BACK TO BASICS...

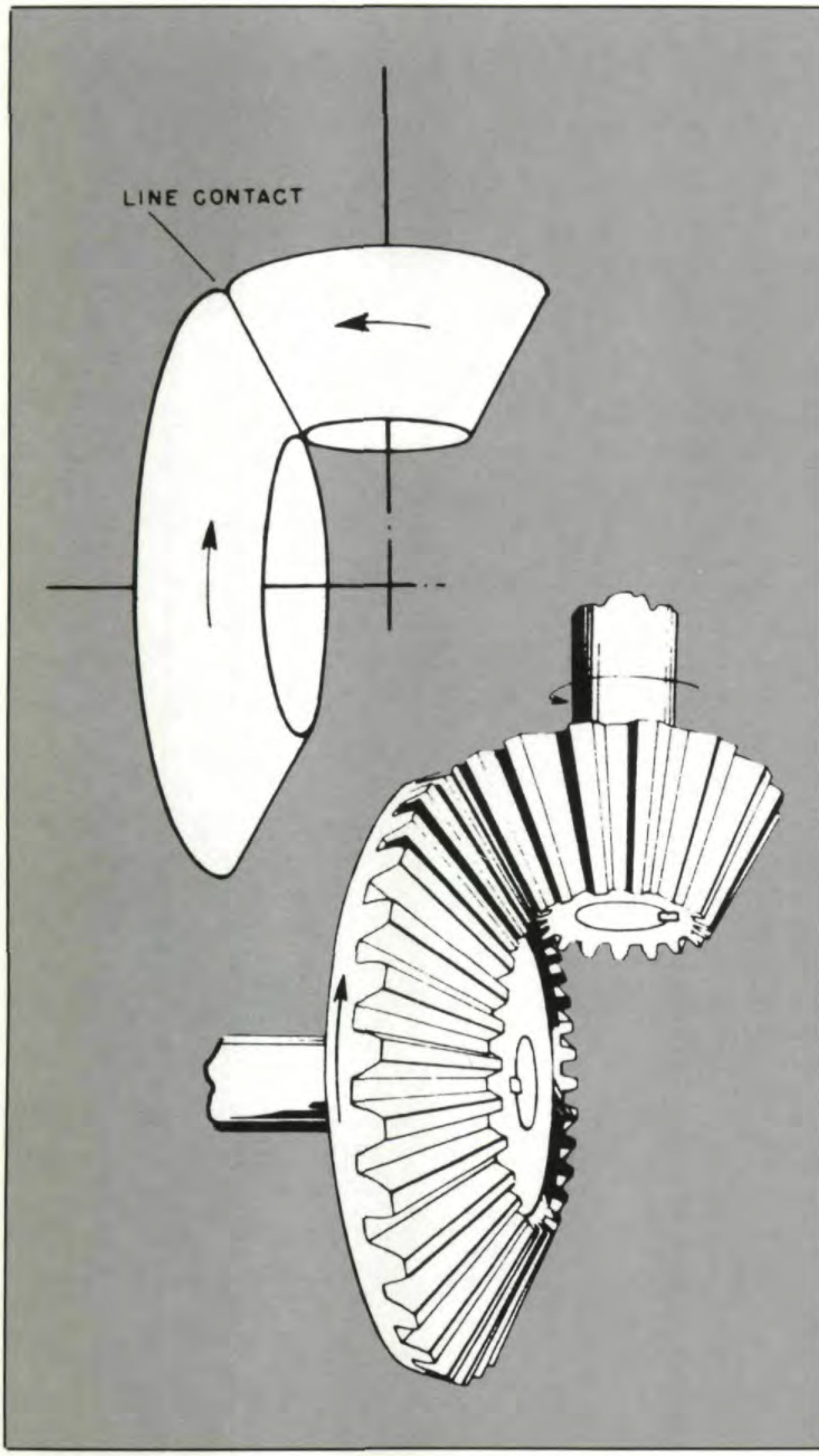

Fig. 1 (Left) – Bevel gears. These shafts intersect at a right angle, although bevel gears may also be used between shafts that intersect at larger or smaller angles. (Courtesy Mobil Oil Corporation.)

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Transmission of power between nonparallel shafts is inherently more difficult than transmission between parallel shafts, but is justified when it saves space and results in more compact, more balanced designs. Where *axial* space is limited compared to *radial* space, *angular* drives are preferred despite their higher initial cost. For this reason, angular gear motors and worm gear drives are used extensively in preference to parallel shaft drives, particularly where couplings, brakes, and adjustable mountings add to the axial space problem of parallel shaft speed reducers.

In angular drives, the gears not only rotate in different planes, but contact is frequently diagonal across the face of mating teeth (Fig. 1). Such gears are generally more difficult to design, manufacture, and install and thus cost more than equivalent spur and parallel-axis helical gears. They are also more sensitive to mounting and manufacturing errors. In addition, mounting on overhanging shafts makes some types sensitive to shaft deflections. As with spur and helical gearing, optimum design becomes a compromise. Two general classifications cover the right-angle gears: *coplanar* types, which have intersecting axes, and *offset* types, which have nonintersecting or skew axes (their axes do not lie in a common plane).

Bevel Gears

Bevel gears provide the most *efficient* means of transmitting power between intersecting shafts. The related friction wheels are frustrums of cones, and the

Gears For Nonparallel Shafts

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gears developed on these conical surfaces are called bevel gears. If teeth are cut straight across the faces of conical blanks, the gears are called *straight* bevel; when the teeth are twisted along a curved path, the gears are termed *spiral* bevel (Figs. 1 and 2.) The involute tooth form is used.

Customarily, tooth dimensions are determined in a transverse plane (perpendicular to the common element of the pitch cones) at the *large* end of the teeth (Fig. 3*a*). Intersection of tooth surfaces with this plane gives a tooth profile as shown in Fig. 3*b*. Fig. 3*a* also shows that the shaft angle equals the sum of the pitch angles. Thus

$$\sum = \Gamma_P + \Gamma_G \qquad (1)$$

where

Σ(sigma) = shaft angle; deg
Γ_P(gamma) = pitch angle of pinion; deg
Γ_G(gamma) = pitch angle of gear; deg

Fig. 3*c* indicates that the following relationship exists for $\Sigma = 90$ deg.

$$\tan \Gamma_P = \frac{d}{D} = \frac{N_P}{N_G} \qquad (2)$$

$$\tan \Gamma_G = \frac{D}{d} = \frac{N_G}{N_P} \qquad (3)$$

where

d = pitch diameter of pinion; mm, in.
D = pitch diameter of gear; mm, in.

N_p = number of teeth in pinion
N_G = number of teeth in gear

Nomenclature and symbols commonly used for straight bevel gears are shown in Fig. 4. The similarity to spur gearing should be noted. With few exceptions, the nomenclature also applies to spiral bevel gears.

Equivalent spur gears is a term commonly used in connection with bevel gears. Two bevel gears roll on each other in the same manner as a pair of spur gears with pitch diameters equal to those of the bevel gears. In Fig. 4 the equivalent spur gear would have a pitch diameter D_G.

