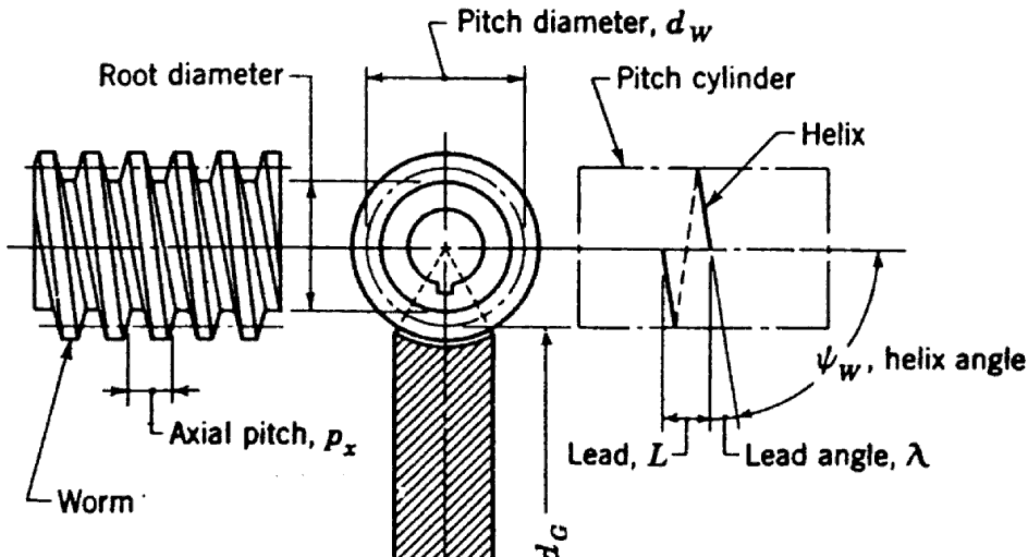
$$\psi_W + \psi_G = 90^o$$



5) For 90° shaft angle $\lambda_W = \psi_G$ and $p_{xW} = p_{tG}$

p_{x_W} = axial pitch of worm

p_{t_G} = transverse pitch of gear



$$d_G = m_{t_G} T_G \qquad d_G = \frac{p_{t_G}}{\pi} T_G$$

in British Units

d_W is "independent" of T_W and may have any value between

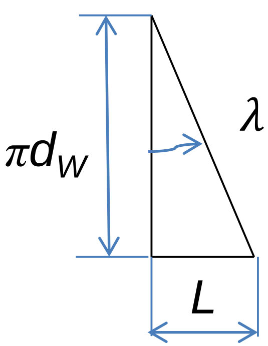
$$\frac{C^{0.875}}{3} \le d_W \le \frac{C^{0.875}}{1.7}$$

for optimum horse power capacity of the gear set

$$C = \frac{d_G + d_W}{2}$$

$$\psi_W + \lambda = 90^o \qquad \psi_W + \psi_G = 90^o$$

$$\text{speed ratio} = \frac{N_W}{N_G} = \frac{T_G}{T_W}$$

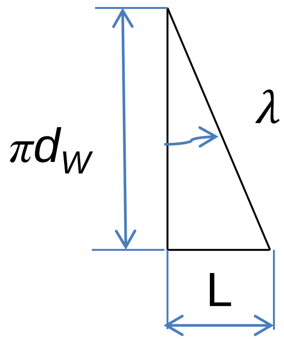


$$\cos\psi_G = \left(\frac{\tan\phi_n}{\tan\phi_t}\right)_G$$

6) The lead L of the worm for T_W number of teeth on the worm is:

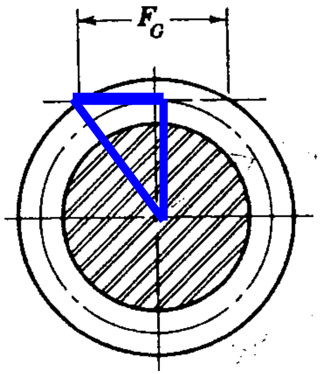
$$L = T_W \times p_x$$

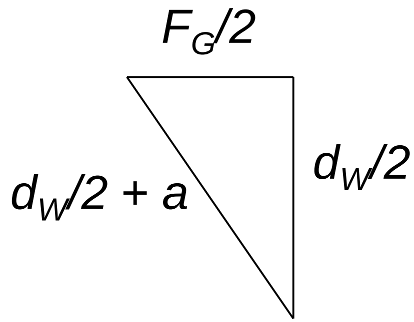
Also from the triangle



$$\tan\lambda = \frac{L}{\pi d_W}$$

7)





$$\left(\frac{d_W}{2}+a\right)^2=\left(\frac{d_G}{2}\right)^2+\left(\frac{F_G}{2}\right)^2 \qquad F_G=? \qquad f(d_W,a)$$

a and b addendum and dedendum is given in Table 14.2 for different λ *and* ϕ_n values of worm gearing.

<u>Effective facewidth of gear</u> $(F_{eff,\,G})$ is taken as the minimum of the two

$$F_G \text{ and } \frac{2}{3}d_W$$

(which ever is smaller)

