6.2.4
GEARS

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USE OF GEARS WITH PUMP DRIVES

The main use of gearing in pump drives is to reduce the speed of the prime mover (a motor or an engine) to the level applicable to the pump. In some cases, however, the gears are employed to step up the speed because the pump must operate at a speed higher than that of the prime mover.

There are other jobs that gear drives must perform with pump units. For instance, a vertical pump may be combined with a prime mover that must operate horizontally (as in the case of a diesel engine). A right-angle gear set (Figures 1 and 4) can be incorporated into the drive of such a combination to transmit the power "around the corner" even if the gears are of a 1:1 ratio and no speed change is involved. See a word of caution concerning even ratio gears, "Minimizing Gear Noise," item 9.

At times, too, a gear drive must combine the power shafts of two prime movers; for example, an electric motor and a diesel engine. This combination is desirable where there is a need for emergency power in cases of electric power failure. The motor is used ordinarily and the diesel is reserved for emergencies. In such an arrangement, the motor may be mounted vertically on top of the gear drive to drive right through the shaft, whereas the diesel is geared at right angles to the motor.

Gear drives are also used to vary the speed of a pump. Change gear arrangements (Figure 2) may be used to vary the gear ratio. Also, numerous types of variable-speed devices, such as variable-speed pulley belts, friction rollers, hydraulic couplings, hydrostatic and hydroviscous drives, eddy-current and electric drives, are often utilized.

TYPES OF GEARS

The gears generally used for pump applications are parallel-shaft helical or herringbone gears and right-angle spiral-bevel gears. Spur gears are used on occasion, particularly in

FIGURE 1 Spiral-bevel gear, right-angle vertical pump drive

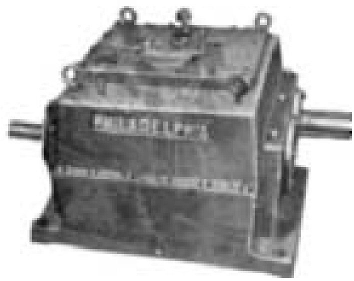

FIGURE 2 Four-speed change gear, variable-speed drive

low-power, low-speed pump drives. Straight bevel and hypoid gears are also occasionally used in right-angle drives. As in the case of spur gears, straight-bevel gears are limited in power and speed. Hypoid gears are employed only infrequently for pumps because they are generally more costly than the other right-angle types. Worm gears, too, are employed only on occasion, in cases where an overall package requires a compact gear arrangement or when a high ratio of speeds is called for. Worm gears are limited in power capacity, and the efficiency of this type of drive is lower than that of other types. Figure 3 shows the various types of gears.

Parallel-Shaft Gearing A high-speed parallel-shaft gear drive is shown in Figure 5.

HELICAL VERSUS SPUR GEARS Spur gears transmit power between parallel shafts without end thrust. They are simple and economical to manufacture and do not require thrust bearings, but they are generally used only on moderate-speed drives.

One of the first decisions that must be made when considering a parallel-shaft gear reducer or power transmission drive is whether the gears should be spur or helical and, if helical, whether they should be single or double. It is generally acknowledged that helical gears offer better performance characteristics than do spur gears, but because helical gears, size for size, often are somewhat more expensive, some users have shied away from them to keep the cost of the gear drive to a minimum. However, cost studies that compared similar size spur and helical gears found that helical gears are actually a better buy.

The geometries of spur and helical gears are both the involute tooth form. Slice a helical gear at right angles to its shaft axis and you have the typical spur gear profile. Typical of a spur gear, however, is that, when driving another spur, its teeth make contact with the teeth of the mating gear along the full length of the face. The load is transferred in sequence from tooth to tooth.

A more gradual contact between mating gears is obtained by slanting the teeth in a way to form helices that make a constant angle—a helix angle with the shaft axis. Tooth contact between the teeth of mating helical gears is gradual, starting at one end and moving along the teeth so, at any instant, the line of contact runs diagonally across the teeth.

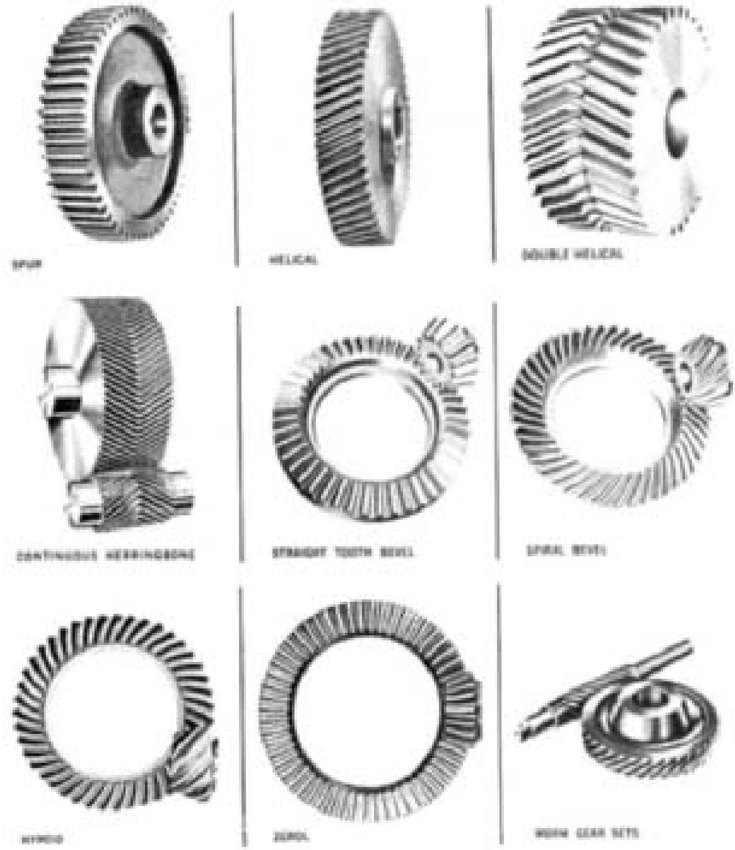

FIGURE 3 Types of gears

The effect of the tooth helix is to give multiple tooth contact at any time. Gear geometry can be arranged to give from two to six or more teeth at any time.

Because of the greater number of teeth in contact, a helical gear has a greater effective face width (up to 75% more) than an equivalent spur gear. Also, the effect of the tooth helix on the profile geometry in the plane of rotation is to make the pinion equivalent to a pinion with a greater number of teeth, thereby increasing its power capacity. A helical gear is capable of transmitting up to 100% more power than an equivalent spur gear. Furthermore, a helical gear set will give smoother, quieter operation.

The recommended upper limit of pitch line velocity for commercial spur gears is around 1000 ft/min (300 m/min). The upper limit for equivalent helical gears is about five times that, or 5000 ft/min (1500 m/min). Of course, as precision goes up, so do the permissible operating speeds for both spur and helical gears. Velocities in the 30,000-ft/min (9100-m/min) range are not uncommon for helical gears.