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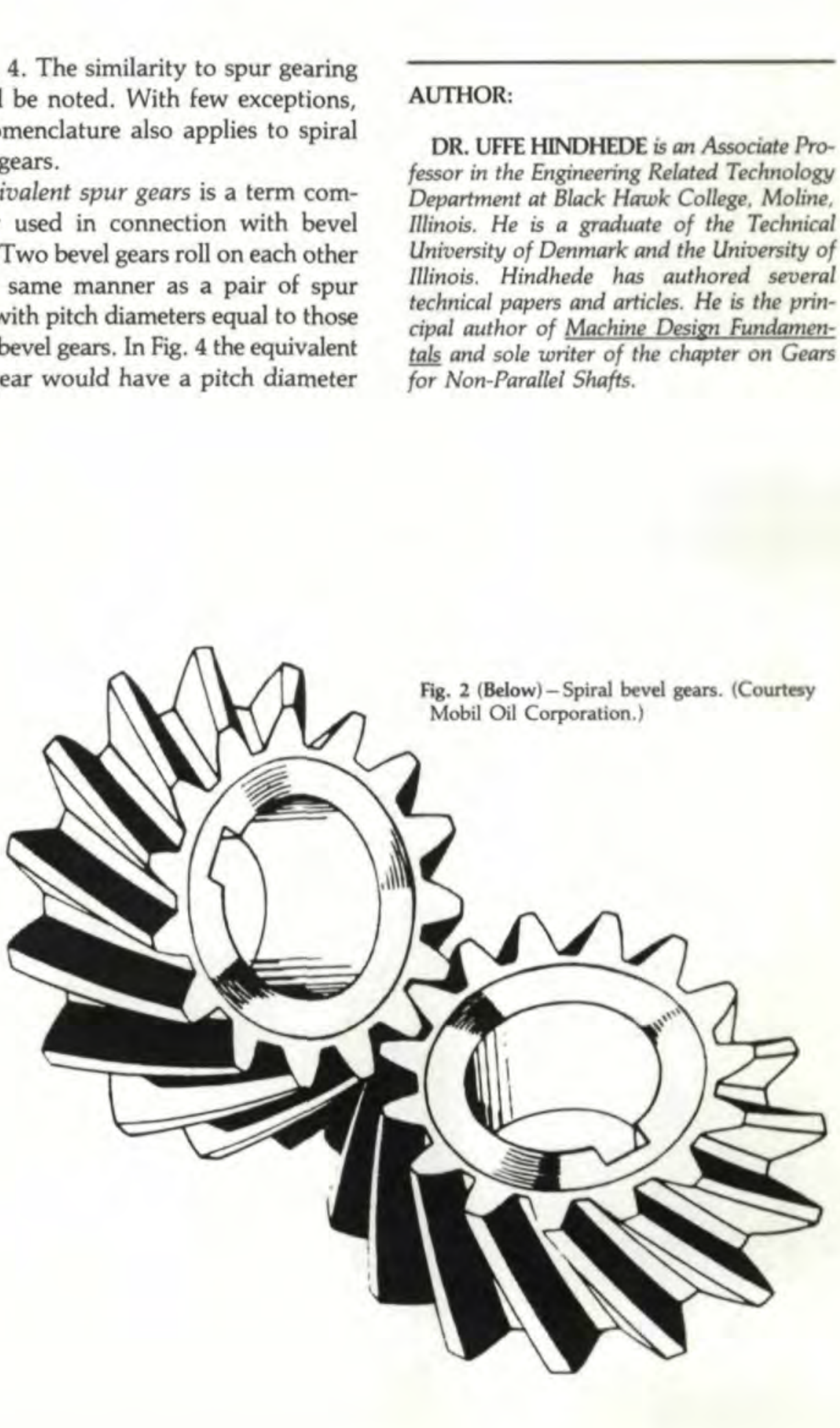


Fig. 2 (Below) – Spiral bevel gears. (Courtesy Mobil Oil Corporation.)

Nomenclature

C = center distance
d = pitch diameter of bevel pinion and worm
D = pitch diameter of bevel gear or worm gear
D_G = pitch diameter of worm gear
D_w = pitch diameter of worm
e = efficiency
L = lead of worm
m = module
m_G = speed ratio
n_G, n_w = speed of gear and worm, respectively
N_p = number of teeth in pinion
N_G = number of teeth in gear
N_w = number of teeth in worm
p_a = axial pitch
p_c = circular pitch
P_d = diametral pitch
TR = thermal rating
λ(lambda) = lead angle
Γ(gamma) = pitch angle
Σ(sigma) = shaft angle
ψ(psi) = helix angle
θ(theta) = friction angle

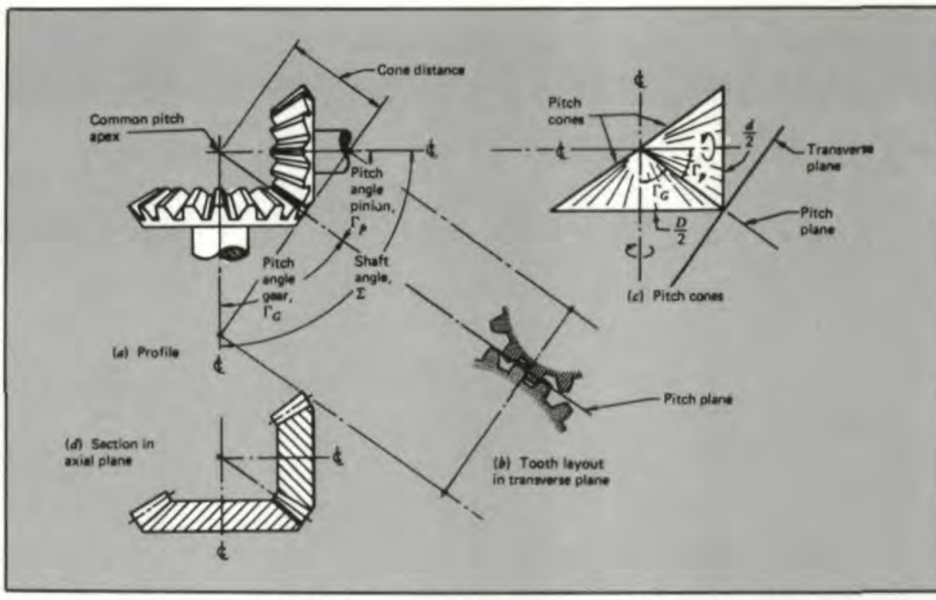


Fig. 3—Basic bevel gear sections. (Courtesy General Motors Corporation.)

