GEARS

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1.0 SCOPE

This data sheet provides loss prevention recommendations, inspection and maintenance guidance, and support for recommendations related to industrial enclosed-gear and open-gear drives, gear sets and gear boxes, as well as ring gears and pinion gears used in the mining industry.

1.1 Changes

July 2023. Interim revision. Guidance on gear examinations was revised to allow for visual internal inspections (borescope) in place of an open cover examination.

2.0 LOSS PREVENTION RECOMMENDATIONS

2.1 Operation

Establish and implement an operator training program. See Data Sheet 10-8, *Operators*, for guidance.

2.1.1 Verify that gears are manufactured through a special heat treatment process. Heat treatment will increase the wear and pitting resistance and strengthen the properties of the gears.

2.1.2 Provide high-viscosity gear oil unless otherwise specified by the gear manufacturer. Verify the type and grade as specified by the OEM or American Gear Manufacturers Association (AGMA) standard.

2.1.3 Provide a positive spray lubrication system with a filter to avoid excessive wear condition for splash-fed gears application.

2.1.4 Verify the load on the tooth surface is maintained within specified limits and evenly distributed between the teeth to prevent destructive pitting.

2.1.5 Provide overload protection devices, such as torque-limiting couplings with shear sections for protection against overload breakage.

2.1.6 Provide protection to keep foreign material out of the mesh for gears that are not required to be enclosed.

2.1.7 Verify replacement oil is sampled and specifications are confirmed prior to use.

2.1.8 Provide properly segregated and labeled storage containers for bulk oil storage. Verify oil is sampled and specifications are confirmed prior to use.

2.2 Inspection, Testing, and Maintenance

Establish and implement a gear inspection, testing and maintenance program. See Data Sheet 9-0, *Asset Integrity* for guidance on developing an asset integrity program.

A gear set must be supplied with copious amounts of oil to lubricate and clean the mesh, carry away the heat and lubricate the bearings.

2.2.1 Initial Two Weeks

2.2.1.1 Install new charge of oil after flushing the casing with light flushing oil. Remove metallic impurities from the tooth surfacess or any impurities the oil may wash out of the casing surfaces.

2.2.1.2 Visually inspect the gear and teeth after the initial oil has been drained, and the system has been flushed.

2.2.1.3 Perform NDE, and check the tooth contact and wear patterns. One recommended method is to use plastic adhesive tapes, which make it easy to establish a permanent record of tooth contact.

2.2.2 Weekly

2.2.2.1 Verify the oil level is maintained.

2.2.2.2 Investigate any oil leaks. Keep the gear casing and the foundation clean and free of oil.

2.2.2.3 Where fixed vibration monitoring is used, visually check the vibration level at each point and record. Investigate all alarms, verify the cause, and take the necessary corrective action; record findings and the data.

2.2.2.4 Where manual (hand-held) vibration monitoring is used, take readings at each point and record the results. Vibration readings should be taken under the same load conditions; increase interval as necessary based on history.

2.2.2.5 Establish baseline any time an overhaul is performed, and more frequently if adjustments to alignment or balancing are made.

2.2.3 Quarterly

2.2.3.1 All gears critical to production (high-speed application) especially when subject to severe loading, Perform a complete spectrographic/ferrography oil analysis where all the elements in an oil sample are identified, especially physical breakdown, metallic elements, the viscosity, and the presence of the following:

- Solids content (contaminants, dirt)
- Water content
- Acidity (total acid number - TAN)
- Base (total base number)
- Wear partials, metal particles: Fe, Cr, Mo, Pd, Sn, Cu
- Additives

Log results and review for wear trends.

2.2.4 Annually

2.2.4.1 Test and calibrate the critical field devices (i.e., pressure, temperature, level, flow, vibration sensors) annually and after every major scheduled outage, or as necessary based on the historical operation of the devices.

2.2.4.2 Perform a visual inspection of all gear surfaces, including teeth flanks and roots, and all accessible shaft bearings at least annually. Use of a borescope through handholes or ports in an enclosed gear casing is acceptable. Additional gear examinations are warranted if either of the following occurs:

A. Visual inspection identifies significant changes in gear or bearing condition, or gear or bearing damage. Evaluate the gear and/or bearing condition by conducting an open cover examination, which may include additional nondestructive examination (NDE) techniques, along with inspecting backlash, casing condition, and gear and shaft alignment.