In addition to the normal radial loads produced by spur gears, helical gearing also produces an end thrust along the axis of rotation. The end thrust is a function of the helix angle: the larger the helix angle, the greater the thrust produced. Mounting assemblies and bearings for helical gearing must be designed to receive this thrust load.

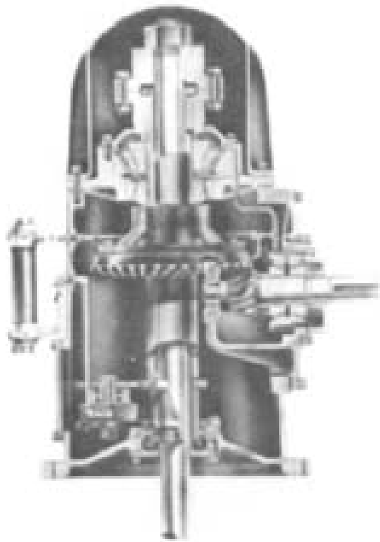

FIGURE 4 Cross section of spiral-bevel vertical pump drive

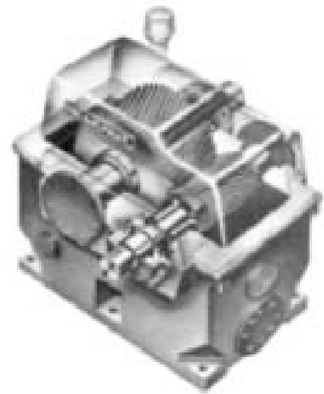

FIGURE 5 High-speed parallel-shaft gear drive

DOUBLE HELICAL GEARS Gears of this type have two sets of opposed helical teeth. Each set of teeth has the same helix angle and pitch, but the helices have opposing hands of cut. Thus, the thrust loads in two sets of teeth counterbalance each other and no thrust is transmitted to shaft and bearings. Also, because end thrust is eliminated, it is possible to cut the teeth with greater helix angles than is generally used in helical gears. Tooth overlap is greater, producing a stronger and smoother tooth action.

The advantages attributed to helical gears are also applicable to double helical gears. Double helical gearing finds application in high-speed pump applications where a large helix angle must be combined with tooth sharing and elimination of end thrust for extremely smooth gear action.

Single helical gears, however, have some attractive advantages over double helical gears, the most significant being that, in the former, the external thrust loads do not affect gear tooth action. With a double helical gearing, a thrust load on the member with a thrust bearing tends to unload one of the helices and overload the opposite one. See transmission of external thrust forces in Section 6.3.1, "Pump Couplings and Intermediate Shafting."

Furthermore, the gear face for a single helical gear can be made narrower than for a double helical gear because the need for a groove between the two helices is eliminated. This leads to the use of a narrower, stiffer pinion with less tooth deflection and torsional windup and, generally, to a more favorable critical speed condition.

An axial vibration of the pinion, without a thrust bearing on a double helical gear set, is sometimes referred to as apex runout. This vibration can be caused by pitch circle runout where one helix is out of phase with the other. This tends to unload one helix cyclically and induce the vibration. The vibration will generally be the pinion because it has no thrust bearing, but it will be at the frequency of the member with the thrust runout problem.

When pitch circle runout, tooth spacing errors or lead errors are present in an element of a single helical gear set, the vibration will, when loaded, be radial because each member has a thrust bearing to restrain its axial movement.

CONTINUOUS-TOOTH HERRINGBONE GEARS Gears of this type are double helical gears cut without a groove separating the two rows of teeth. Because of the arched construction of these gears, they are often known as “the gears with a backbone.” Continuous-tooth herringbone gears are used for the transmission of heavy loads at moderate speeds where continuous service is required, where shock and vibration are present, or where a high reduction ratio is necessary in a single train. Because of the absence of a groove between opposing teeth, a herringbone gear has greater active face width than the hobbed double helical gear and therefore is stronger. There is also no end thrust, as the opposing helices counterbalance one another. The bearing arrangement of herringbone gears is usually the same as double helix in that the pinion does not usually have a thrust bearing. See transmission of external forces under Section 6.3.1.

Much of the success of the continuous-tooth herringbone gear is due to the greater number of teeth in contact and to the continuity of tooth action, which is an outgrowth of the larger helix angle. These larger helix angles can be fully utilized without creating bearing thrust loads. Continuous-tooth herringbone gears normally are furnished with a helix angle of 30°. Herringbone gears are generally of a lower American Gear Manufacturers Association (AGMA) quality level than hobbed or ground gears.

Crossed-Axis Gearing

STRAIGHT-BEVEL GEARS Gears of this type transmit power between two shafts usually at right angles to each other. However, shafts other than 90° can be used. The speed ratio between shafts can be decreased or increased by varying the number of teeth on pinion and gear. These gears are designed to operate at speeds up to 1000 ft/min (300 m/min) and are more economical than spiral-bevel gears for right-angle power transmission where operating conditions do not warrant the superior characteristics of spiral-bevel gearing. When shafts are at right angles and both shafts turn at the same speed, the two bevel gears can be alike and are called *miter gears.*

SPIRAL-BEVEL GEARS Spiral-bevel gear teeth and straight-bevel gear teeth are both cut on cones. They differ in that the cutters for straight-bevel teeth travel in a straight line, resulting in straight teeth, whereas the cutters for spiral-bevel gear teeth travel in the arc of a circle, resulting in teeth that are curved and are called spiral. Figure 4 shows a cross section of a spiral-bevel vertical pump drive.

Spiral-bevel gearing is superior to straight-bevel in that, in the former, loading is always distributed over two or more teeth in any given instant. Recommended maximum pitch line velocity for spiral bevels is about 8000 ft/min (2400 m/min). Spiral bevels are also smoother and quieter in action because the teeth mesh gradually. Because of the curved teeth, spiral-bevel pinions may be designed with fewer teeth than straight-bevel pinions of comparable size. Thrust loads are greater for spiral-bevel gearing than for straight-tooth bevels, however, and vary in axial direction with the direction of rotation and hand of cut of the pinion and gear. Where possible, the hand of spiral should be selected such that the pinion tends to move out of mesh. To assure that the pinion thrust is away from the cone center, out of mesh, the following applies:

Driving Member	Hand of Spiral	Rotation Direction
Pinion	Left Hand	CW
Pinion	Right Hand	CCW
Gear	Right Hand	CCW
Gear	Left Hand	CW

Reversing the direction of rotation should be avoided.