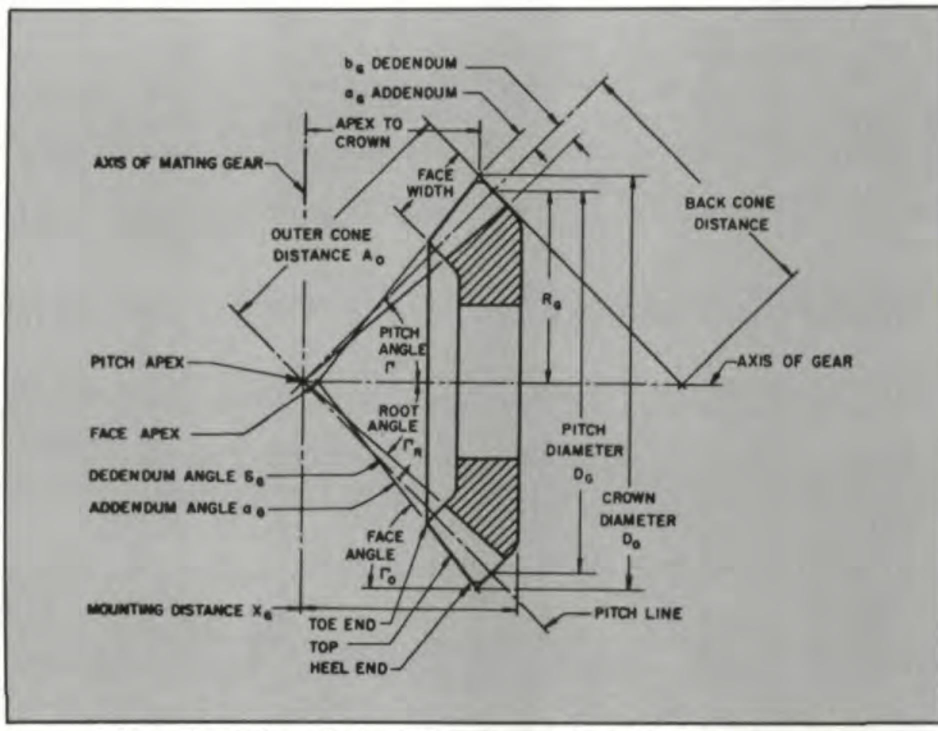


Fig. 4—Nomenclature for bevel gears. (Courtesy General Motors Corporation.)

The difference between spiral and straight bevel gears is that spiral teeth have a *gradual* pitch line contact and a larger number of teeth in contact. Their teeth, instead of engaging in a full line contact at once, engage with one another gradually. This continuous contact makes it possible to obtain smoother action than is possible with straight bevels.

Arrangement

Bevel gears are widely used where a right-angle change in direction of shafting is required, although the shafts occasionally may intersect at acute or obtuse angles (Fig. 5). When of equal size and mounted on shafts at right angles, they are referred to as *miter* gears (Fig. 5*b*). A bevel gear with a right (90 deg) pitch angle is a *crown* gear (Fig. 5*e*). Internal bevel gearing, like internal spur or helical gearing, is sometimes used in planetary or internal gear arrangements (Fig. 5*f*).

Application

Straight bevel gears, like spur gears, are well suited for manual and low-speed operations, such as in small hoists, valves, gates, or doors. When greater speed and more power are required, spiral bevel gears are preferable.

Mounting

Bevel gears require larger shaft diameters and heavier bearings because they impose high reaction loads on bearings.

Speed Ratio

As in spur gears, the speed ratio is the ratio of tooth numbers. For bevel gears, it is also the ratio of corresponding pitch radii or diameters of the pitch cones (Fig. 4).

Design of Bevel Gears

Bevel gears are simple modifications of spur gears. Beam straight and surface durability determine size and surface hardness. The AGMA formulas used are therefore simple modifications of the formulas used for calculating spur gears.

Example 1

A pair of straight-toothed bevel gears

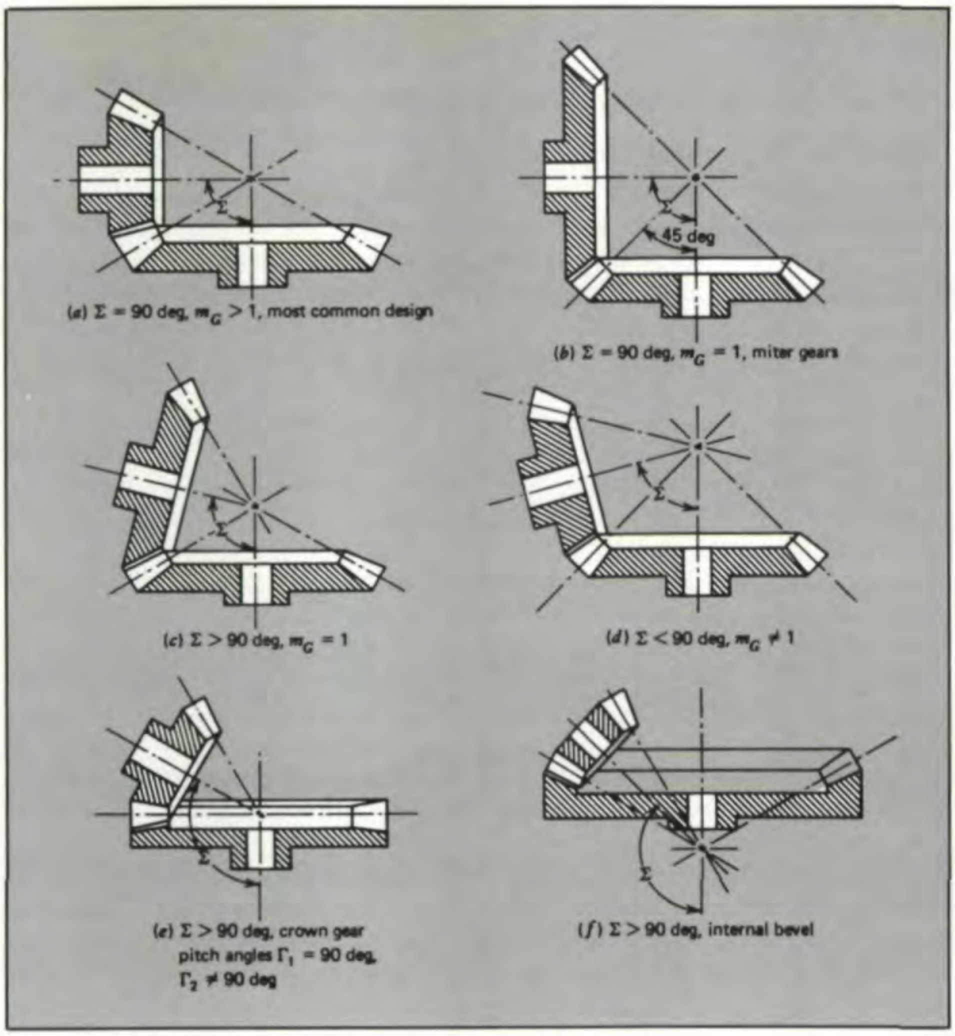
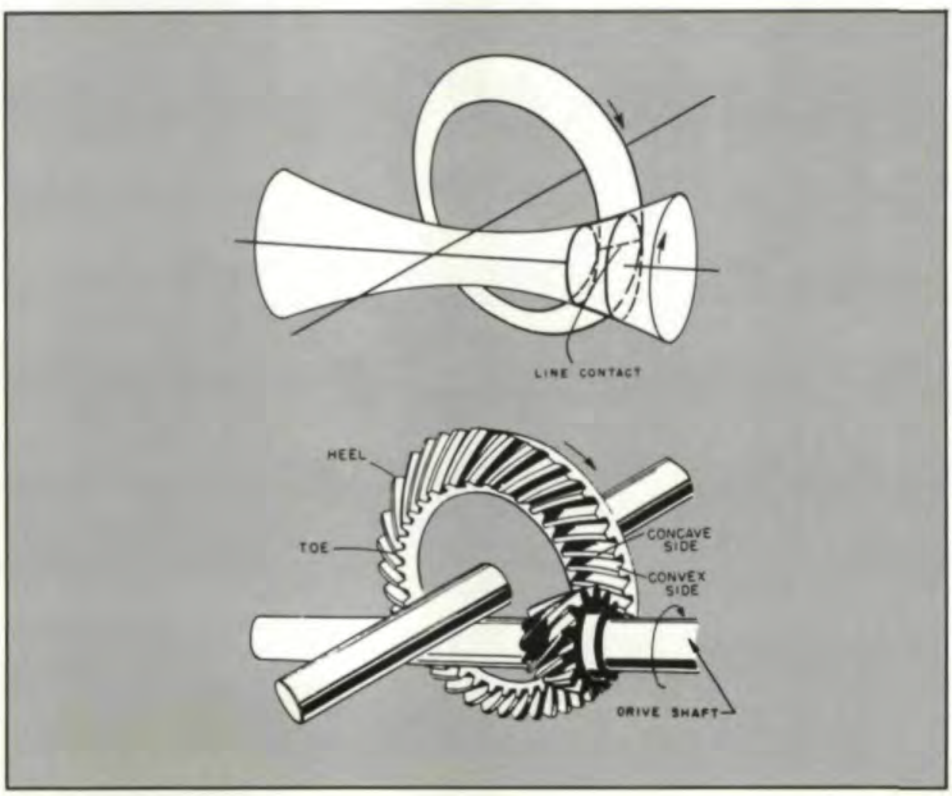