Table 14-2 RECOMMENDED PRESSURE ANGLES AND TOOTH DEPTHS FOR WORM GEARING

Lead angle λ, degrees	Pressure angle ϕ_n, degrees	Addendum a	Dedendum b_G
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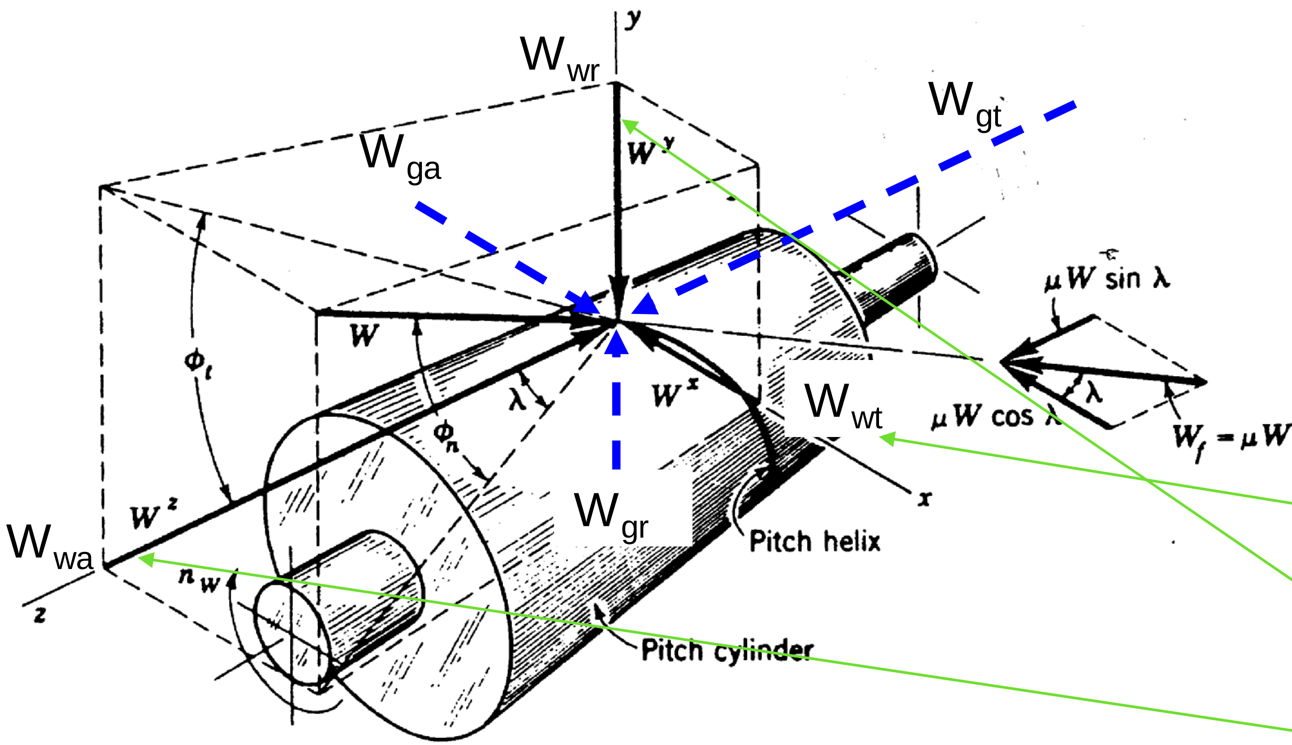
40–45	30	$0.2228p_x$	$0.2578p_x$

B) Tooth forces

$$W_{W_t} = -W_{G_a} = W^x$$

$$W_{W_r} = -W_{G_r} = W^y$$

$$W_{W_a} = -W_{G_t} = W^z$$



$$W^x = W \cos\phi_n \sin\lambda$$

$$W^y = W \sin\phi_n$$

$$W^z = W \cos\phi_n \cos\lambda$$

FIGURE 14-15 Drawing of the pitch cylinder of a worm, showing the forces exerted upon it by the worm gear.

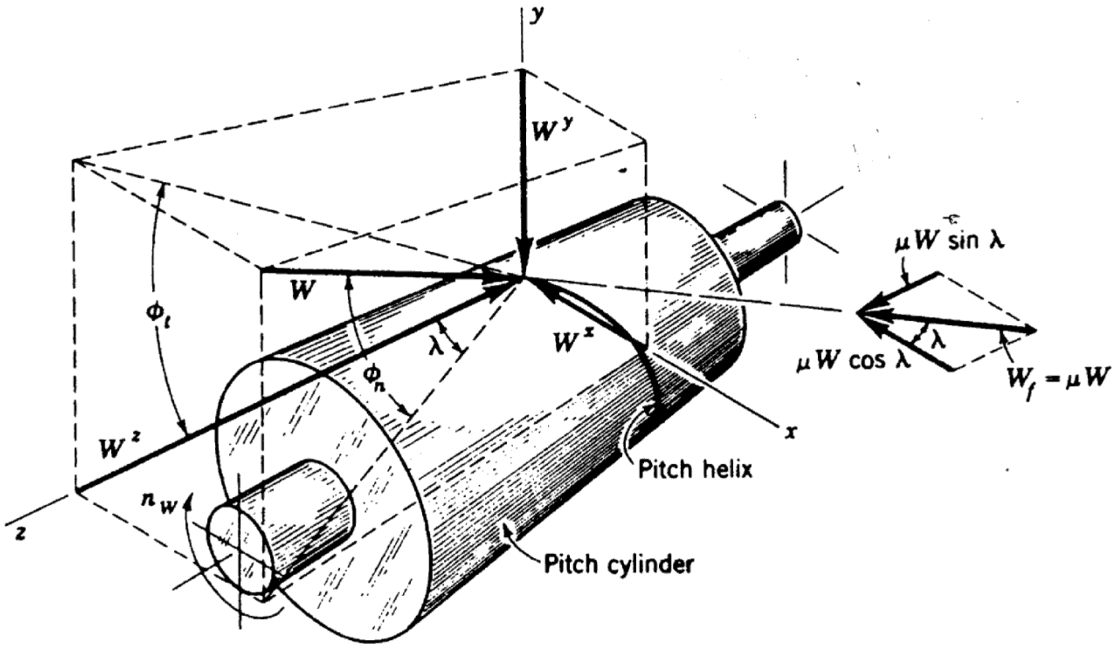


FIGURE 14-15 Drawing of the pitch cylinder of a worm, showing the forces exerted upon it by the worm gear.