ZEROL GEARS Zerol-bevel gears are cut on conical gear blanks and have curved teeth similar to the spiral bevels, but the teeth are cut with a circular cutter that does not pass through the cone apex. Thus, these are spiral-bevel gears with zero spiral angle (hence the name). Furthermore, the tooth bearing is localized as in spiral-bevel gearing; thus stress concentration at the tips of the gear teeth is eliminated. Zerol-bevel gears are replacing straight-bevel gearing in many installations because their operation is

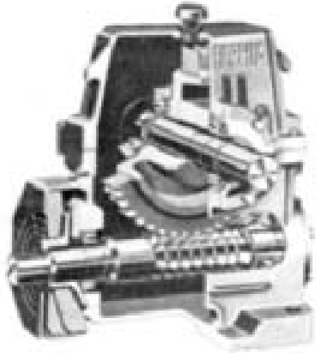

FIGURE 6 Fan-cooled worm gear drive

smoother and quieter as a result of their curvature and their operating life is longer. Like straight-bevel gears, zerol gears have the advantage of no inward axial thrust under any conditions. The zero spiral angle produces thrust loads equivalent to those in straight-bevel gears.

HYPOID GEARS Hypoid-bevel gears have the general appearance of spiral-bevel gears but differ in that the shafts supporting the gears are not intersecting. The pinion shaft is offset to pass the gear shaft. The pinion and gear are cut on a hyperboloid of revolution, the name being shortened to *hypoid.* Hypoid gears can be made to provide higher ratios than spiral-bevel gears. They are also stronger and operate even more smoothly and quietly. The fact that two supporting shafts can pass each other, with bearings mounted on opposite sides of the gear, provides the ultimate rigidity in mounting.

WORM GEARS In operation, the teeth on the worm of a worm gear set (Figure 6) slide against the gear teeth and at the same time produce a rolling action similar to that of a rack against a spur constant output speed completely free of pulsations. Worm gearing is particularly adaptable to service where heavy shock loading is encountered.

Worm gearing is extremely compact, considering load-carrying capacity. Much higher reduction ratios can be attained through a worm gear set on a given center distance than through any other type of gearing. Thus, the number of moving parts in a speed-reduction set is reduced to the absolute minimum. However, worm gearing is limited in power capacity and has lower efficiencies than parallel-shaft and bevel-gear types. Extremely high worm thrust loads are generated by work gearing. Therefore, never reverse the rotation unless the unit is specifically designed for operation in both directions.

GEAR MATERIALS AND HEAT TREATMENT

Two of the most important factors in dictating the success or failure of a gear set are the choice of material and heat treatment. This is especially true for gears designed for higher power. American Gear Manufacturers Association (AGMA) ratings for gear strength and durability are dependent on the choice of material and heat treatment. It is often possible to reduce the size of the gear box substantially by simply changing from low-hardened or medium-hardened gears (about 300 Brinell) to full-hardened gears (about 55 to 60 Rockwell C). Generally used methods for hardening gear sets are

1. Through hardening
2. Nitriding

3. Induction hardening
4. Carburizing and hardening
5. Flame hardening

The use of high-hardness, heat-treated steels permits smaller gears for given loads. Also, hardening can increase service life up to 10 times without increasing size or weight. After hardening, however, the gear must have at least the accuracy associated with softer gears and, for maximum service life, even greater precision. Furthermore, carburized and ground gears must be aligned within their housings to a higher degree of accuracy than through-hardened gears. This is because their ability to "comply with" misalignment is less, although they are capable of transmitting much higher loads and longer life when properly aligned.

Through Hardening Suitable steels for medium to deep hardening are 4140 and 4340. These steels, as well as other alloy steels with proper hardenability characteristics and carbon content of 0.35 to 0.50, are suitable for gears requiring maximum wear resistance and high load-carrying capacity. Relatively shallow-hardening carbon steel gear materials, types 1040, 1050, 1137, and 1340, cannot be deep hardened and are suitable for gears requiring only a moderate degree of strength and impact resistance. A 4140 steel will produce a hardness of 300 to 350 Brinell. For heavy sections and applications requiring greater hardness, a 4340 steel will provide 350 to 400 Brinell. Cutting of gears in the 380 to 400 Brinell range, although practical, is generally difficult and slow.

Nitriding Nitriding is especially valuable when distortion must be held to a minimum. It is done at a low temperature (975 to 1050°F, 524 to 566°C) and without quenching—eliminating the causes of distortion common to other methods of hardening and often necessary for finish machining after hardening. Nitrided case depths are relatively shallow so nitriding is generally restricted to finer pitch gears (four-diameter pitch or finer). However, double-nitriding procedures have been developed for nitriding gears with as coarse as two diametral pitch.

Any of the steel alloys that contain nitride-forming elements, such as chromium, vanadium, or molybdenum, can be nitrided. Steels commonly nitrided are 4140, 4340, 6140, and 8740. It is possible with these steels to obtain core hardnesses of 300 to 340 Brinell and case hardnesses of 47 to 52 Rockwell C. Where harder cases are required, one of the Nitralloy steels may be used. These steels develop a case hardness of 65 to 70 Rockwell C with a core hardness of 300 to 340 Brinell. The depth of case in a 4140 or 4340 steel varies as the length of time in the nitriding furnace. A single nitride cycle will produce a case depth of 0.025 to 0.030 in (0.64 to 0.76 mm) in 72 h. Doubling the time will produce a case depth of 0.045 to 0.050 in (1.14 to 1.27 mm). For the majority of applications, the case depth obtained from a single cycle is ample.

Case depth for Nitralloy steels is somewhat less than the depths obtainable for other alloy steels. In general, alloy steels 4140 and 4340 give up to 50% deeper case than Nitralloy steels for the same furnace time. These cases are tougher but less hard.

Induction Hardening Two basic types of induction hardening are used by gear manufacturers: coil and tooth-to-tooth. The coil method consists of rotating the work piece inside a coil producing high-frequency electric current. The current causes the work piece to be heated. It is then immediately quenched in oil or water to produce the desired surface hardness. Hardnesses produced by this method range from 50 to 58 Rockwell C, depending on the material. The coil method hardens the entire tooth area to below the root.

Tooth-to-tooth full-contour induction hardening is an economical and effective method for surface hardening larger spur, helical, and herringbone gearing. In this process, an inductor passes along the contour of the tooth, producing a continuous hardened area from one tooth flank around the root and up the adjacent flank. The extremely high localized heat allows small sections to come to hardening temperature while the balance of the gear dissipates heat. Thus major distortions are eliminated. The 4140 and 4340 alloy steels are widely used for tooth-to-tooth induction hardening. The hardness of the case produced by this method ranges from 50 to 58 Rockwell C, and the flanks may be hardened to a depth of 0.160 in (4.06 mm). These steels are air-quenched in the hardening process. Plain carbon steels, such as 1040 and 1045, may be used for induction hardening, but they must be water-quenched.