Fig. 5 – Bevel-gear arrangements. (Courtesy General Motors Corporation.)



are to be designed for a shaft angle of 90 deg and a reduction ratio of roughly 3:1. If the pinion is to have a minimum of 17 teeth, find pitch angles and the number of teeth in the gear.

Solution

$N = (3)(17) + 1 = $ **52** leads to a hunting ratio equal to 3.06:1. Assuming a module m, we obtain for the equivalent spur gears:

$$d = m\ (17\ \text{mm}) \qquad D = m\ (52\ \text{mm})$$

$$\tan \Gamma_p = \frac{N_p}{N_G} = \frac{17}{52} \longrightarrow \Gamma_p = \mathbf{18.1\ deg}$$

$$\tan \Gamma_G = \frac{N_G}{N_p} = \frac{52}{17} \longrightarrow \Gamma_G = \mathbf{71.9\ deg}$$

Check:

$$\sum = \Gamma_p + \Gamma_G = 18.1\ \text{deg} + 71.9\ \text{deg}$$

$$= 90\ \text{deg}$$

Hypoid Gears

Hypoid gears closely resemble spiral bevel gears except that the pinion does not meet the ring gear at its center. It meets it at a *lower* point (Fig. 6). The pitch surfaces are hyperboloids from which the term "hypoid" was derived. Curved teeth contribute to smooth, *noiseless* operation even at high speed.

Hypoid gears grew out of a need for *silent* automotive differentials that would also allow the drive shaft to be placed well below the centerline of the rear axle, thus contributing to a lower body design. Because the two shafts do not intersect, (1) two rear axles, instead of one, can be successively driven from the same transmission shaft, and (2) bearings can be mounted on both sides of the pinion. Although the altered tooth shape results in more sliding, lower efficiency, and the need for special lubricants, it provides a perfectly smooth drive and solves a major automotive problem – noise.

Helical Gearing

Helical gearing is a term applied to all types of gears whose teeth are of helical form. Helical gears work equally well to connect parallel shafts and nonparallel, nonintersecting shafts. In the latter case, however, a distinction must be made be-

Fig. 6 (Bottom Left) – Hypoid gear and pinion. These gears transmit motion between nonintersecting shafts crossing at a right angle. The pitch surfaces are hyperbolic in form. (Courtesy Mobil Oil Corporation.)

Fig. 7—Crossed helical gears. These gears transmit motion between nonintersecting shafts crossing at an acute angle. The teeth are developed on cylindrical pitch surfaces. Since only point contact exists between the gears, this arrangement is rarely used to transmit loads of any magnitude. (Courtesy of Mobil Oil Corporation.)

tween a gear and a worm, even though both have helical teeth. As seen in Fig. 7, the teeth on this gear make only a *fraction* of a revolution on the base cylinder. That the tooth curvature is helical is not even obvious. What the teeth lack in length, however, they make up for in number, which always exceeds 10.

Crossed Helical Gearing

Fig. 7 shows this form of gearing. For what they can do kinematically, crossed helical gears are the acme of simplicity. Tooth contact, however, is only a point which greatly limits their power transmission capability.

Worm Gearing

Worm gear drives are used on right-angle applications with nonintersecting shafts. They provide smooth, quiet action and maximum reduction ratios for a given center distance. These favorable characteristics are obtained by using a worm gear combination. As seen in Fig. 8, the gear has teeth inclined at the same angle as the threads in the worm. For speed reduction or torque amplification, the worm is the driver.

In a *worm*, the number of teeth *rarely* exceeds 10 (one to four teeth is common, as shown in Figs. 8 and 9). Each tooth, however, makes at least one revolution on the base cylinder. If only one tooth is used, it winds around the base cylinder several times like a screw thread.

Worm gearing derives its characteristics from the two simple machines of which it is composed: the screw and the lever. From the screw it obtains a *large* mechanical advantage but a somewhat *lower* efficiency because of larger friction forces. The conjugate tooth action is identical to that of a large spur gear and rack (Fig. 8). As the worm revolves, the thread form advances along its axis and the worm gear rotates a corresponding amount. The presence of a screw instead of a rack ensures a vastly greater output torque on the gear shaft. The net effect is a torque converter of superior capacity but, because of inherent sliding action, one of reduced efficiency.

Worm gear terms are shown in Fig. 9. In this particular case the *lead*, the distance advanced during one revolution, is three times greater than the pitch. Triple-threaded worms are more efficient than single-threaded worms (due to less friction), but their reduction ratio is only one-third that of the single-threaded worm.

Fig. 10 shows the basic difference between single- and double-thread worms. The slope or helix angle of the double-thread worm is *twice* that of the single-thread worm, as is the lead. Thus, for one revolution the double-thread worm will advance or turn its mating gear an angle *twice* that of the single-thread worm.