- The relative motion between worm and worm-gear teeth is <u>pure sliding</u>, and so we must expect that friction plays an important role in the performance of worm gearing.
- By introducing a coefficient of friction μ we can develop another set of relations.
- In Fig. 14-15 we see that the force <u>*W* acting normal to the worm-tooth profile</u> produces a frictional force

  $W_f = \mu * W$

  having a component

  (μ W cos λ)

  in the negative *x* direction and
- another component

  (μ W sin λ)

  in the positive *z* direction.
- Equation therefore becomes

$$W^x = W(\cos\phi_n \sin\lambda + \mu\cos\lambda)$$

$$W^y = W\sin\phi_n$$

$$W^z = W(\cos\phi_n \cos\lambda - \mu\sin\lambda)$$

$$W_{W_t} = -W_{G_a} = W^x \longrightarrow W^x = W(\cos\phi_n \sin\lambda + \mu\cos\lambda)$$

$$W_{W_r} = -W_{G_r} = W^y \longrightarrow W^y = W\sin\phi_n$$

$$W_{W_a} = -W_{G_t} = W^z \longrightarrow W^z = W(\cos\phi_n \cos\lambda - \mu\sin\lambda)$$

$$W_f = \mu W = \frac{\mu W_{Gt}}{\mu\sin\lambda - \cos\phi_n\cos\lambda}$$

$$W_{Wt} = W_{Gt}\frac{\cos\phi_n\sin\lambda + \mu\cos\lambda}{\mu\sin\lambda - \cos\phi_n\cos\lambda}$$

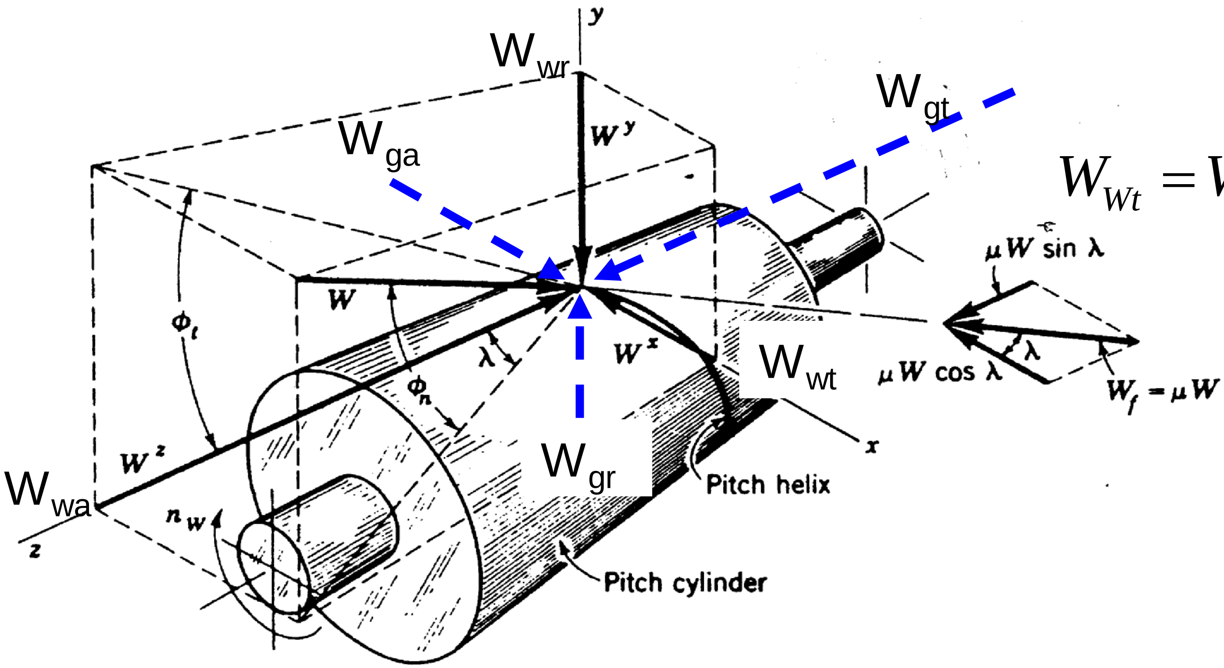


FIGURE 14-15 Drawing of the pitch cylinder of a worm, showing the forces exerted upon it by the worm gear.

Efficiency η can be defined by using the equation

$$\eta = \frac{W_{Wt}(without\ friction);\ \mu = 0}{W_{Wt}(with\ friction)}$$

$$W_{Wt} = W_{Gt}\frac{\cos\phi_n \sin\lambda + \mu\cos\lambda}{\mu\sin\lambda - \cos\phi_n\cos\lambda}$$

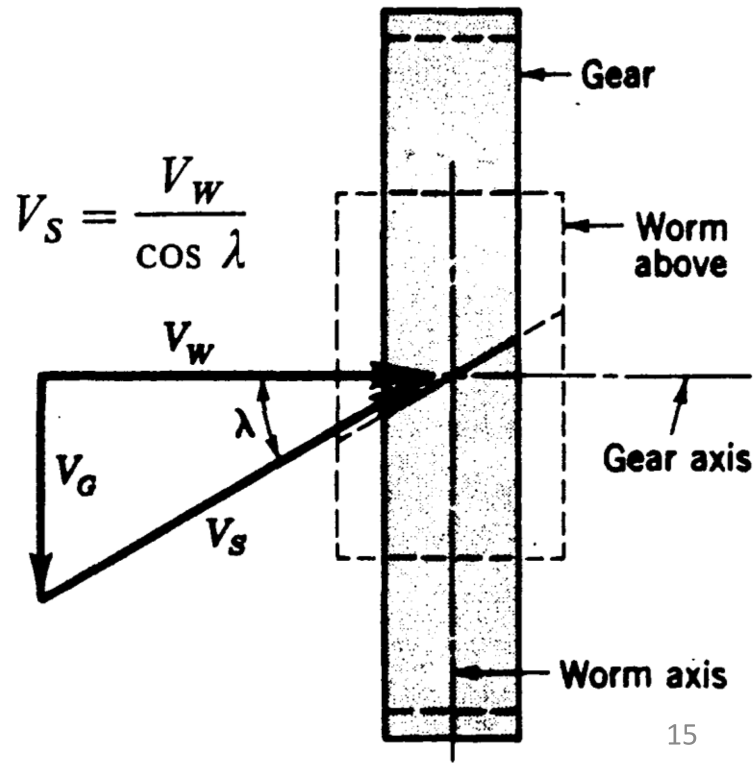
$$\cot\lambda = \frac{\cos\lambda}{\sin\lambda}$$