Carburizing Carburizing with subsequent surface hardening offers the best way to obtain the very high hardness needed for optimum gear life. It also produces the strongest gear, one that has excellent bending strength and high resistance to wear, pitting, and fatigue. The residual compressive stresses inherent in the carburized case substantially improve the fatigue characteristics of this heat-treated material. Normal case depths range from approximately 0.030 to 0.250 in (0.76 to 6.3 mm). Case hardnesses range from 55 to 62 Rockwell C, and core hardness from 250 to 320 Brinell. Recommended carburizing-grade steels are 4620, 4320, 3310, and 9310. The main limitation to carburizing and hardening is that the process tends to distort the gear. Techniques have been developed to minimize this distortion, but generally after carburizing and hardening, it is necessary to grind or lap the gear to maintain the required tooth tolerances.

Flame Hardening In tooth-to-tooth progressive flame hardening, an oxyacetylene flame is applied to the flanks of the gear teeth. After the surface has been heated to the proper temperature, it is air- or water-quenched. This method has some limitations; because the case does not extend into the root of the tooth, the durability is improved but the overall strength of the gear is not necessarily. In fact, stresses built up at the junction of a hardened soft material may actually weaken the tooth.

Many times it is desirable to use different heat treatments for the pinion and gears. Heat-treatment combinations used for pinions and gears are here listed in order of preference for optimum gear design.

1. Carburized pinion, carburized gear
2. Carburized pinion, through-hardened gear
3. Carburized pinion/nitrided gear
4. Nitrided pinion/nitrided gear
5. Nitrided pinion/through-hardened gear
6. Induction-hardened pinion/through-hardened gear
7. Carburized pinion/induction-hardened gear
8. Induction-hardened pinion/induction-hardened gear
9. Through-hardened pinion/through-hardened gear

OPTIMIZING THE GEARING

The most important factor influencing the durability of a gear set, and hence the gear size, is the hardness of the gear teeth. It is often possible to reduce considerably—sometimes by as much as half—the overall dimensions of a gear set by changing from medium-hardened gears (about 300 Brinell) to full-hardened gears (about 55 to 60 Rockwell C).

Other factors play a role in minimizing the dimensions of a gear set; for example, the ratio between face width and pitch diameter and the proper pressure angle and pitch of teeth. Thus, when it comes to deciding between a set of standard catalog gears and gears designed specifically to meet the requirements of the application, the question of cost versus optimizing comes to bear. In general, where there is only a limited number of units to be made, the catalog gears are much less expensive and also much more readily available. There are many applications that call for critical power, speed, or space requirements, however, and it may pay in these applications to select gears that are designed for that application.

Minimizing Gear Noise Specifying or designing a gear set to produce low noise and vibration levels frequently leads to choices that are the opposite of those for optimizing the gears for strength and size. Generally, a parallel-shaft gearing rather than right-angle gearing is preferred for quiet operation because of greater geometric control, inherent ability to maintain tight manufacturing tolerances, and minimum friction during tooth contact. Helical gears, in particular, can have more than one tooth in contact (helical overlap), and some experience has shown as much as a 12-dB reduction in noise using helical instead of spur gears. Double helical or herringbone gearing has the problem of manufacturing the two helices with precisely the same phase and accuracy. Helical gearing must have thrust bearings or collars on each element and so produce an overturning moment. The overturning moment is more pronounced with single stage high ratio units such as large diameter, narrow face-width gears, and small diameter pinions. These problems concerning double and single helical units are easily handled with proper design and manufacturing.

For quiet, smooth operation, the gears should be designed with some or all of the following properties:

1. Select the finest pitch allowable under load considerations.
2. Employ the lowest pressure angle: $14\frac{1}{2}$ and 20° are most commonly used.
3. Modify the involute profile to include tip and root relief with a crowned flank to ensure smooth sliding into and out of contact without knocking and to compensate for small misalignments.
4. Allow adequate backlash (clearance) for thermal and centrifugal expansion, but not so much as to prevent proper contact.
5. Specify the higher AGMA quality levels, which will reduce the total dynamic load. Generally, AGMA quality 12 or better is required for smooth, quiet operation.
6. Maintain surface finishes of at least 20 microinches Ra (surface roughness average value).
7. Maintain rotor alignments and runouts accurately.
8. Limit rotor unbalance per plane to less than

   Umax = 3 W/N in US units, and Umax = 4760 W/N in SI units, where

   Umax = residual unbalance, oz · in (g · mm)

   W = static weight on the journal, lb (kg)

   N = maximum continuous speed, rpm

9. Provide a nonintegral ratio ("hunting tooth") to prevent a tooth on the pinion from periodically contacting the same teeth on the mating gear.
10. Have resonances of rotating system members (critical speeds) at least 30% away from operating speed, multiples of rotating speeds, and tooth-mesh frequencies.
11. Have resonances of gear cases and other supporting members 20% away from operating speeds, multiples, and tooth-mesh frequencies.
12. Specify the highest-viscosity lubricant consistent with design and application.
13. Select rolling element bearings to minimize noise generation. Generally, hydrodynamic sleeve bearings are quieter than antifriction types but are more difficult to apply.
14. Because housing design is another area where noise and vibration reductions can be obtained, select an acoustically absorbent material for the housing or design the housing with built-in isolation mounts to cut down any vibration attenuation.
15. For parallel shaft gearing, it is recommended that speed increasers be up meshed and speed reducers be down meshed to prevent rotor instability.

PACKAGED GEAR DRIVES

In many cases, it is preferable to select a packaged gear drive rather than a set of open gears that must be mounted and housed.

The relative merits of a packaged drive, or "gear reducer," versus open gearing are many. The packaged drive consists essentially of gears, housing, bearings, shafts, oil seals, and a positive means of lubrication. Frequently, reducers also include any or all of the following: electric motor and accessories, bedplates or motor supports, outboard bearings, a mechanical or electric device providing overload protection, a means of preventing reverse rotation, and other special features as specified.

The advantages of packaged gear drives have been well established and should be given consideration when selecting the type of gear drive for the application:

1. *Power conservation.* Because of accurate gear design, quality construction, proper bearings, and adequate lubrication, minimum loss between applied and delivered power is assured.
2. *Low maintenance.* If the correct design and power capacity for the requirements are selected and the recommended operating instructions are followed, low maintenance costs will result.
3. *Operating safety.* All gears, bearings, and shafts are enclosed in oil-tight, strongly built cast iron or steel housings.
4. *Low noise and vibration level.* Precision gearing is carefully balanced and mounted on accurate bearings. The transmitting motion is uniform and shock-free. The entire mechanism is tightly sealed in sound-damping rigid housing. Noise and vibration are reduced to a minimum.
5. *Space conservation.* Units are entirely self-contained and extremely compact; therefore, they require a small space. This also enables them to be installed in out-of-the-way locations.
6. *Adverse operating conditions.* Enclosure designs have been developed to protect the mechanism from dirt, dust, soot, abrasive substances, moisture, or acid fumes.
7. *Economy.* Units permit the use of high-speed prime movers directly connected to low applied speeds.
8. *Life expectancy.* The life of a unit can be predetermined by design and made unlimited if it is correctly aligned and properly maintained.
9. *Power and ratios.* Units are available in almost all desired ratios and for all practical power requirements.
10. *Cooling systems.* Greater attention to sump capacity for oil and the use of fan air cooling have allowed higher powers to be transmitted through smaller units without overheating.
11. *Appearance.* Housings have been streamlined for eye appeal as well as for reduction of weight and space.
12. *Rugged capabilities.* The ruggedness of steel-constructed welded housings and modern housings produces higher reliability and service life.