Tooth breakage from bending action is not prevalent in worm gear sets. With relatively high sliding velocities, the design criteria are usually based on scoring and pitting. *Scoring* is wear resulting from failure of the lubricant film due to localized overheating of the mesh, per-

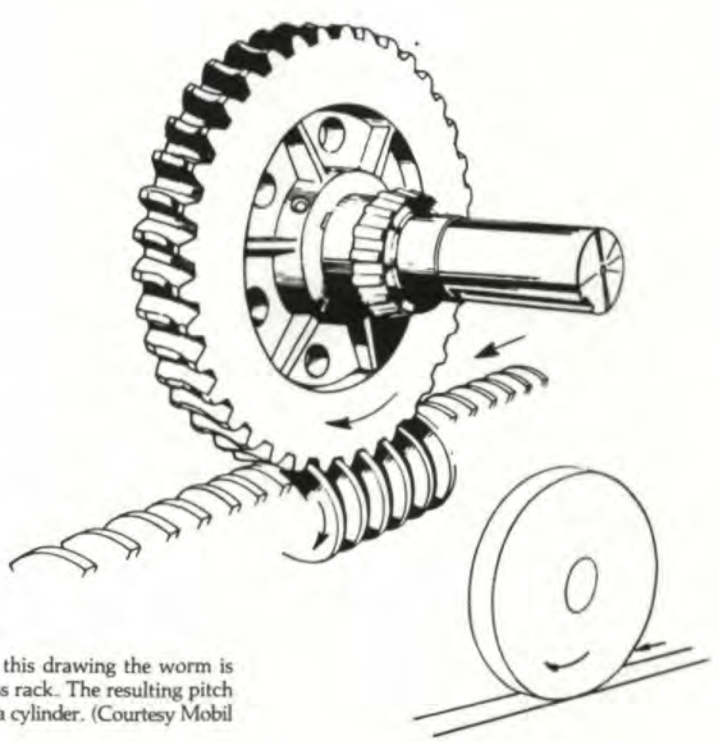


Fig. 8—Worm gear. In this drawing the worm is represented as an endless rack. The resulting pitch surfaces are a plane and a cylinder. (Courtesy Mobil Oil Corporation.)

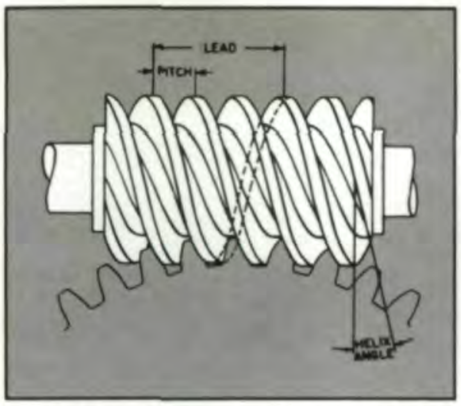


Fig. 9—Worm gear terms. (Courtesy Mobil Oil Corporation.)

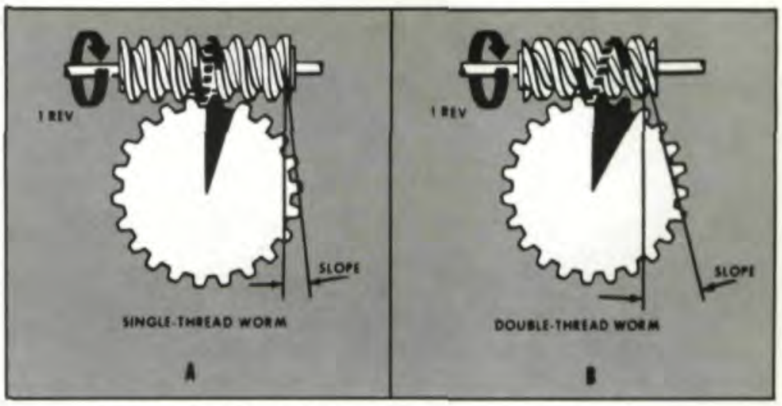


Fig. 10—The difference between a single- and a double-thread worm. (Courtesy Bureau of Naval Personnel.)

mitting metal-to-metal contact. Because more teeth are in contact simultaneously, worm gearing provides smoother operation than involute gearing. The contact area is also larger. Thus load capacity is high despite sliding action and line contact. Entering side wedges are produced on modern worm gears to generate a load-supporting oil film.

The advantages of worm gearing compared to spur and helical gearing are:

1. A more compact design for the same reduction ratio or power capacity.
2. Much greater speed reduction and torque amplification in a single step.
3. Smooth and silent operation that can withstand higher shock loads and higher momentary loads.

The disadvantages compared with ordinary gearing are:

1. Lower and varying efficiency.
2. Greater axial forces requiring costlier bearings.
3. Overheating that limits duration and capacity for power transmission.

Worm Gear Terminology and Kinematics

Fig. 11 shows worm gear terminology and development of a worm thread. For reasons of clarity, a triple-threaded worm was used. A worm can be single, double, triple, quadruple, or multi-threaded, plus left or righthanded. Threads in excess of 10 are rarely advantageous.

Axial pitch (p_a) of a worm (Fig. 11) is the distance measured axially from a point on one thread to the corresponding

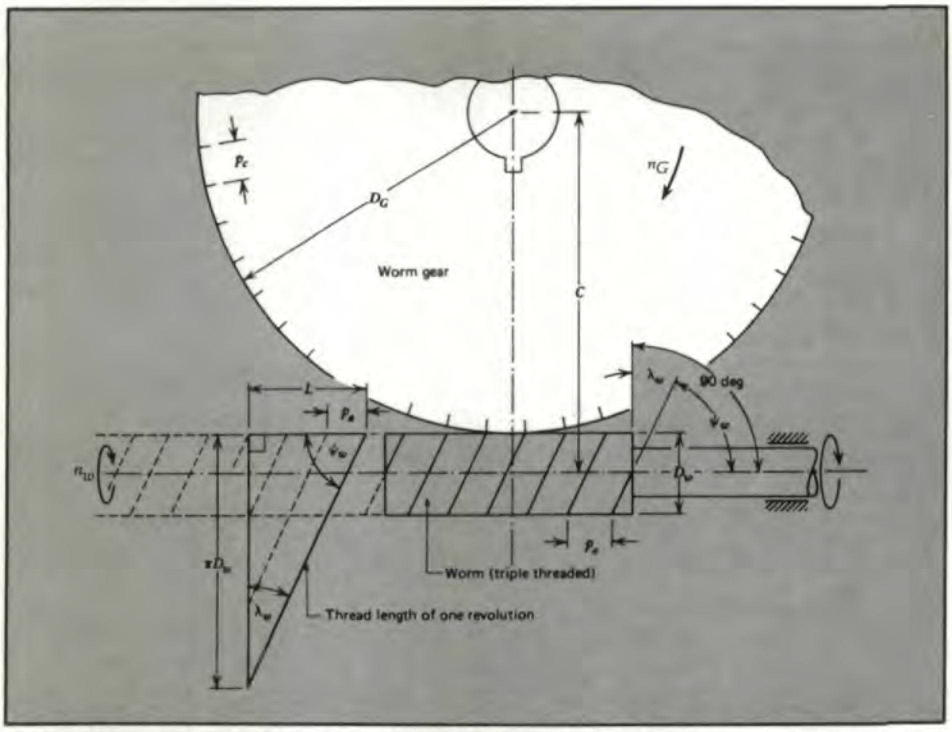


Fig. 11 – Worm gear terminology and development of a worm thread. (For clarity, a multithreaded worm was used.)