$$\eta = \frac{\cos\phi_n - \mu\tan\lambda}{\cos\phi_n + \mu\cot\lambda}$$

For 90° shaft angle $\lambda_W = \Psi_G$ and $p_{xW}=p_{tG}$

Table 14-3 EFFICIENCY OF WORM GEARSETS FOR $\mu = 0.05$

Helix angle ψ, of gear degrees	Efficiency η, percent
1.0	25.2
2.5	46.8
5.0	62.6
7.5	71.2
10.0	76.8
15.0	82.7
20.0	86.0
25.0	88.0
30.0	89.2



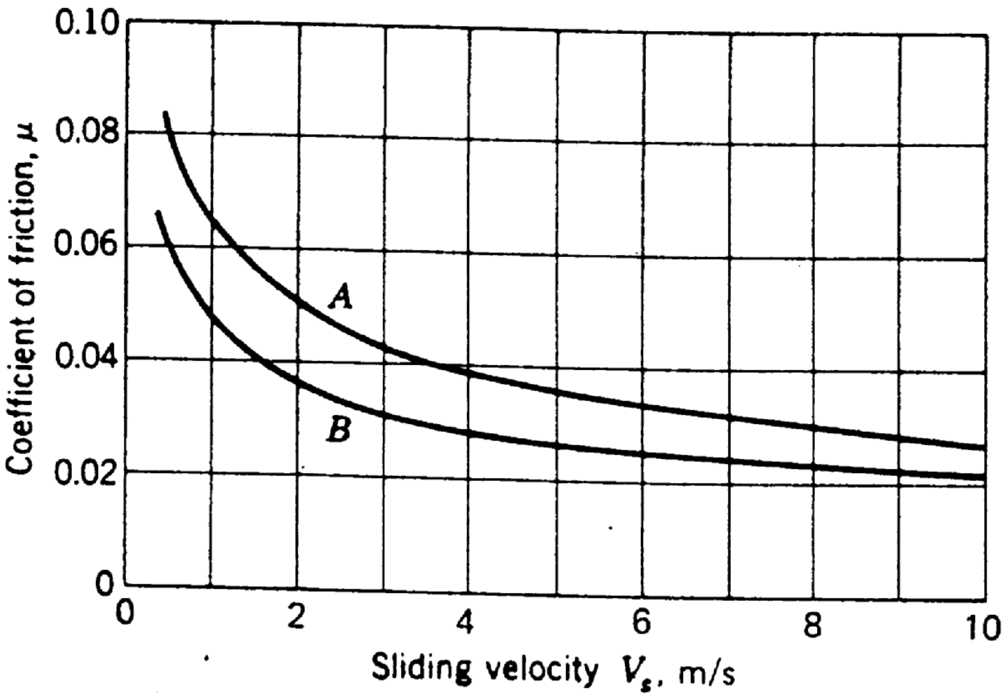
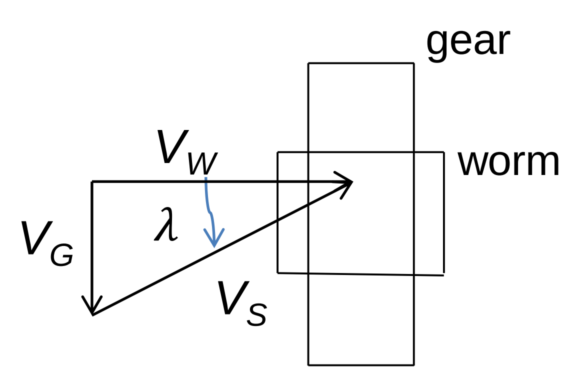
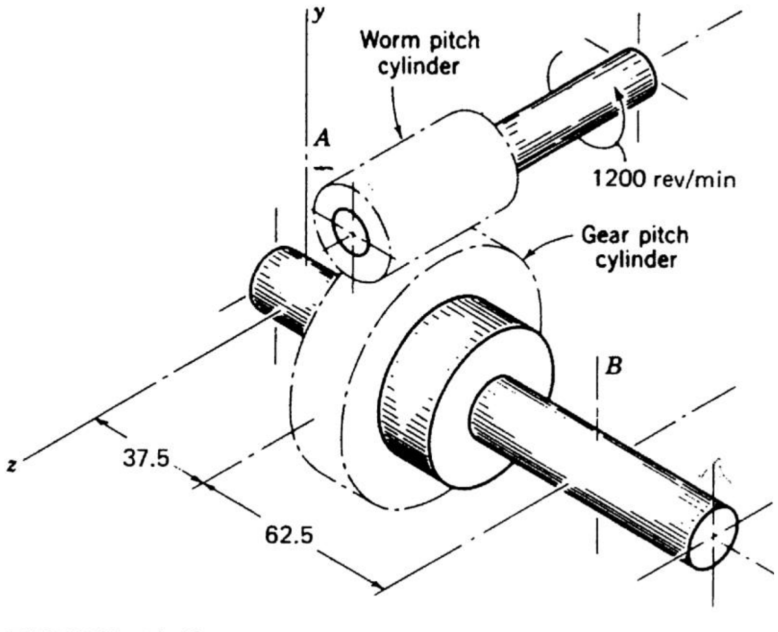