Types of Gear Packages Gear packages are used in multiple combinations to produce the ratio desired in the unit. Units are available in single-, double-, and triple-reduction configurations. Three stages of reduction are generally the maximum number used in standard reducers, although it is possible to use four or even more stages. Units for increasing the output speed generally have only one gear set, although at times two-stage units have been used successfully as speed increasers. Gear packages may be assembled with shaft arrangements that are right-handed or left-handed (Figure 7).

ALLOWABLE SPEEDS OF GEAR REDUCERS The maximum speed of a gear reducer is limited by the accuracy of the machined gear teeth, the balance of the rotating parts, the allowable

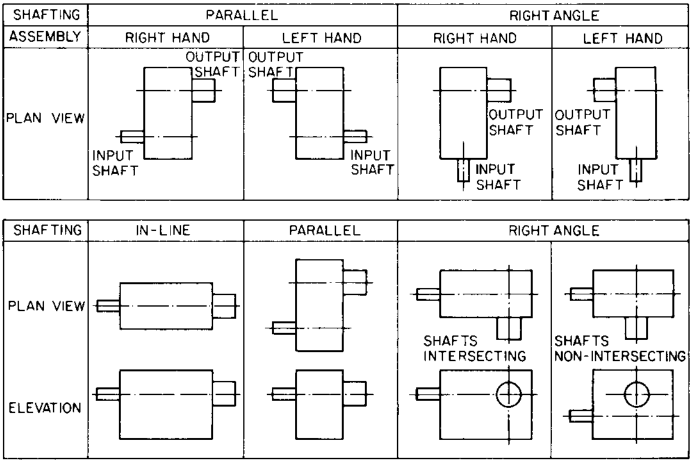

FIGURE 7 Shaft arrangements for gear drives

noise and vibration, the allowable maximum speed of the bearings, the pumping and churning of the lubricating oil, the friction of the oil seals, and the heat generated in the unit.

At high speeds, it is possible for inaccuracies in the gear teeth to produce failure even though no power is being transmitted. Gear reducers built in accordance with AGMA specifications are recommended to operate at speeds given in Table 1.

POWER RANGE OF GEAR REDUCERS The power-transmitting capacity of a reduction gear unit is a function of the output torque and the speed of the reducers. Some types of reducers, such as worm gear reducers, are more satisfactory for high torques and low speeds, whereas others, such as helical herringbone (or double helical), are suitable for high torques and also high speeds. Therefore the range of powers suitable for various units is considerable. A listing of this range obtainable in standard types of reducers is given in Table 1, with a brief explanation of why the range indicated is maintained. These powers are not fixed at the values given because they are continually changing. Although the values given are general, there are many special reducers available outside this range.

RATIOS AND EFFICIENCIES The ratio of a gear reducer is defined as the ratio of the input shaft speed to the output shaft speed. Different types of gearing allow different ratios per gear stage. Spur gears usually are used with a ratio range of 1:1 to 6:1; helical, double helical, and herringbone with ratios of 1:1 to 10:1; straight-bevel with ratios of 1:1 to 4:1; spiral-bevel (also zerols and hypoids) with ratios of 1:1 to 9:1; and worm gears with ratios of $3\frac{1}{4}$:1 to 90:1. Planetary gear arrangements allow ratios of 4:1 to 10:1 per gear stage.

Factors influencing the efficiency of a gear reducer are

1. Frictional loss in bearings
2. Losses due to pumping lubricating oil
3. Windage losses due to rotation of reducer parts
4. Frictional losses in gear tooth action

It is not uncommon in many types of reducers to have the combined losses due to items 1, 2, and 3 greater than the loss due to item 4. For this reason, in some cases the power lost in the reducer remains practically constant regardless of the power transmitted. Therefore, it must be realized that the efficiency specified for a reducer applies only when the

TABLE 1 Guide for gear selection and design (reflecting generally accepted design criteria)

	Spur gearing		Helical gearing	
	External	Internal	External	Internal
Shaft arrangement	Parallel axis	Parallel axis	Parallel axis	Parallel axis
Ratio range	1:1 to 10:1	$1\frac{1}{2}$:1 to 10:1	1:1 to 15:1	2:1 to 15:1 (generally feasible) Ratio depends on pinion gear tooth combination because of clearance requirements
Size availability (including maximum face widths)	Up to 150-in (381-cm) OD, 30-in (76.2-cm) face width. Larger segmental gears can be produced with special processing and tooling	Up to 100-in (254-cm) OD, 16-in (406-cm) face width	Up to 150-in (381-cm) OD, 30-in (76.2-cm) face width	Up to 100 in (254 cm), depending on blank configuration; 16-in (406-cm) maximum face width
Gear tolerances (quality requirements)	See footnote.	See footnote.	See footnote.	See footnote.
Finishing methods (singly or in combination)	Cast: rotary cut, shaped; hobbed: shaved, ground	Same as external spurs	Shaped, bobbed, shaved, ground	Shaped, bobbed, shaved, honed, lapped, ground
Power range, hp (kW)	Commercial: less than 1000 (750)		Commercial: generally up to 50,000 (37,000) However, power limited only by maximum size capacity of design	Same as external helical gearing
Speed range, pitch line velocity	Commercial: normal up to 1000 (300); special precision up to 20,000 (6100)	Commercial, standard manufacture: up to 1000 (300); precision manufacture: up to 20,000 (6100)	To 30,000 (9100)	

TABLE 1 Continued.

	Spur gearing		Helical gearing	
	External	Internal	External	Internal
Gear efficiency, %	Commercial: 95 to 98		97 to 99	
Quietness of operation	Commercial: quiet under 500 ft/min (150 m/min); noise increases with increasing pitch-line velocity		Noise level depends on quality of gear. Higher pitch line velocity (above 5000 ft/min (1500 m/min)) requires higher precision gear. Gears operating 30,000 ft/min (9100 m/min) have been made with overall noise level below 90 dB. Quieter than spur gears.	
Load imposed on bearings	Radial only		Radial and thrust	Radial and thrust

	External double helical gearing	Continuous-tooth herringbone gearing
Shaft arrangement	Parallel axis	Parallel axis
Ratio range	1:1 to 15:1	1:1 to 10:1
Size availability (including maximum face widths)	Up to 150-in (381-cm) OD, 30-in (76.2-cm) face width	Up to 150-in (381-cm) OD, 30-in (76.2-cm) face width
Gear tolerances (quality requirements)	See footnote.	See footnote.
Finishing methods (singly or in combination)	Same as helical gearing	Shaped, shaved

TABLE 1 Continued.