B. Following an upset condition at the driven object or driver that can lead to gear overload (excessive torque), conduct a series of frequent internal visual inspections discussed previously; or consider conducting an open cover examination per Part A.

2.2.4.3 Visually check the contact lines and tooth profiles if excessive wear is occurring.

2.2.5 Biennially (Every Two Years)

2.2.5.1 Examine sleeve bearings unless conditions warrant more frequent inspection.

2.2.6 Major Outage

Thoroughly clean the lubricating system and replace the gear lubricant.

2.3 Monitoring and Protection for High-Speed Gear Sets Critical to Production

Provide gearboxes with monitoring and protective devices as listed in Table 1.

Table 1. Gearboxes Monitoring and Protective Devices for High-Speed Gear Sets Critical to Production

Monitoring & Protective Device	*Local Indication*	*Automatic Trip*	*Alarm (Control room) + Trip*
Vibration (input & output shaft only)		X	
Vibration (input & output shaft + Inner bearings)			X
Bearing temperature (input & output shaft only)	X	X	
Bearing temperature (input & output shaft + inner bearings)			X
Oil temperature	X	X	
Oil pressure	X	X	
Oil Level	X	X	
Overload (overcurrent [electric motor driven]) or shear coupling		X	X

2.3.1 Vibration monitoring is necessary to assist in the evaluation of the gear condition. Install vibration monitoring devices alarming and tripping on both the shaft and casing of gear sets critical to production (high speed). Monitors should be set to shut the units down if vibration exceeds limits. A set of established limits should be available from the manufacturer giving normal, tolerable or alarming vibrations. Contact the manufacturer for advice when excessive vibration or upward trend is noted.

2.3.2 Monitor oil pressure, oil temperature, and level-indicating devices regularly to determine any changes/ variation.

2.4 Grinding Mills and Gears in the Mining Industry

Large hardened ring (also called girth, bull) gears are used for rotating rod, ball and semi-autogenous grinding (SAG) mills in the mining industry. Ring gears are also used for rotating equipment such as kilns in cement plants, but these are usually smaller in diameter and run slower than those in the mainstream mining industry.

Mine industry ring gears can be up to 38 ft (12 m) in diameter and have large (up to 40,000 hp) motors. Due to size and difficulties in shipping and installation they normally are built in three or four sections.

The industry has had many losses due to broken or cracked teeth, and one damaged tooth can mean total gear replacement as these gears can rarely be repaired. Replacement times have historically been as high as 4 years but are now in the 12 to 18 month range (there are only three major global producers of mining ring gears and there has been a long lead time due to demand during “boom” times).

The following section is specific for grinding mill ring gears used in the mining industry. See Data Sheet 7-12, *Mining and Mineral Processing*, for additional ring gear guidance.

2.4.1 Verify an initial baseline inspection was completed (i.e., 100% examination using dye penetrant or magnetic particle testing to ensure there are no relevant indications prior to placing the gear in service).

2.4.2 For a cast gear, obtain a report on quality acceptance and a mapping of any inherent flaws that are present. These can also be used for establishing a calibration standard for the NDE firm.

2.4.3 Provide gear drawings and specifications.

2.4.4 Provide temperature monitoring and alarm of the lubricating oil circuit of gears for large SAG/Ball/ autogenous mills, where the pumped lube oil leaves the gears.

2.4.5 Apply the following recommendations (gear evaluation and performance monitoring) for gears that have been placed in service. When evaluating a gear, the following should be verified:

A. Operating conditions (daily)

1. Verify effective loading on the mill is within design specification.

2. Record number of starts and stops as this could place heavy stress on a gear. These should be used as a parameter for gear visual inspection.

3. Provide a starting interlock to ensure proper lubrication for spray-type lube oil system, and verify alarm for low oil pressure, bearing temperature, high oil temperature, oil level, filter pressure differential and for an air injection spray system (lube fault alarms for the injectors) at a minimum.

4. Visually inspect the lube system, filters and gages for signs of leakage, abnormal or unusual noises, and vibration. Some lube systems have external filtration systems that should be included.

B. Gear alignment (daily)

Failure due to high tooth temperatures can be prevented with the knowledge of temperature distribution in gear teeth under operation. Verify one of the following methods is utilized to detect gear alignment daily. Results should be recorded and trended.