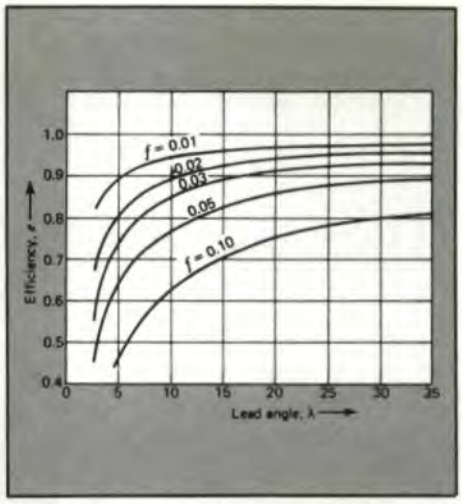


Fig. 12 – Efficiency as a function of lead angle and coefficient of friction.

point on the next thread. For proper mesh, p_a must equal the circular pitch p_c of the gear.

Lead (L) is the distance L that a thread advances in one turn of the worm. Thus

$$L = N_w p_a \quad (4)$$

where N is the number of threads on the worm; e.g., $N_w = 2$ for a double-threaded worm.

Lead angle (λ_w) is the angle between a tangent to the thread at the pitch diameter and a plane normal to the worm axis.

Velocity ratio (m_G) is the ratio of pitch circumference of gear to lead of worm, which equals tooth ratio. It is also the ratio of worm speed to gear speed. Thus

$$m_G = \frac{p_a N_G}{L} = \frac{N_G}{N_w} = \frac{n_w}{n_G} = \frac{D_G}{D_w \tan \lambda_w} \quad (5)$$

For worm and gear to mesh properly, the lead angle of the worm must equal the helix angle of the gear ($\lambda_w = \psi_G$), and axial pitch of the worm must equal circular pitch of the gear ($p_a = p_c$). Since the circumference of the pitch circle of the gear can be expressed as πD_G or $p_c N_G$, this leads to

$$p_c = \frac{\pi D_G}{N_G} = p_a \quad (6)$$

Substitution into Eq. 5 gives

$$n_G = \frac{L}{\pi D_G} n_w \text{ rpm} \quad (7)$$

as another expression of the speed relationship.

Development of one turn of the worm leads to a triangle, as shown in Fig. 11. This triangle shows that

$$\tan \lambda_w = \frac{L}{\pi D_w} \quad (8)$$

Center distance is 0.5 ($D_w + D_G$), which can also be expressed as

$$C = \frac{L}{2\pi} (m_G + \cot \lambda_w) \quad (9)$$

Thermal Ratings

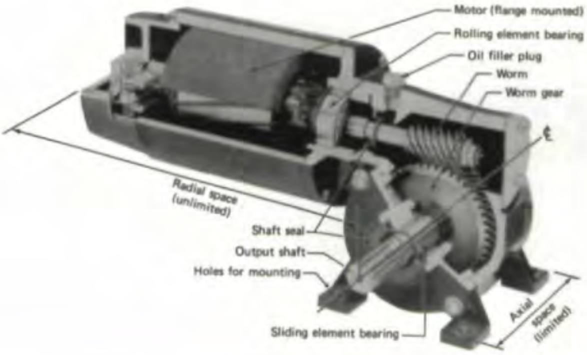
When worm gear teeth slide across the surfaces of mating worm threads, far more heat is generated than when the same load is carried by any other type of gearing. These thermal conditions raise the operating temperature of the oil, thereby reducing its load-carrying capacity. To keep oil temperatures below critical levels, manufacturers publish thermal ratings for each unit that indicate the maximum power input that produces a safe rise in oil temperature. Because common gear lubricants deteriorate rapidly at temperatures above 90°C (195°F), operating levels are usually kept at or below 75°C (170°F). Clearly, design of worm gearing must include temperature effects, as well as strength and wear, as a limiting factor. Of the three, overheating is the controlling parameter. For example, a worm gear may have a thermal rating of 5 kW and a mechanical rating of 7 kW for the higher speed ranges. This means that the gearing is capable of transmitting more than 5 kW as far as strength and wear are concerned – but not without overheating. Recently, however, the use of computers has improved worm geometry, thereby narrowing the gap between thermal and mechanical capacity.

Efficiency of Worm Gearing

While spur and helical gearing exhibit very high and virtually constant efficiencies (0.98 to 0.99), those of worm gearing may range from a low of 0.50 to a high of around 0.98. Generally, efficiency varies inversely with speed ratio provided the coefficient of friction does not change. The various parameters determining efficiency do not have a linear relationship (Fig. 12). Instead, efficiency e increases with decreasing coefficients of friction. Efficiency is greatly influenced by the lead angle for small values, but less and less as λ increases. A maximum is reached for $\lambda = 45$ deg.

For a well-designed, well-lubricated unit, the following is a fair approxima-

Fig. 13 (Right) – Typical worm gear reduction unit. (Courtesy Bodine Electric Company.)



tion to efficiency.

$$e = \frac{\tan \lambda}{\tan(\lambda + \theta)} \qquad (10)$$

where

$\theta = \arctan f =$ friction angle

Back-driving is the term used when the gear drives the worm. In this reverse action speed is increased at the expense of force. Back-driving can thus be used to advantage in speedup drives for centrifuges and turbochargers. The corresponding expression for efficiency is

$$e = \frac{\tan(\lambda - \theta)}{\tan \lambda} \qquad (11)$$

Theoretically, back-driving is possible only for $\lambda > \theta$, that is, when the lead angle is greater than the friction angle. In reality, this reverse action occurs for higher values of θ. Vibration, present in most mechanical equipment, effectively lowers friction, thereby reducing "self-locking" to "a mechanical fringe benefit." Only a brake can effectively prevent back-driving.

Optimum Design

Even though worm gear drives are among the oldest mechanisms, they are one of the least understood. Because of their inherent complexities, precise analytical methods have not evolved, so design relies heavily on trial-and-error testing. Worm gears have, however, reached a high degree of perfection. Thus, by analyzing the expression for efficiency, we may single out major design parameters and compare them with the industrial end product.

Low coefficients of friction are obtained by (1) using dissimilar metals for worm and gear, (2) providing smooth tooth surfaces, and (3) ensuring adequate lubrication. For instance, hardened and ground steel worms are used with gears of phosphor bronze or cast iron. Special lubricants are available for worm gearing.