	External double helical gearing	Continuous-tooth herringbone gearing
Power range, hp (kW)	Same as helical gearing	Up to 2000 (1500)
Speed range pitch line velocity, ft/min (m/min)	Same as helical gearing	Commercial: up to 5000 (1500)
Gear efficiency, %	Same as helical gearing	96 to 98
Quietness of operation	Same as helical gearing	Quiet operation up to 5000 ft/min (1500 m/min). Not generally used at extremely high pitch line velocity, over 20,000 ft/min (6100 m/min).
Load imposed on bearings	Radial only	Radial only

	Straight-bevel gearing	Spiral-bevel gearing	Zerol-bevel gearing	Hypoid gearing
Shaft arrangement	Intersecting axis	Intersecting axis	Intersecting axis	Nonintersecting, nonparallel axis
Ratio range	1:1 to 6:1	1:1 to 10:1	1:1 to 10:1	1:1 to 10:1
Size availability (including maximum face widths)	Up to 102-in (55-cm) OD, 12-in (30.5-cm) face width	Up to 102-in (55-cm) OD, 12-in (30.5-cm) face width	102-in (55-cm) OD, 12-in (30.5-cm) face	102-in (55-cm) OD, 12-in (30.5-cm) face
Gear tolerances (quality requirements)	See footnote.	See footnote.	See footnote.	
Finishing methods (singly or in combination)	Cast, generated, planed	Generated, planed, ground	Generated, planed, ground	Generated, planed, ground
Power range, hp (kW)	Up to 1,500 (1120)	Up to 20,000 (15,000), depending on speed	Same as straight bevel gears	Same as spiral bevel gears, with use of EP lubricants

TABLE 1 Continued.

	Straight-bevel gearing	Spiral-bevel gearing	Zerol-bevel gearing	Hypoid gearing
Speed range pitch line velocity, ft/min (m/min)	Same as spur gearing	Commercial, normal: up to 5,000 (1500); special precision; up to 15,000 (4500)	Up to 15,000 (4500) for ground gears	6,000 (1800) to 10,000 (3000), depending on offset
Gear efficiency, %	Same as spur gearing	96 to 98 (commercial)	94 to 98	85 to 98, depending on offset
Quietness of operation	Same as spur gearing	Noise level depends on quality of gear. Higher pitch line velocity—above 5,000 ft/min (1500 m/min) —requires higher precision gear.	Quieter than straight-bevel gears	Quiet
Load imposed on bearings	Radial and thrust	Radial and thrust	Radial and thrust	Radial and thrust

	Worm gearing	
	Cylindrical	Double-enveloping cone-drive gears
Shaft arrangement	Nonintersecting, nonparallel axis	Right angle in single-reduction units
Ratio range	$3\frac{1}{2}$:1 to 100:1	5:1 to 70:1
Size availability (including maximum face widths)	300-in (762-cm) OD, 4-in (10.2-cm) circular pitch	2- to 24-in (5.1- to 61-cm) center distance. For special requirements, larger and smaller sizes are available.
Gear tolerances (quality requirements)	Worm gear tolerances given in AGMA Standard: Inspection of Course-Pitch Cylindrical Works and Worm Gears, Standard No. 234.01	Standard AGMA commercial tolerances. Closer tolerances available for special applications

TABLE 1 Continued.

	Worm gearing	
	Cylindrical	Double-enveloping cone-drive gears
Finishing methods (singly or in combination)	Worm gears: hobbed, worm-milled, and ground	Worms: threads generated with cutter and finished by polishing. Gears: hobbed. Both members then lapped and matched together
Power range, hp (kW)	Up to 400 (300)	Fractional to 1,430 (1070) dependent on ratio, center distance, and speed
Speed range pitch line velocity, ft/min (m/min)	Up to 6000 (1830)	0 to 2400 rpm, or 2000 (610) rubbing speed with splash lubrication. Higher speeds permissible with special combinations.
Gear efficiency, %	From 25 to 95, depending on ratio	52 to 94, depending on ratio and speed
Quietness of operation	Relatively quiet operation up to 6000 ft/min (1830 m/min)	Smooth and quiet up to 2000 ft/min (610 m/min); can run quietly at higher speeds with special attention to lubrication, mounting, materials, balancing, and so on.
Load imposed on bearings	Radial and thrust	Radial and thrust

Gear tolerances are dependent on method of manufacture, application, load requirements, and speeds. For spur, helical, and herringbone gears, ANSI/AGMA Gear Classification Manual 2000 A-88 lists quality numbers from 3 to 15 for coarse-pitch gears and 5 to 16 for fine-pitch gears, quality increasing as quality number increases. The general range of quality for gears now being manufactured is from quality numbers 5 to 14. Quality numbers relate to runout, tooth-to-tooth, spacing, profile, total composite, and lead tolerances. Bevel and hypoid gear tolerances range from 3 to 13.

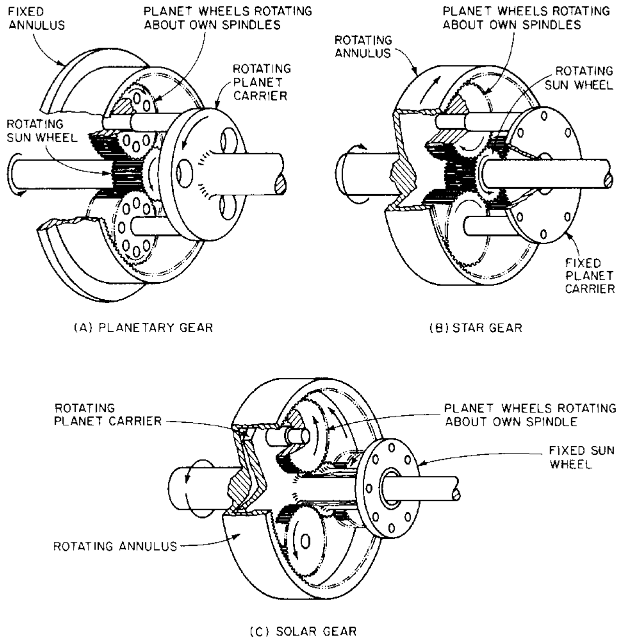

FIGURE 8 Single epicyclic gear drives

unit is transmitting its rated power because when no power is being transmitted through the reducer, all the input power (small as it may be) is used in friction and the efficiency is zero.

Ratios available in standard reducers and efficiencies to be expected when units are transmitting rated power are given in Table 1.

Epicyclic Gear Units Epicyclic gear units are sometimes used for pump drive applications. The important advantages are compact configuration, coaxial shafts, and light weight.