1. Strobbing the gear

2. Infrared temperature

3. Temperature profiling

C. Vibration monitoring (max interval 4 months)

1. Where fixed vibration monitoring devices are provided with alarms, and trip functions built in to the PLC logic. Verify vibration readings are recorded and trended.

2. Where manual (hand-held) vibration monitoring is used, take readings at each point, record results. Vibration readings should be taken under the same load conditions; increase interval as necessary based on history.

D. Lubricant

1. Sample lubricating oil for entrapped metals, particle count, viscosity, contamination, and additive breakdown every 6 months.

2. Visually inspect the lube oil system daily and replace filters as needed.

E. Tooth Contact

1. Annually, verify tooth contact is within 100% contact at the pitch line. The backlash and the alignment tolerance are acceptable to the gear manufacturer specification or AGMA standards. Taking backlash measurements can be effective in depicting outside tolerances so corrective action can be taken.

2.5 Contingency Planning

2.5.1 Equipment Contingency Planning

When a gear breakdown would result in an unplanned outage to site processes and systems considered key to the continuity of operations, develop and maintain a documented, viable gear equipment contingency plan per Data Sheet 9-0, *Asset Integrity*. See Appendix C of that data sheet for guidance on the process of developing and maintaining a viable equipment contingency plan. Also refer to sparing, rental, and redundant equipment mitigation strategy guidance in that data sheet.

In addition, include the following elements in the contingency planning process specific to gears:

A. Gear design/construction information, including materials of construction, drawings and specifications.

B. Gear component repair/replacement options/sources strategy and repair history.

C. Gear system dismantle/re-assembly considerations.

3.0 SUPPORT FOR RECOMMENDATIONS

A gear is generally defined as a toothed wheel, which may be driven by a prime mover or motor and used to move another similar toothed wheel, thereby transmitting power. When gears are used to transmit power, change in speed, direction of rotation, or axis of rotation are required.

Gears are designed for a specific loading arrangement. The most common and important design in a set of gears is speed change or speed ratio. This ratio is determined by the number of teeth in the driving gear and the number of teeth in the driven gear.

Selection of the type of gear depends upon a variety of operational parameters, such as; the shaft configuration, gear ratios, power transition curves, operating conditions (environment, loading, forward and reversing cycles), etc.

Gear lubrication and the selection of the type of lube oil is important for the operation and the gear application.

The gear manufacturer, lube oil contractor, and/or lubricating engineer will supply all of the information for a lubricating program. Every effort is made by gear manufacturers to make their products compatible with common lube oil inventories.

Oil, like everything else, comes in many grades, types, qualities, brand names, etc. (Some plants that have made a critical study of their lube oil inventory were astonished to find out how many different types they had on hand. The main difference in oils lies between industrial and automotive lubricants.)

Using the incorrect gear configuration will result in failure of the gear under load. A gear is considered to have failed when it can no longer do the job for which it was designed. There are several types of failure: wear, pitting, scoring, fracture, and plastic flow.

3.1 Gear Failure

A. Typical cause of gear failure include the following:

- Improper/poor lubrication. Most common failure mode on gearbox result of bearing failures.
- Improper alignment of the gears shafts
- Improper operation
- Overloading
- Excessive heating
- Mechanical failures
- Wear/erosion/corrosion
- Poor maintenance

B. Types of gear failure:

- Wear
- Pitting/surface fatigue
- Scoring
- Fracture/breakage/cracked teeth
- Plastic flow

3.2 Gear Alignment

Gear alignment is essential for operation, so a specific maintenance routine needs to be adopted using proven inspection methods. The specific methods are as follows:

- **Strobbing**. This is a relatively simple and revealing method to dynamically and safely inspect drive gear during operation. Using a strobe light to ″freeze″ the image of mating gears can reveal how well the subject gears are meshing and transmitting drive power to the section. This is an excellent method to detect proper mesh contact and gear alignment. It takes a little time and practice to become proficient at understanding what is being seen during the strobe inspection, but the investment in time is worthwhile. Performing periodic strobe inspections will help identify problems before they become catastrophes, and is a good method for planning gear maintenance ahead of a shutdown.

- **Infrared temperature measurements while gear is operating**. This monitors load distribution, and provides the operator with a method of scheduling and planning the specifics of any alignment that may need to be done due to foundation shifts.
- **Temperature profiling**. Use a heat gun to measure the temperature looking for refractory failures, and surface fatigue criteria. If the hot spots are found, a corrective action should be taken. A relatively uniform temperature is the desired result.