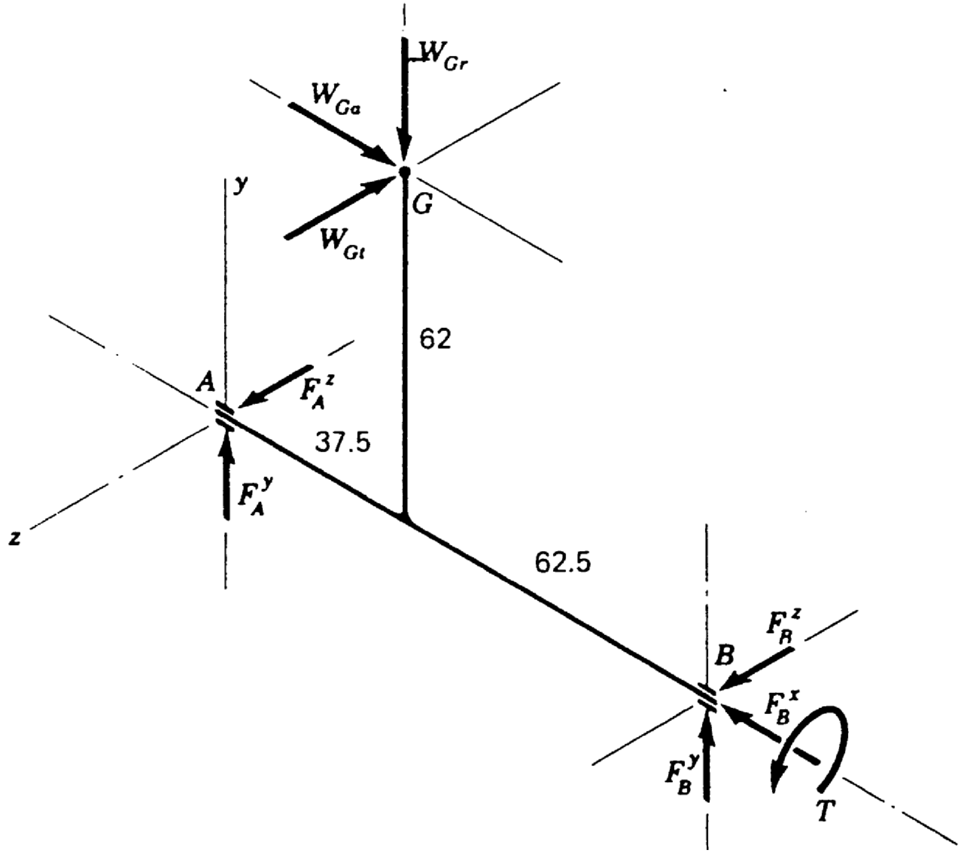
FIGURE 14-17 Representative values of the coefficient of friction for worm gearing. These values are based on good lubrication. Use curve *B* for high-quality materials, such as a case-hardened worm mating with a phosphor-bronze gear. Use curve *A* when more friction is expected, as for example a cast-iron worm and worm gear.

$$V_s = \frac{V_W}{\cos\lambda} \qquad V_W = \frac{\pi d_W N_W}{60}\ (m/\sec) \qquad V_G = \frac{\pi d_G N_G}{60}\ (m/\sec)$$

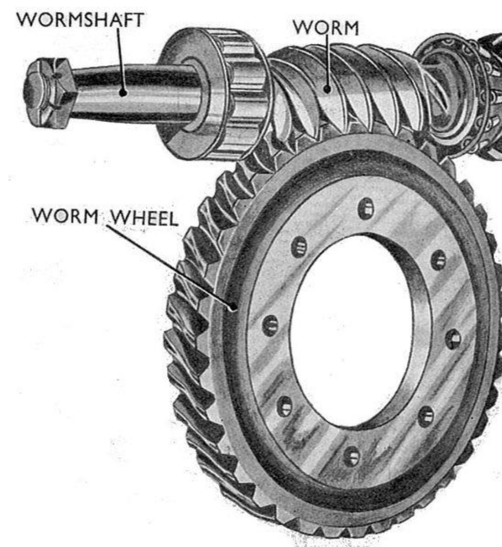
$$V_W = V_s \cos\lambda \qquad V_G = V_s \sin\lambda$$







- High gear tangential forces occur due to low speed of gear.
- Due to action reaction equality high axial forces occur on worm too.
- These high axial forces are usually carried by the strong tapered roller bearings of the worm shaft



Due to sliding between worm and gear teeth a friction force W_f occurs on the teeth as seen below in addition to the 3 components W_x ,W_y *and* W_z *of W.*

Hence final force components on the worm teeth are:

$$W_x : W_{W_t} = W \times \cos\phi_n \sin\lambda + \mu \times W \times \cos\lambda$$

$$W_y : W_{W_r} = W \times \sin\phi_n$$

$$W_z : W_{W_a} = W \times \cos\phi_n \cos\lambda - \mu \times W \times \sin\lambda$$

Opposite but equal magnitude forces occur on the gear teeth:

$$W_{G_t} = \text{-}W_{W_a} \rightarrow$$

$$W_{G_t} = \mu W \sin\lambda - W\cos\phi_n \cos\lambda = W(\mu \sin\lambda - \cos\phi_n \cos\lambda) \quad \textbf{(1)}$$

$$W_{G_r} = -W_{W_r} \rightarrow$$

$$W_{G_a} = -W_{W_t}, \quad W_{W_t} = W(\cos\phi_n \sin\lambda + \mu\cos\lambda) \quad \textbf{(2)}$$

By using 1 & 2

$$W_{Wt} = W_{Gt}\,\frac{\cos\phi_n \sin\lambda + \mu\cos\lambda}{\mu\sin\lambda - \cos\phi_n \cos\lambda}$$

Also

$$W_{Wt} = \frac{\text{power input to worm}}{V_w\left(=\frac{\pi d_w N_w}{60}\right)}$$

$$\text{Power output} = \eta \times \text{Power input}$$

Since there is sliding velocity in worm-gear sets hence a loss due to friction between the teeth efficiency of worm gear sets is defined as:

$$\eta = \frac{W_{Wt}(without\ \ friction);\ \mu = 0}{W_{Wt}(with\ \ friction)}$$