Large lead angles are also desirable, but are obtained at the expense of lower speed ratios. Consider the following equation.

$$\tan\lambda = \frac{L}{\pi D_w} = \frac{p_a N_w}{\pi D_w} \qquad (12)$$

A large efficiency requires a large lead; therefore the worm should be multithreaded. Consequently, when worm gearing is designed primarily for transmitting power, it should be multithreaded, as is common practice.

To obtain a given ratio, some number of worm wheel teeth divided by some number of worm threads must equal the ratio. Thus, if the ratio is 6:

$$m_G = \frac{6}{1} = \frac{12}{2} = \frac{18}{3} = \frac{24}{4} = \frac{30}{5} = \frac{36}{6} = \frac{42}{7}$$

Any of these combinations may be used. The numerators represent the number of worm wheel teeth, and the denominators are the number of worm threads. As the total number of teeth increases, so does efficiency, but only at higher initial cost.

The expression for λ also indicates that a small worm diameter D_w is most desirable because it lowers rubbing velocity. As can be seen in Fig. 13, the worm diameter is small relative to the thread height.

Example 2

A right-angle speed reducer has a triple-threaded worm and a 41-tooth gear. Find the speed ratio, lead, lead angle, helix angle, pitch diameter of gear, center distance, efficiency, power output, and transmitted force for the following data.

$p_c = 32$ mm $\qquad n = 900$ rpm

$D_w = 44$mm

$f = 0.05 \qquad P = 0.75$ kW (input)

Will the unit back-drive?

Solution

$$m_G = \frac{N_G}{N_p} = \frac{41}{3} = \mathbf{13.67}$$

From Eq. 4:

$$L = N_w p_a = 3(32\text{ mm}) = \mathbf{96\ mm}$$

From Eq. 8:

$$\lambda_w = \arctan\frac{L}{\pi D_w}$$

$$= \arctan\frac{96\text{ mm}}{\pi(44\text{ mm})} = \mathbf{34.78\ deg}$$

For proper mesh, the lead angle of the worm must equal the helix angle of the gear. Thus $\psi_G = \lambda_w = \mathbf{34.78\ deg}$

As can be seen in Fig. 12

$$\psi_w = 90\text{ deg} - \lambda_w = \mathbf{55.22\ deg}$$

From Eq. 6:

$$D_G = \frac{p_a N_G}{\pi}$$

$$= \frac{(32\text{ mm})41}{\pi} = \mathbf{417.62\ mm}$$

From Eq. 9:

$$C = \frac{L}{2\pi}(m_G + \cot\lambda_w)$$

$$= \frac{96\text{ mm}}{2\pi}(13.67 + \cot 34.78\text{ deg})$$

$$= \mathbf{230.86\ mm}$$

From Eq. 10:

$$e = \frac{\tan\lambda_w}{\tan(\lambda_w + \theta)}$$

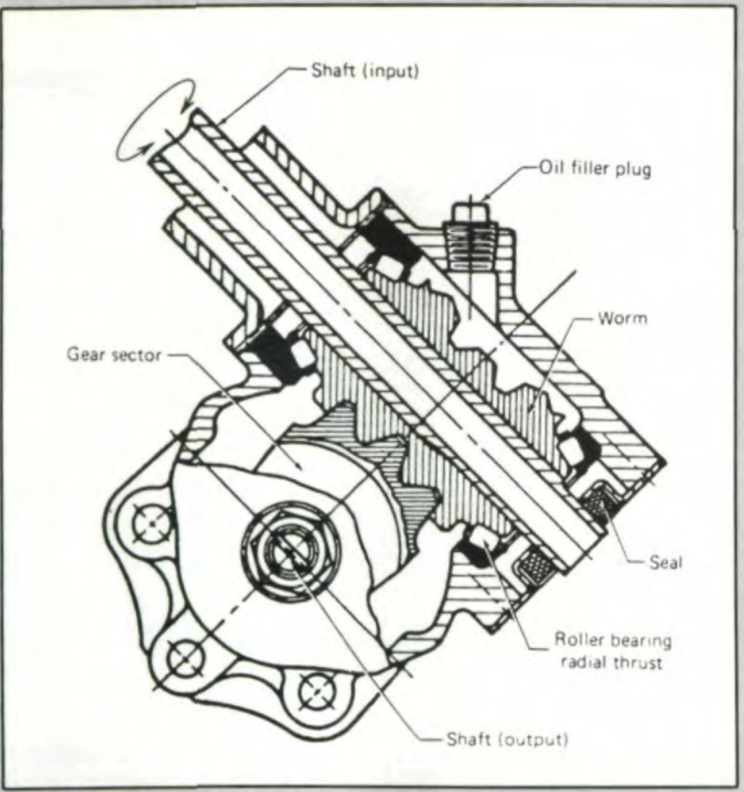


Fig. 14 – Steering mechanism. Note that only a gear sector is needed. (Courtesy The Timken Company.)

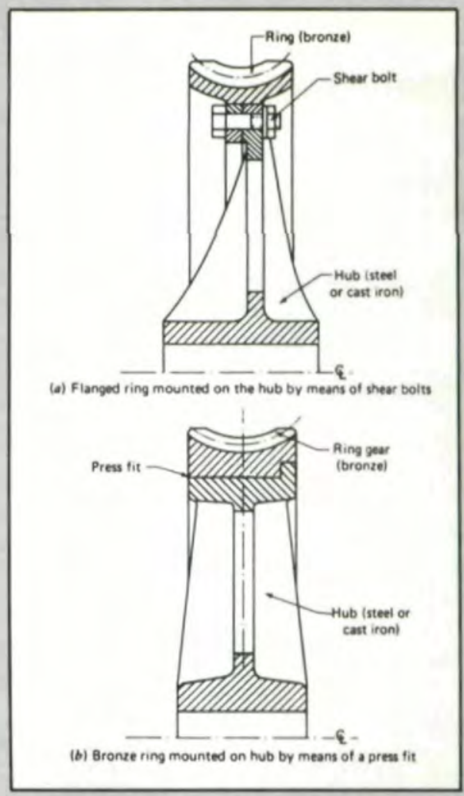


Fig. 16 – Design of a large worm gear.