4.0 REFERENCES

4.1 FM

Data Sheet 7-12, *Mining and Mineral Processing*
Data Sheet 9-0, *Asset Integrity*
Data Sheet 10-8, *Operators*

4.2 Other

The American Standards Association (ASA) standards
American Gear Manufacturers Association (AGMA) standards

APPENDIX A GLOSSARY OF TERMS

Refer to Section C.1.2, Gear Nomenclature, for defined terms.

APPENDIX B DOCUMENT REVISION HISTORY

The purpose of this appendix is to capture the changes that were made to this document each time it was published. Please note that section numbers refer specifically to those in the version published on the date shown (i.e., the section numbers are not always the same from version to version).

July 2023. Interim revision. Guidance on gear examinations was revised to allow for visual internal inspections (borescope) in place of an open cover examination.

July 2021. Interim revision. Minor editorial changes were made.

July 2020. Interim revision. Updated contingency planning and sparing guidance.

April 2014. Minor editoril changes were made.

October 2013. This document has been updated to reflect current technology and industry practice.

January 2000. This revision of the document has been reorganized to provide a consistent format.

March 1983. Completely rewritten.

March 1980. First issued.

APPENDIX C SUPPLEMENTARY INFORMATION

C.1 Description

C.1.1 General

Gears are the most durable and rugged of all mechanical drives. Their primary function is to transmit rotary motion from one shaft to another without slippage. They can transmit high power at efficiencies up to 98% and with long service lives. For this reason, gears rather than belts or chains are found in most heavy-duty machine drives.

Power transmission can be effected by gears mounted on parallel, intersecting, or skewed shafts, and operating in an open environment or in enclosed gear cases.

C.1.2 Gear Nomenclature (Fig. 1)

Involute teeth of spur gears, helical gears, and worms are those in which the profile in a transverse plane (exclusive of the fillet curve) is the involute of a circle. Since the earliest appearance of geared mechanisms, many curves have been used as a basis for the forms of the gear teeth. In all cases curves were selected to provide constant relative velocity of the gear tooth faces. The introduction of gear-cutting machinery resulted in a gradual transition of tooth forms to the point where the involute curve (Fig. 2) is now considered to be the most practical curve for gear teeth. Present day power transmission demands precision gearing and the involute system provides a means of controlling quality to meet exacting requirements. In addition, gears of the involute type can be operated on variable center distances without affecting the theoretically correct action of the teeth in mesh. These factors have resulted in universal acceptance of the involute curve system as the basis for gearing. The America Standards Association (ASA), as well as the American Gear Manufacturers Association (AGMA), uses the involute system in its standardization program.

The *base circle* is the circle from which involute tooth profiles are derived.

A *pitch circle* is the curve of intersection of a pitch surface of revolution and a plane of rotation. According to theory, it is the imaginary circle that rolls without slipping with a pitch circle of a mating gear.

A *pitch line* corresponds in the cross section of a rack to the pitch circle in the cross section of a gear.

The *addendum* circle coincides with the tops of the teeth in a cross section.

The *root circle* is tangent to the bottoms of the tooth spaces in a cross section.

The *line of action* is the path of contact in involute gears. It is the straight line passing through the pitch point and tangent to the base circles.

Pressure angle is the angle between a tooth profile and the line normal to a pitch surface, usually at the pitch point of profile. This definition is applicable to every type of gear. The term pressure angle originally meant an angle between the line of pressure and the pitch circle. In involute teeth, pressure angle is often described as the angle between the line of action and the line tangent to the pitch circles.

Center distance is the distance between parallel axes of spur gears and parallel helical gears or between the crossed axes of crossed helical gears and worm gears. Also, it is the distance between the centers of pitch circles.

Addendum (a) is the height by which a tooth projects beyond the pitch circle or pitch line; also, it is the radial distance between the pitch circle and the addendum circle.

Dedendum (b) is the depth of a tooth space below the pitch circle or pitch line; also, it is the radial distance between the pitch circle and the root circle.

Clearance is the amount by which the dedendum in a given year exceeds the addendum of its mating gear.

Working depth is the depth of engagement of two gears; that is, the sum of their addendums.

Whole depth is the total depth of a tooth space, equal to addendum plus dedendum, also equal to working depth plus clearance.

Pitch diameter (D, d) is the diameter of the pitch circle.