The most common types of epicyclic arrangements are planetary, star, and solar (Figure 8). The planetary configuration (with the planet carriers integral with the output shaft) is the most commonly used arrangement. It is simple and rugged and gives the maximum ratio for the size of gears. The star arrangement (where the planet carrier is fixed and the internal gear rotates integrally with the output shaft) is used for higher-speed applications because centrifugal loads of the planet gears are eliminated with nonrotating planet carrier. The solar arrangement (where the pinion is fixed and the planet carrier is integral with the output shaft) has the input through the internal gear. This arrangement gives epicyclic advantages but allows a low ratio, generally less than 2:1. Higher ratios (in the range of 8:1 to 60:1) can be obtained utilizing double-reduction or compound-planetary arrangements.

INSTALLATION

The basic gear unit is generally shipped from the factory completely assembled. Mating gears and pinions are carefully assembled at the factory to provide proper tooth contact. Nothing should be done to disturb this setting.

Solid Foundation The reducer foundation should be rigid enough to maintain correct alignment with connected machinery. The foundation should have a flat mounting surface in order to assure uniform support for the unit. If the unit is mounted on a surface other than horizontal, consult the factory to ensure that the design provides for proper tooth contact and adequate lubrication.

The design of fabricated pedestals or baseplates for mounting speed reducers should be carefully analyzed to determine that they are sufficiently rigid to withstand operating vibrations. Vibration dampening materials may be used under the baseplate to minimize the effect of vibrations.

When mounting a drive on structural steel, the use of a rigid baseplate is strongly recommended. Bolt unit and baseplate securely to steel supports with proper shimming to ensure a level surface.

If a drive is mounted on a concrete foundation, allow the concrete to set firmly before bolting down the unit. For the best mounting, grout structural steel mounting pads into the concrete base rather than grouting the gear unit directly into the concrete.

The gear unit inspection covers should be removed and a tooth contact check be performed using a color transfer material to ensure that the contact in the field is the same as when checked in the manufacturer's shop during assembly. This should be done regardless of whether the gear unit was mounted in the field or received on a bedplate.

Leveling If shims are employed to level or align the unit, they should be distributed evenly around the base under all mounting pads to equalize the support load and to avoid distortion of the housing and highly localized stresses. All pads must be squarely supported to prevent distortion of the housing when the unit is bolted down.

Alignment If the equipment is received mounted on a bedplate, it has been aligned at the factory. However, it may have become misaligned in transit. During field mounting of the complete assembly, it is always necessary to check alignment by breaking the coupling connection and shimming the bedplate under the mounting pads until the equipment is properly aligned. All bolting to the bedplate and foundation must be pulled up tight. After satisfactory alignment is obtained, close up the coupling.

Couplings Drive shafts should be connected with flexible couplings. The couplings should be aligned as closely as possible following the manufacturer's instructions. However, many of the coupling manufacturers publish catalog values that are actually "jam angles." A coupling aligned to within four (4) minutes of a degree at operating temperature will operate satisfactorily in nearly any combination of torque and speed within its design limits. Four (4) minutes of a degree is about 0.0006 in/in (0.0006 mm/mm) of engagement separation.

Thrust from the connected equipment can be transmitted across the coupling to the gear. Caution should be exercised to assure that the proper shaft to shaft end spacing is maintained to minimize external thrust force.

Alignment and Bolting The gear unit, together with the prime mover and the driven machine, should be correctly aligned. After precise alignment, each member must be securely bolted and doweled in place. Coupling alignment instructions should be carefully followed. Prior to initial operation, each member must be shallow doweled in place. After the system has been checked at operating temperature, the dowels should be sunk to their proper depth.

LUBRICATION

Types of Lubricant The recommended types of oil for use in gear units are either straight mineral oil or extreme-pressure (EP) oil. In general, the straight mineral oil should be a high grade, well-refined petroleum oil within the recommended viscosity range. It must be neutral in reaction and not corrosive to gears and ball or roller bearings. It should have good defoaming properties and good resistance to oxidation for high operating temperatures.

Gear drives that are subject to heavy shock, impact loading, or extremely heavy duty should use an EP lubricant. EP gear lubricants are petroleum-based lubricants containing special chemical additives. The ones most recommended contain sulfur-phosphorous additives. Sulfur-phosphorous EP oils may be used to a maximum sump temperature of 180°F (82°C).

In general, if units are subjected to unusually high ambient temperatures (100°F, 38°C or higher), extreme humidity, or atmospheric contaminants, use the straight mineral oil recommended.

Grease Lubrication The lubricant should be high-grade, nonseparating, ball bearing grease suitable for operating temperatures to 180°F (82°C). Grease should be NLGI No. 2 consistency.

The grease lubricant must be noncorrosive to ball or roller bearings and must be neutral in reaction. It should contain no grit, abrasive, or fillers; it should not precipitate sediment; it should not separate at temperatures up to 300°F (149°C); and it should have moisture-resistant characteristics and good resistance to oxidation.

Grease Lubrication of Bearings Pressure fittings are often supplied in gear units for the application of grease to bearings that are shielded from the oil. Although a film or grease over the rollers and races of the bearing is sufficient lubrication, drives are generally designed with ample reservoirs at each grease point.

Greased bearings should be lubricated at definite intervals. Usually one-month intervals are satisfactory unless experience indicates that regreasing should occur at shorter or longer intervals.

Oil Seals Oil seals require a small amount of lubricant to prevent frictional heat and subsequent destruction when the shaft is rotating. Normally when a single seal is utilized, sufficient lubricant is provided by spray or splash. Certain design or application requirements dictate that double seals be used at some sealing points. When this is the case, a grease fitting and relief plug are located in the seal retainer to provide lubricant to the outer seal. Grease must periodically be applied between the seals by pumping through the fitting until overflow is noted by the relief plug. The greases recommended for bearings may also be used for seals.

TROUBLESHOOTING TIPS

Improper lubrication causes a high percentage of gear reduction unit failures. Too frequently, speed reducers are started up without any lubricant at all. Conversely, units are sometimes filled to a higher oil level than specified in the mistaken belief that better lubrication is obtained. This higher oil level usually results in more of the input power going into churning the oil, creating excessive temperatures with detrimental results to the bearings and gearing. Insufficient lubrication gives the same results.

Gear failure due to overload is a broad and varied area of misapplication. The nature of the load (input torque, output torque, duration of operating cycle, shocks, speed, acceleration, and so on) determines the gear unit size and other design criteria. Frequently, a gear drive must be larger than the torque output capability of the severity of application conditions by providing a higher nominal power that in effect increases the size of the gear unit. If there is any question in the user's mind that the actual service conditions may be more severe than originally anticipated, it is recommended that this information be communicated to the gear manufacturer before start-up. Often there are remedies that can be suggested before a gear unit is damaged by overload, but none are effective after severe damage.