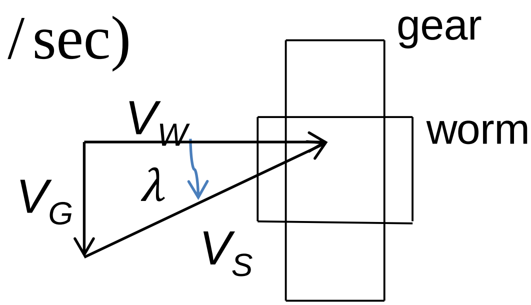
$$\eta = W_{Gt}\,\frac{\cos\phi_n - \mu\tan\lambda}{\cos\phi_n + \mu\cot\lambda} \qquad \cot\lambda = \frac{\cos\lambda}{\sin\lambda}$$

For a typical μ value of 0.05 and standard $\varnothing_n$ values in Table 14.2 of gear.

ψ_G	1°	5°	10°	15°	20°	25°	30°
η%	25	62	77	83	86	88	89

Coefficient of friction μ is dependent on sliding velocity V_s and given in figure 14.17.

$$V_s = \frac{V_W}{\cos\lambda} \qquad V_W = \frac{\pi d_W N_W}{60}\,(m/\sec) \qquad V_G = \frac{\pi d_G N_G}{60}\,(m/\sec)$$



$$V_w = V_s \cos\lambda \qquad V_g = V_s \sin\lambda$$

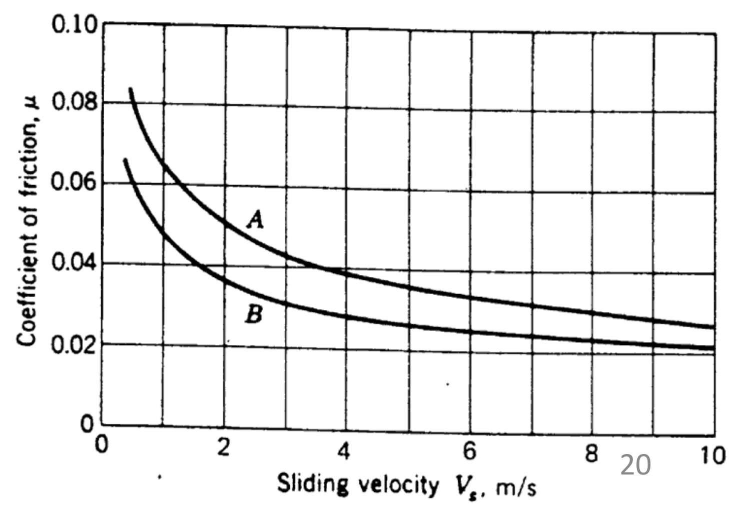
In Amer. System.

(inches)

$$V_W = \frac{\pi d_W N_W}{12}\,(fpm)$$

(rpm)

$$V_G = \frac{\pi d_G N_G}{12}\,(fpm)$$

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Tooth Stress and Power Rating of Worm Gearing

Generally the teeth of the worm is stronger than the teeth of the gear in terms of bending strength. Thus the bending fatigue failure of worm gear set is based on the bending strength (stress) value of the gear tooth.

Buckingham adopts the Lewis equation of tooth stress as follows for the gear tooth stress.

$$\sigma = \frac{W_{Gt}}{p_n F_G y} \qquad (14.28)$$

$$p_n = p_x \times \cos\lambda \qquad (14.29)$$

since $p_{xw} = p_{tG}$ & $\lambda_w = \psi_G$

for 90^{o} worm-gear set

where σ = bending stress, psi

W_{Gt} = transmitted load, lb

p_n = normal circular pitch of the gear, in

p_x = axial circular pitch of the worm, in

F_G = (*effective*) face width of gear, in

y = Lewis form factor referred to the circular pitch

λ = lead angle of worm = ψ of gear

Since the equation is only a rough approximation, stress concentration is not considered. Also, for this reason, the form factors are not referred to the number of teeth, but only to the normal pressure angle. The values of *y* are listed in Table 14.4.

Table 14-4 VALUES OF *y* FOR WORM GEARS

Normal pressure angle ϕ_n, degrees	Form factor y
$14\frac{1}{2}$	0.100
20	0.125
25	0.150
30	0.175

In (SI) module.

$$\sigma = \frac{W_{Gt}}{F_G \times y \times p_{nG}} = \frac{W_{Gt}}{F_G \times y \times \pi . m_{nG}}$$

$$\sigma = \frac{W_{Gt}}{F_G \times y \times \pi . m_{tG} . \cos \psi_G}$$

$$m_{tG} = \frac{P_{tG}}{\pi} = \frac{P_{xw}}{\pi}$$

The AGMA equation for input-horse-power rating of worm gearing is:

$$H = \underbrace{\frac{W_{Gt} \times d_G \times N_W}{126000 \times m_G}}_{\text{Output horse power from gear shaft}} + \underbrace{\frac{V_S \times W_f}{33000}}_{\text{Power loss due to friction between teeth of Worm and Gear.}} \qquad (14.30)$$

$$\begin{aligned}
W_{Gt} &= \text{transmitted load, lb} \\
d_G &= \text{pitch diameter of the gear, in} \\
N_W &= \text{speed of worm, rpm} \\
m_G &= \text{gear ratio } T_G / T_W \\
V_S &= \text{sliding velocity at mean worm diameter, fpm} \\
W_f &= \text{frictional force, lb}
\end{aligned}$$

The permissible transmitted load W_{Gt} (transmitted load by the gear) in power equation is calculated by the equation

$$W_{Gt} = d_G^{0.8} \times F_e \times K_s \times K_m \times K_v \quad (14.31)$$

The notation of Eqs. (14.30) and (14.31) is as follows:

W_{Gt} = transmitted load, lb

d_G = pitch diameter of the gear, in

N_W = speed of worm, rpm

m_G = gear ratio T_G / T_W

V_S = sliding velocity at mean worm diameter, fpm

W_f = frictional force, lb

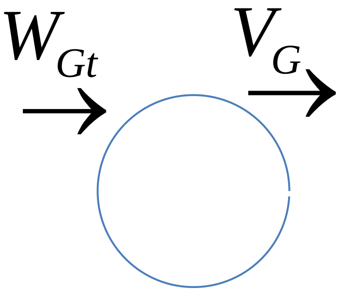
F_e = effective face width of gear; the effective face width is the face width of the gear or two-thirds of the worm pitch diameter, whichever is less

K_s = materials and size correction factor (Table 14.5)

K_m = ratio correction factor (Table 14.6)

K_v = velocity factor (Table 14.7)

Module, m_n in SI



$$H_{out} = W_{Gt} \times V_G$$

$$V_G = \frac{\pi d_G N_G}{60}; \quad N_G = N_w \frac{T_w}{T_G}$$

$$H = H_{out} + H_{loss}$$

$$H = W_{Gt} \times V_G + W_f \times V_s$$

$$V_G = \frac{\pi d_G}{60} \times N_w \times \left(\frac{T_w}{T_G} \right)$$

$$= \frac{1}{m_G}$$

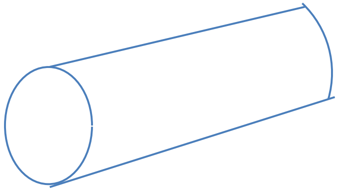
Gear Ratio

$$H_{out} = W_{Gt} \left(\frac{d_G N_w}{m_G} \right) \times \frac{\pi}{60}$$

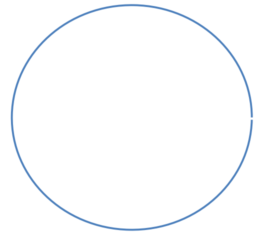
$$H_{loss} = V_s W_f = \frac{V_w}{\cos \lambda} \mu W$$

$$H_{loss} = W_{Gt} \frac{\mu}{\mu \sin \lambda - \cos \phi_n \sin \lambda} \times \frac{V_w}{\cos \lambda}$$

Example 8.1 (14-19)



2T worm,case hardened



40T sand cast bronze gear

$$P = 10 \; teeth/inches$$

$$\phi_n = 14.5^o$$

$$\lambda = 9.083^o$$

$$d_W = 1.25^{"}$$

$$F_W = 2^{"}$$

Worm teeth are ground and polished

$$F_G = \frac{5^{"}}{8}$$

$$d_G = 4^{"}$$

$$N_W = 1720 \; rpm$$

Find maximum safe horse power output and efficiency of the set.

$$H_{out} = ?$$

$$\eta = ?$$

Force components on gear?

$$W = ?$$

$$W_{G_t} = ?$$

$$W_{G_a} = ?$$

$$W_{G_r} = ?$$

1)

$$H_{out} = \frac{W_{Gt} \times d_G \times N_W}{126000 \times m_G} = ?$$

$$H_{loss} = \frac{V_S \times W_f}{33000}$$

$$W_{Gt} = d_G^{0.8} \times F_e \times K_s \times K_m \times K_v$$

700 (Table 14.5)

4"

$\frac{2}{3} d_w = 0.833''$

$F_G = 0.625''$ $\therefore smaller$

0.820 (m_G=40/2=20)

Table 14.6

$$\eta = \frac{H_{out}}{H_{in}} = \frac{H_{out}}{H_{out} + H_{loss}} = ?$$

$$W_{Gt} = (4)^{0.8} \times 0.625 \times 700 \times 0.820 \times 0.352$$

$$W_{Gt} = 383\, lb$$

$$V_s = \frac{V_W}{\cos\lambda} = \frac{\frac{\pi d_W N_W}{12}}{\cos\lambda} = \frac{\frac{\pi \times 1.25 \times 1720}{12}}{\cos(9.083)}$$

$$H_{out} = \frac{383 \times 4 \times 1720}{126000 \times 20} = \underline{\underline{1.045\, hp}}$$

$$V_s = 570\ fpm \rightarrow K_v = 0.352\ (Table\, 14.7)$$

$$V_W = \frac{\pi d_W N_W}{12}\ (fpm)$$

$$V_G = \frac{\pi d_G N_G}{12}\ (fpm)$$

Thus $\mu = 0.032 \rightarrow$ from Fig.14.17 curve B for $V_s = 2.8956\ m/s$

$$W_f = \mu W = \frac{\mu W_{Gt}}{\mu \sin\lambda - \cos\phi_n \cos\lambda}$$

$$V_s = 570\ fpm \times \frac{\min}{60\sec} \times \frac{0.3048\ m}{ft}$$

$$V_s = 2.8956\ m/s$$

$$W_f = \frac{0.032 \times 383}{0.032 \times \sin(9.083) - \cos(14) \times \cos(9.083)} = -12.9\ lb$$

$$H_{loss} = \frac{V_S \times W_f}{33000} \qquad H_{loss} = \frac{570 \times 12.9}{33000} = 0.222\ hp$$