Outside diameter is the diameter of the addendum (outside) circle. In a bevel gear, it is the diameter of the crown circle. In a throated worm gear, it is the maximum diameter of the blank. The term applies to external gears.

Root diameter is the diameter of the root circle.

Circular thickness is the length of arc between the two sides of a gear tooth on the pitch circles unless otherwise specified.

Chordal thickness is the length of the chord subtending a circular-thickness arc.

Chordal addendum is the height from the top of the tooth to the chord subtending the circular-thickness arc.

Number of teeth or threads (N) is the number of teeth contained in the whole circumference of the pitch circle.

Gear ratio — Speed ratio is the relationship in rpm (revolutions per minute) of the driver to the driven equipment in rpm. This ratio is determined by the number of teeth in the driven gear. For example, if the driving

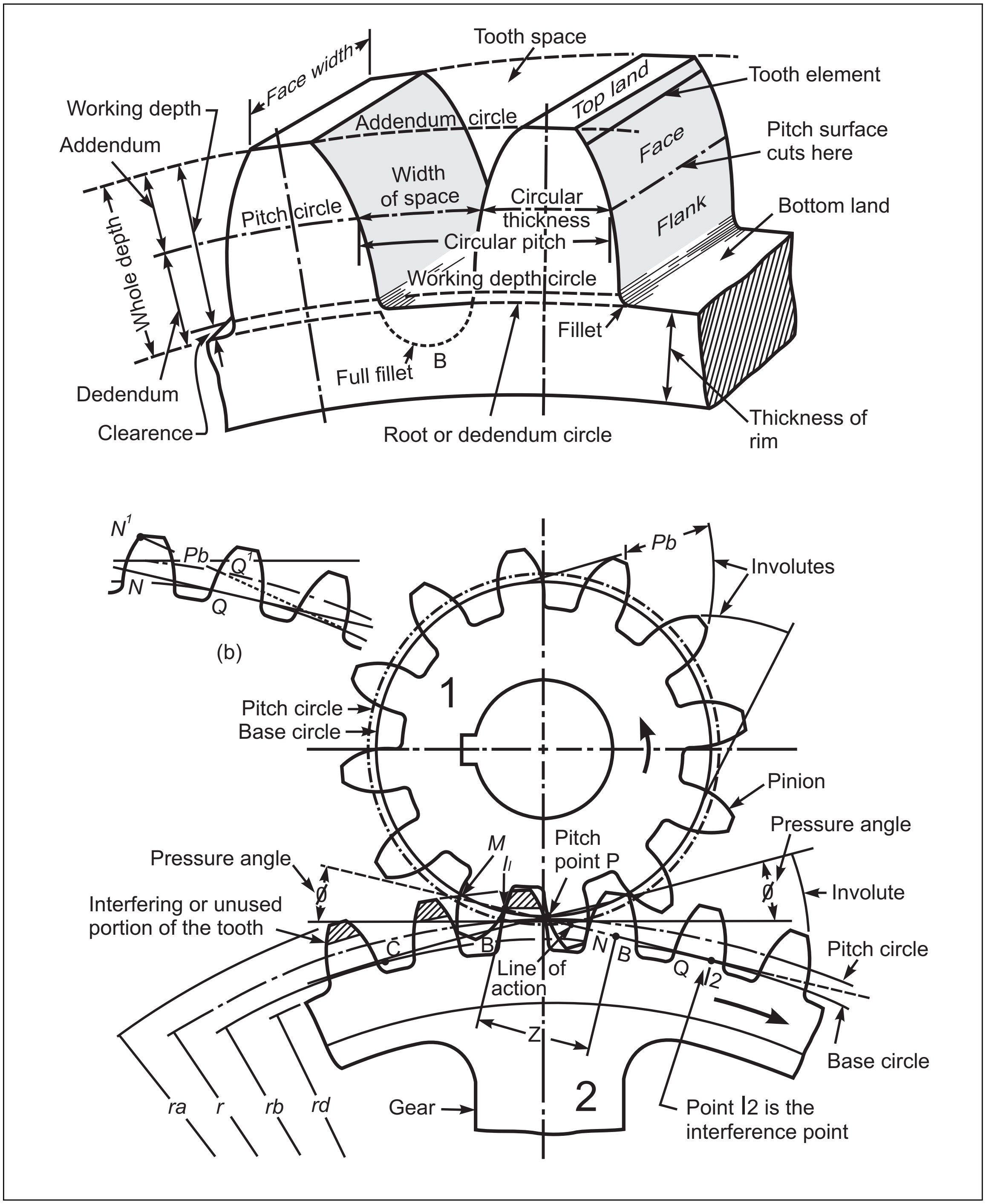

Fig. 1. Gear nomenclature.

gear, which is called the pinion, has 50 teeth and the driven gear has 500 identical teeth, then when the pinion has made one revolution, the driven gear has made 1/10 of a revolution (500/50) and the gear ratio is designated as 10 to 1.