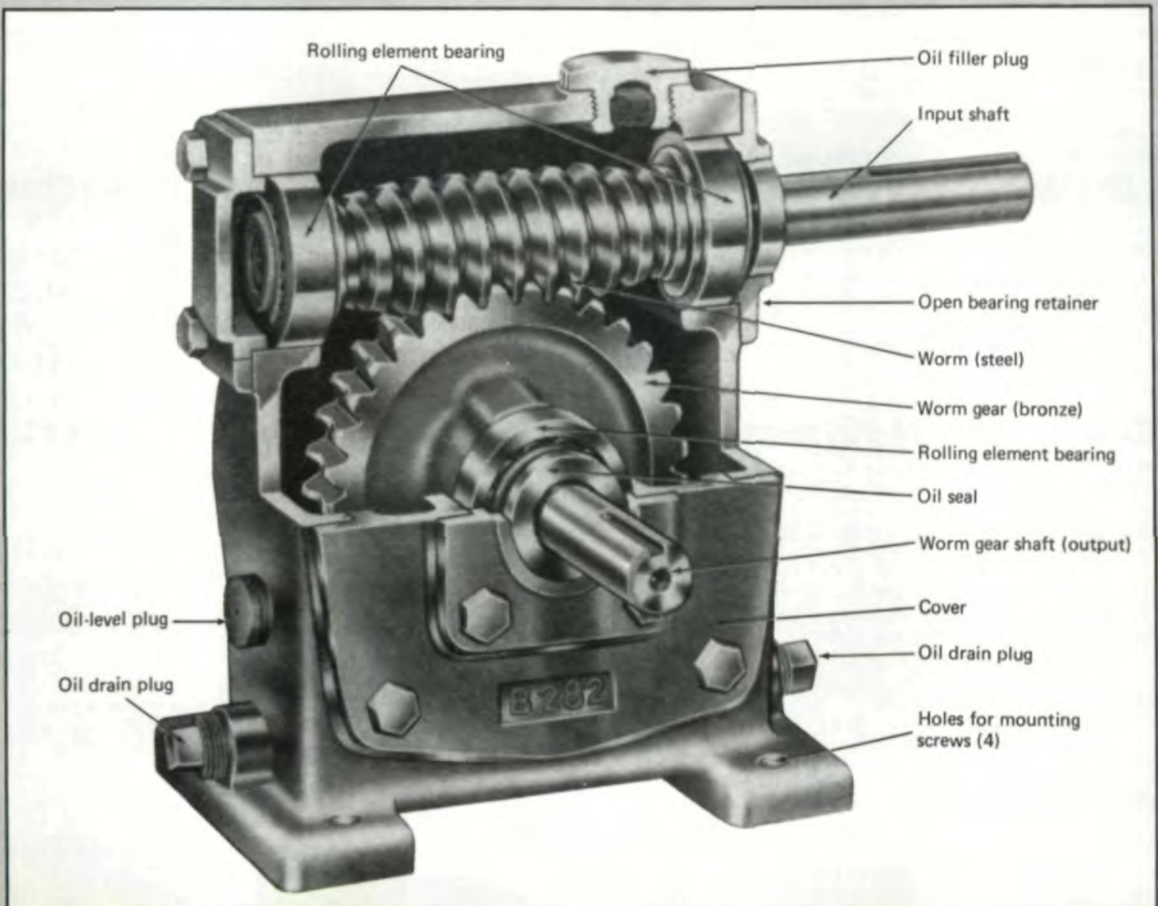


Fig. 15 – Typical medium-size worm gear speed reducer. (Courtesy of Morse Industrial Corp., subsidiary of Emerson Electric Co.)

(continued on page 63)

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(continued from page 60)

$$= \frac{\tan 34.78 \text{ deg}}{\tan(34.78 \text{ deg} + 2.86 \text{ deg})} = \mathbf{0.90}$$

Output: $P = 0.90(0.75 \text{ kW}) = \mathbf{0.675 \text{ kW}}$

The unit will back-drive because the lead angle of the worm (λ_w = 34.78 deg.) is much greater than the friction angle Θ where

$$\Theta = \arctan f$$

$$= \arctan 0.05 = 2.86 \text{ deg.} \ll 34.78 \text{ deg.}$$

From the power equation:

$$W_t = \frac{9550\,P}{(0.5\,D_w)n}$$

$$= \frac{9550(0.675 \text{ kW})}{0.5(0.044\text{m})(900 \text{ rpm})} = \mathbf{326 \text{ N}}$$

Applications

As indicated in Fig. 12, worm gear efficiency varies widely from 0.5 to 0.98. It also varies inversely with speed ratio or mechanical advantage. Thus, single-threaded worm gear drives yield large reduction ratios, but at the expense of efficiency. In contrast, a multithread speed reducer will have high efficiency, but a somewhat lower reduction ratio. This fact leads to three major areas of application for worm gearing.

1. *Intermittent, infrequent* operations where a small, low-cost motor moves a heavy load, as in small hoists. Efficiency, of minor importance, is thus

(continued on page 64)

IMPROVED GEAR LIFE . . .

(continued from page 18)

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PRACTICAL ANALYSIS OF HIGHLY LOADED . . .

(continued from page 26)

Conclusion

Based on the DIN/ISO formulae for scoring capacity, a simplified method adopting a modified scoring index has been developed. As can be seen from typical applications, this method works with sufficient accuracy.

The calculation of scoring capacity will become more and more important in parallel with an increasing demand in transmitted power per gear volume. The practical experience with highly-loaded gears with regard to scoring will give more safety in the application of this calculation method and will possibly permit a reduction of the safety margins used today.

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GEARS FOR NONPARALLEL SHAFTS . . .

(continued from page 62)

sacrificed to obtain a large mechanical advantage. Typical applications are standby pumps, large valves, and gates.

2. *Intermittent, manual* operations requiring a *large mechanical advantage, such as in steering mechanisms and* opening and closing of valves and gates by means of handwheels (Fig. 14).
3. *Motorized, nearly continuous* operations where worm gearing competes with gear reduction units. When space is at a premium, as in machine tools, packaged, motor-driven worm reduction units are used in preference to gear reducers (Fig. 13). Depending on size and application, the unit may be self-contained or built integrally with an electric motor. Because of silent operation, such units are preferred in machine tools and also in elevators. These units all require multithreaded worms and ratios not exceeding 1:18. Larger ratios are achieved by connecting two units in series.

Design Detail of Worm Gearing

The unit shown in Fig. 15 is a typical, medium-size worm gear speed reducer. Smaller units of this type usually have housings of cast aluminum alloys for maximum thermal rating. For larger units the preferred material is cast iron. The worm is case-hardened and ground alloy steel of integral shaft design. The gear is cast bronze with generated teeth and keyed to the output shaft. Larger worm gears are often composed of a ring of bronze mounted on a center or hub of less expensive material. A common design utilizes a flanged rim mounted on the hub by means of shear bolts (Fig. 16*a*). Equally common is mounting by means of a press fit (Fig. 16*b*) assisted by a pin connection. The output shaft is high-quality, medium-carbon steel, ground to close tolerances. The worms and output shafts are frequently mounted on roller bearings. All shaft extensions are equipped with lip style, synthetic oil seals.

Lubrication

Generally, oil is contained within the housing and directed by splash to the bearings and to the zone of tooth and thread contact. Natural splash may be augmented by flingers, scrapers, and cups attached to the gear. Channels or ribs may be furnished inside the housing to help direct oil to the bearings.

Summary

Despite higher initial cost, gears for nonparallel shafts are justified because they often save space and lead to a better design. Kinematically, these gears all perform the very difficult task of changing the plane of rotation. With the exception of crossed helical gears, all have reached a high degree of perfection and a long, useful life of transmitting power. Hypoid gears for automotive differentials, for instance, rarely fail during the life of a car. The versatility of worm gearing is due to the inverse relationship of efficiency to torque and reduction ratio. Table 1 summarizes comparative characteristics of speed reducer gear families.

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