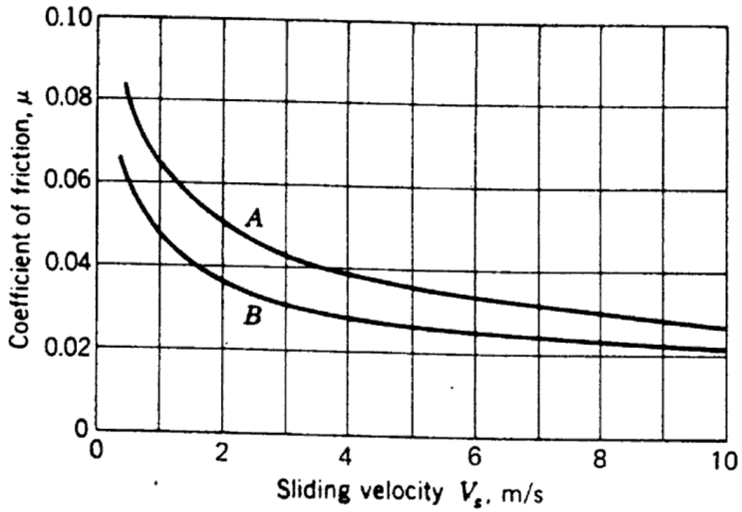
Maximum safe horse power output

Efficiency of the set.

$$\eta = \frac{H_{out}}{H_{in}} = \frac{H_{out}}{H_{out} + H_{loss}} = \frac{1.045}{1.045 + 0.222}$$

$$\eta = \underline{\underline{0.825}} = 82.5\ \%$$

Coefficient of friction μ is dependent on sliding velocity V_s and given in Figure 14.17.



Force components on gear

$$W_{Gt} = W \cos\phi_n \cos\lambda$$

$$W = \frac{W_{Gt}}{\cos\phi_n \cos\lambda} = \frac{383}{\cos(14.5)\times\cos(9.083)} = 400.62\ lb$$

$$W_{G_a} = W \times \cos\phi_n \times \sin\lambda = 400.62 \times \cos(14.5) \times \sin(9.083)$$

$$W_{G_a} = \underline{\underline{61.23\ lb}}$$

$$W_{G_r} = W \times \sin\phi_n = 400.62 \times \sin(14.5)$$

$$W_{G_r} = \underline{\underline{100.3\ lb}}$$

14-8 POWER RATING OF WORM GEARING

When worm gearsets are used intermittently or at slow gear speeds, the bending strength of the gear tooth may become a principal design factor. Since the worm teeth are inherently stronger than the gear teeth, they are usually not considered, though the methods of Chap. 8 can be used to compute worm-tooth stresses. The teeth of worm gears are thick and short at the two edges of the face and thin in the central plane, and this makes it difficult to determine the bending stress. Buckingham* adapts the Lewis equation as follows:

$$\sigma = \frac{W_{Gt}}{p_n F_G y} \qquad (14\text{-}28)$$

$$p_n = p_x \cos \lambda \qquad (14\text{-}29)$$

where σ = bending stress, psi
W_{Gt} = transmitted load, lb
p_n = normal circular pitch, in of the gear
p_x = axial ~~circular~~ pitch, in of the worm
F_G = face width of gear, in
y = Lewis form factor referred to the circular pitch
λ = lead angle

Since the equation is only a rough approximation, stress concentration is not considered. Also, for this reason, the form factors are not referred to the number of teeth, but only to the normal pressure angle. The values of y are listed in Table 14-4.

The AGMA equation for *input-horsepower rating* of worm gearing is

$$H = \frac{W_{Gt} d_G n_W}{126\,000 m_G} + \frac{V_S W_f}{33\,000} \qquad (14\text{-}30)$$

The first term on the right is the *output horsepower*, and the second is the *power loss*.

Table 14-7 VELOCITY FACTOR K_v

Velocity V_S, fpm	K_v	Velocity V_S, fpm	K_v	Velocity V_S, fpm	K_v
1	0.649	300	0.472	1400	0.216
1.5	0.647	350	0.446	1600	0.200
10	0.644	400	0.421	1800	0.187
20	0.638	450	0.398	2000	0.175
30	0.631	500	0.378	2200	0.165
40	0.625	550	0.358	2400	0.156
60	0.613	600	0.340	2600	0.148
80	0.600	700	0.310	2800	0.140
100	0.588	800	0.289	3000	0.134
150	0.558	900	0.269	4000	0.106
200	0.528	1000	0.258	5000	0.089
250	0.500	1200	0.235	6000	0.079

Source: Darle W. Dudley (ed.), *Gear Handbook*, McGraw-Hill, New York, 1962.

Table 14-4 VALUES OF y FOR WORM GEARS

Normal pressure angle ϕ_n, degrees	Form factor y
$14\frac{1}{2}$	0.100
20	0.125
25	0.150
30	0.175

The *permissible* transmitted load W_{Gt} is computed from the equation

$$W_{Gt} = K_s d_G^{0.8} F_e K_m K_v \qquad (1$$

The notation of Eqs. (14-30) and (14-31) is as follows:

W_{Gt} = transmitted load, lb
d_G = pitch diameter of gear, in
n_W = speed of worm, rpm
m_G = gear ratio, N_G/N_W
V_S = sliding velocity at mean worm diameter, fpm
W_f = frictional force, lb
K_s = materials and size-correction factor
F_e = effective face width of gear; the effective face width is the face width of the gear or two-thirds of the worm pitch diameter, whichever is less
K_m = ratio-correction factor
K_v = velocity factor

Table 14-5 MATERIALS FACTOR K_s FOR CYLINDRICAL WORM GEARING*

Face width of gear F_G, in	Sand-cast bronze	Static chill-cast bronze	Centrifugal-cast bronze
Up to 3	700	800	1000
4	665	780	975
5	640	760	940
6	600	720	900
7	570	680	850
8	530	640	800
9	500	600	750

Table 14-6 RATIO-CORRECTION FACTOR K_m

Ratio m_G	K_m	Ratio m_G	K_m	Ratio m_G	K_m
3.0	0.500	8.0	0.724	30.0	0.825
3.5	0.554	9.0	0.744	40.0	0.815
4.0	0.593	10.0	0.760	50.0	0.785
4.5	0.620	12.0	0.783	60.0	0.745
5.0	0.645	14.0	0.799	70.0	0.687
6.0	0.679	16.0	0.809	80.0	0.622
7.0	0.706	20.0	0.820	100.0	0.490