A speed ratio of about 6 is the upper limit for a fairly high load on a single reduction gear set with straight tooth gears. Higher single reductions up to 12 are possible with helical gears. Ratios above 12 require double reduction.

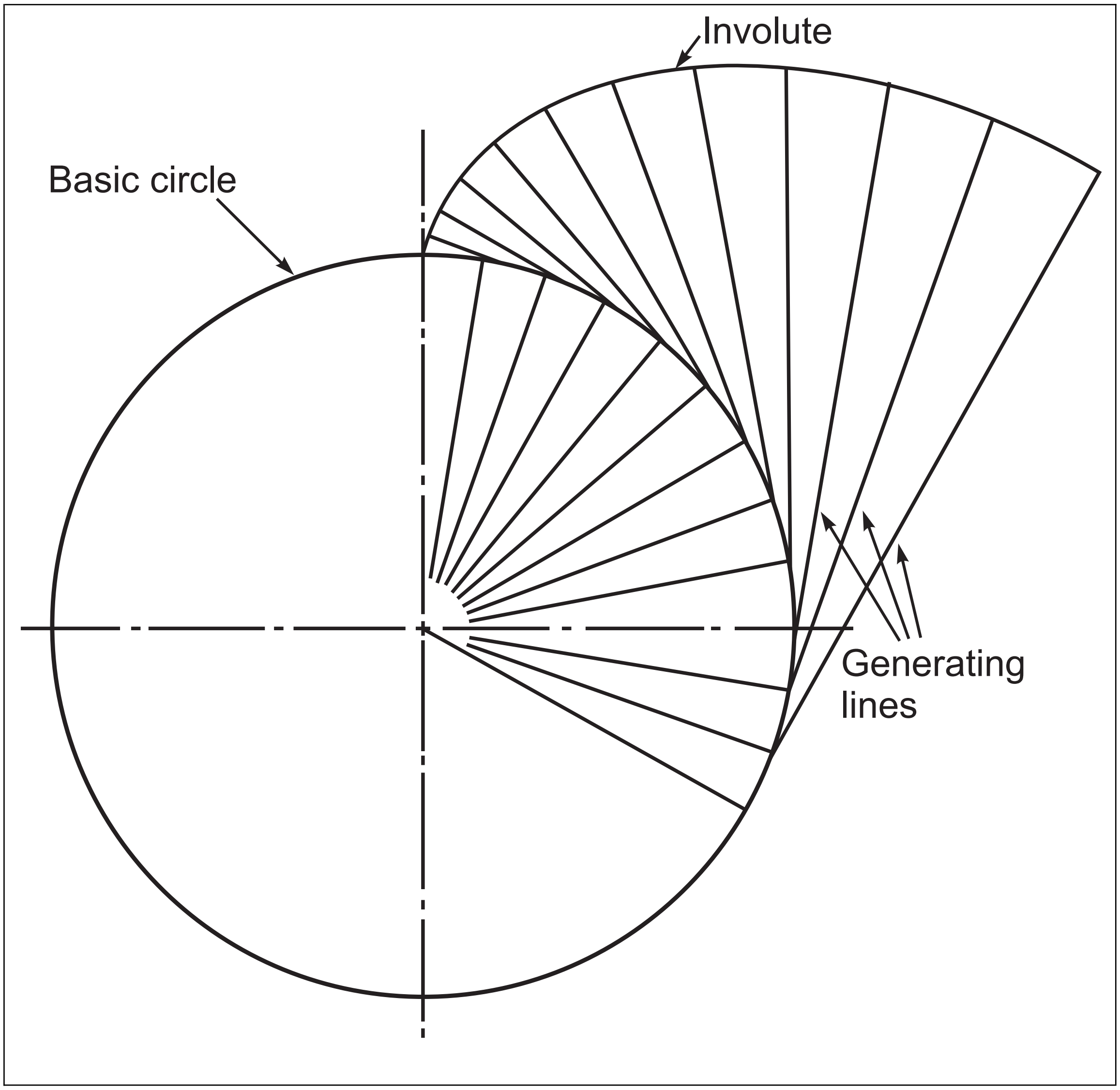

Fig. 2. Developed involute.