Motors and other prime movers should be analyzed while driving the gear unit under fully loaded conditions to determine that the prime mover is not overloaded and thus putting out more than the rated torque. If it is determined that overload does exist, the unit should be stopped and steps taken either to remove the overload or to contact the manufacturer to determine suitability of the gear drive under the observed conditions.

Table 2 is an extensive troubleshooting chart that should be consulted whenever necessary.

TABLE 2 Troubleshooting chart

Trouble	What to inspect	Action
Overheating	1. Unit overload	Reduce loading or replace with drive of sufficient capacity.
	2. Oil-cooler operation	Check coolant and oil flow. Vent system of air. Oil temperatures into unit should be approximately 110°F (43°C). Check cooler internally for build-up of deposits from coolant water.
	3. Oil level	Check oil level indicator to see that housing is accurately filled with lubricant to the specified level.
	4. Bearings adjustment	Bearings must not be pinched. Adjustable tapered bearings must be set at proper bearing lateral clearance. All shafts should spin freely when disconnected from load.
	5. Oil seals or stuffing box	Oil seals should be greased on those units having grease fitting for this purpose. Otherwise, apply small quantity of oil externally at the lip until seal is run in. Stuffing box should be gradually tightened to avoid overheating. Packing should be self-lubricating braided-type.
	6. Breather	Breather should be open and clean. Clean breather regularly in a solvent.
	7. Grade of oil	Oil must be of grade specified in lubrication instructions. If not, clean unit and refill with correct grade.
	8. Condition of oil	Check to see if oil is oxidized, dirty, or of high sludge content; change oil and clean filter.
	9. Forced-feed lubrication system	Make sure oil pump is functioning. Check that oil passages are clear and permit free flow of lubricant. Inspect oil line pressure regulators, nozzles, and filters to be sure they are free of obstructions. Make sure pump suction is not sucking air.
	10. Coupling alignment	Disconnect couplings and check alignment. Realign as required.
	11. Coupling lateral float	Adjust spacing between drive motor, and so on, to eliminate end pressure on shafts. Replace flexible coupling with type allowing required lateral float.
	12. Speed of unit	Reduce speed or replace with drive suitable for speed.

TABLE 2 Continued.

Trouble	What to inspect	Action
Shaft failure	1. Type of coupling used	Rigid couplings can cause shaft failure. Replace with coupling that provides required flexibility and lateral float.
	2. Coupling alignment	Realign equipment as required.
	3. Overhung load	Reduce overhung load. Use outboard bearing or replace with unit having sufficient capacity.
	4. Unit overload	Reduce loading or replace with drive of sufficient capacity.
	5. Presence of high-energy loads or extreme repetitive shocks	Apply couplings capable of absorbing shocks and, if necessary, replace with drive of sufficient capacity to withstand shock loads.
	6. Torsional or lateral vibration condition	These vibrations can occur through a particular speed range. Reduce speed to at least 25% below critical speed. System mass elastic characteristics can be adjusted to control critical speed location. If necessary, adjust coupling weight, as well as shaft stiffness, length, and diameter. For specific recommendations, contact factory.
	7. Alignment of outboard bearing	Realign bearing as required.
Bearing failure	1. Unit overload	See “Overheating” (item 1). Abnormal loading results in flaking, cracks, and fractures of the bearing.
	2. Overhung load	See “Shaft Failures” (item 3).
	3. Bearing speed	See “Overheating” (item 12).
	4. Coupling alignment	See “Overheating” (item 10).
	5. Coupling lateral float	See “Overheating” (item 11).
	6. Bearings adjustment	See “Overheating” (item 4). If bearing is too free or not square with axis, erratic wear pattern will appear in bearing races.
	7. Bearings lubrication	See “Overheating” (items 2, 3, 7, 8, 9). Improper lubrication causes excessive wear and discoloration of bearing.
	8. Rust formation due to entrance of water or humidity	Make necessary provisions to prevent entrance of water. Use lubricant with good rust-inhibiting properties. Make sure bearings are covered with sufficient lubricant. Turn over gear unit more frequently during prolonged shutdown periods.

TABLE 2 Continued.

Trouble	What to inspect	Action
	9. Bearing exposure to abrasive substance	Abrasive substance will cause excessive wear, evidenced by dulled balls, rollers, and raceways. Make necessary provision to prevent entrance of abrasive substance. Clean and flush drive thoroughly and add new oil.
	10. Damage due to improper storage or prolonged shutdown	Prolonged periods of storage in moist air and at ambient temperatures will cause destructive rusting of bearings and gears. When these conditions are found to have existed, the unit must be disassembled and inspected and damaged parts either thoroughly cleaned of rust or replaced.
Oil leakage	1. Oil	Check through level indicator that oil level is precisely at level indicated on housing.
	2. Open breather	Breather should be open and clean.
	3. Open oil drains	Check that all oil drain locations are clean and permit free flow. Drains are normally drilled in the housing between bearings and bearing cap where shafts extend through caps.
	4. Oil seals	Check oil seals and replace if worn. Check condition of shaft under seal and polish if necessary. Slight leakage normal, required to minimize friction and heat.
	5. Stuffing boxes	Adjust or replace packing. Tighten packing gradually to break in. Check condition of shaft and polish if necessary.
	6. Force-feed lubrication to bearing	Reduce flow of lubricant to bearing by adjusting orifices. Refer to factory.
	7. Plugs at drains, levels, and so on, and standard	Apply pipe joint sealant and tighten fittings.
	8. Compression-type pipe fittings	Tighten fitting or disassemble and check that collar is properly gripping tube.
	9. Housing and caps	Tighten cap screws or bolts. If not entirely effective, remove housing cover and caps. Clean mating surfaces and apply new sealing compound (Permatex #2 or equal). Reassemble. Check compression joints by tightening fasteners firmly.
Gear wear	1. Backlash	Gear set must be adjusted to give proper backlash. Refer to factory.
	2. Misalignment of gears	Make sure that contact pattern is above approximately 75% of race, preferably in center area. Check condition of bearings.
	3. Twisted or distorted housing	Check shimming and stiffness of foundation.

TABLE 2 Continued.

Trouble	What to inspect	Action
	4. Unit overload	See "Overheating" (item 1).
	5. Oil level	See "Overheating" (item 3).
	6. Bearings adjustment	See "Overheating" (item 4).
	7. Grade of oil	See "Overheating" (item 7).
	8. Condition of oil	See "Overheating" (item 8).
	9. Forced-feed lubrication	See "Overheating" (item 9).
	10. Coupling alignment	See "Overheating" (item 10).
	11. Coupling lateral float	See "Overheating" (item 11).
	12. Excessive speeds	See "Overheating" (item 12).
	13. Torsional or lateral vibration	"See Shaft Failure" (item 6).
	14. Rust formation due to entrance of water or humidity	See "Bearing Failure" (item 8).
	15. Gears exposure to abrasive substance	See "Bearing Failure" (item 9).

FURTHER READING

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