The computation of the ratios of multiple speed reductions (trains) is the multiplication of the separate single ratio values. For example, if the total speed reduction gear is 30 to 1, it may be made up of a set of gears with a reduction ratio of 5 to 1 in the first reduction and a ratio of 6 to 1 in the second reduction.

Full-depth teeth are those in which the working depth equals 2.000 divided by normal diametral pitch.

Stub teeth are those in which the working depth is less than 2.000 divided by normal diametral pitch.

C.1.3 Parallel Shafts

Parallel shafts utilize the following types of gears: straight, helical, double helical, and herringbone.

C.1.3.1 Straight Gear (Spur Gear) (Fig. 3).

The straight gear is the simplest type, having straight teeth cut parallel to its shaft In this type, not more than two teeth are in mesh at any one time and load is transferred from one tooth to another very quickly. Because of this action, spur gears are limited to comparatively slow speeds and are inclined to be noisy in operation. They are used, however, very extensively on many applications where a reliable slow-speed drive is required.

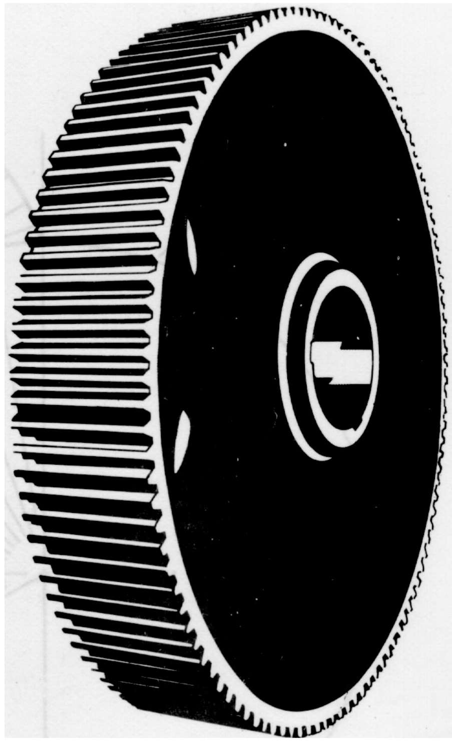

Fig. 3. Straight spur gear.

C.1.3.2 Helical Gears (Fig. 4).

The helical gear has teeth that form part of a helix. With this gear, several teeth are in mesh at the same time. The teeth mesh at one end, gradually roll together and break contact at the other end. Due to this action, helical gears are generally quieter than spur gears. On the other hand, an end thrust is produced by one gear, tending to push the other axially (which action increases with helix angle). Therefore, thrust bearings are required for the gear shaft.

C.1.3.3 Double Helical Gears (Fig. 5).

Double helical gears not only cancel out the thrust forces set up in the single helical types, but permit the use of greater helix angles. This, in turn, permits more tooth overlap with greater overall strength and smoother load transfer.

C.1.3.4 Herringbone Gear.

The continuous and separated design of herringbone gears is a further extension of double helical advantages. In the continuous design there is no space separating the two opposed sets of helical teeth, and this contributes to the gears' great strength. Fig. 6 illustrates a herringbone gear with gear teeth separated only sufficiently for cutter clearance. Herringbones are particularly suitable for heavy load transmission at high speeds in continuous service. (See Table 2.)

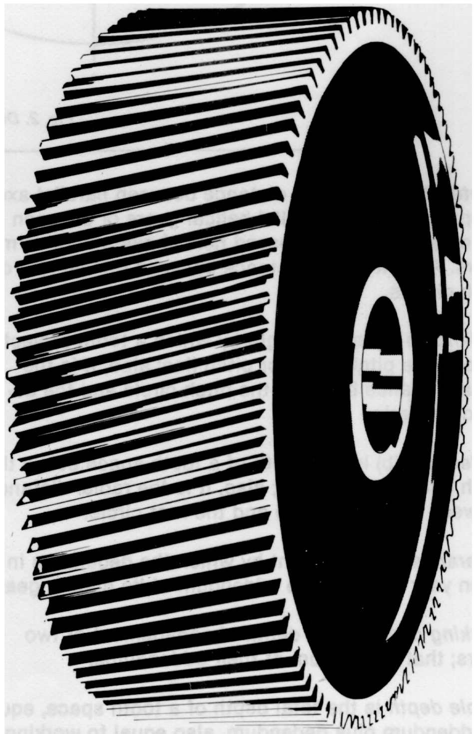

Fig. 4. Helical gear.

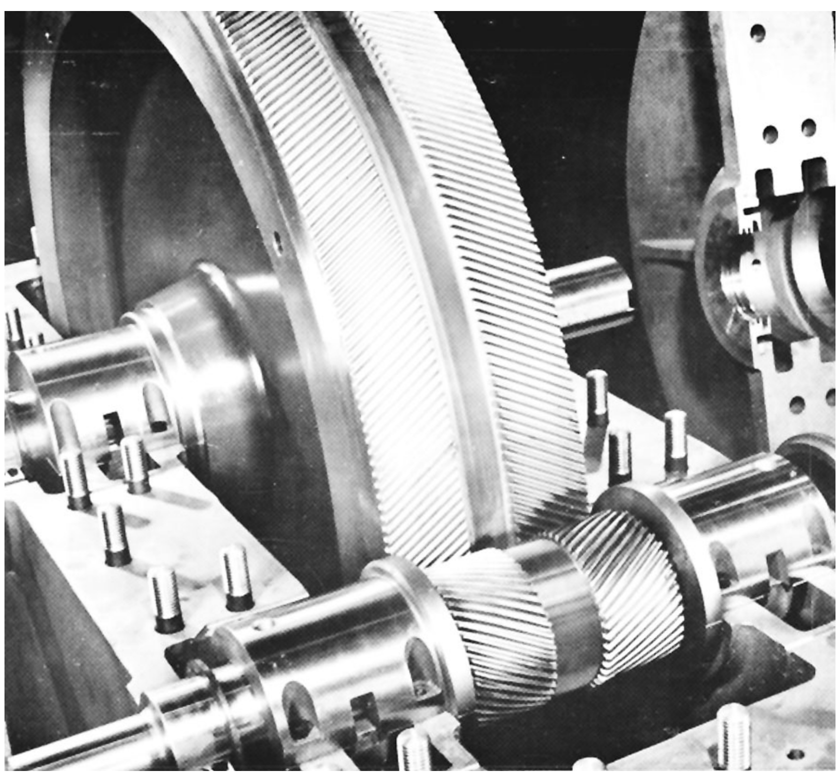

Fig. 5. Double helical gear. (Worthington Corp.)

Fig. 6. Herringbone gear

Table 2. Comparison of Single Mesh Gears.

Type of Gearing	*Gear Ratio Range*	*Efficiency at Rated Power,*	*Maximum Pitchline Velocity ft/min.*	*m/s*
Spur	1 to 10	98	2,000	10
Helical & Herringbone	1 to 15	98	5,000	25
Helical and Double Helical	1 to 15	98	30,000	150
Crossed Helical	1 to 10	98	4,000	20
Straight Bevel	1 to 6	98	1,000	5
Spiral Bevel	1 to 9	98	8,000	40
Zerol	1 to 9	98	4,000	20
Hypoid	1 to 9	98	4,000	20
Worm	3½ to 90	50 to 90	6,000	30
Double-Enveloping Worm	3½ to 90	50 to 98	4,000	20
Face	3 to 8	95 to 99	4,000	20
Spiroid	10 to 100	50 to 97	6,000	30
Helicon	3 to 100	50 to 98	6,000	30
Beveloid	1 to 100	50 to 95	4,000	20

C.1.4 Intersecting Shafts

Intersecting shafts utilize the following types of gears: straight bevel, spiral bevel, and zerol.

C.1.4.1 Straight Bevel (Fig. 7).

Straight bevel gears transmit power between two shafts which are usually at right angles with each other; however, shafts at other than 90° can be used. Speeds between shafts can be increased or decreased by varying the number of teeth on pinion and gear. They are constructed in pairs and are not always interchangeable.

Straight bevel gears are considered satisfactory for pitchline velocities up to 1,000 ft/min (5 m/s). These gears are more economical than spiral bevel gears for right angle power transmission where operating conditions do not warrant the superior characteristics of spiral tooth.

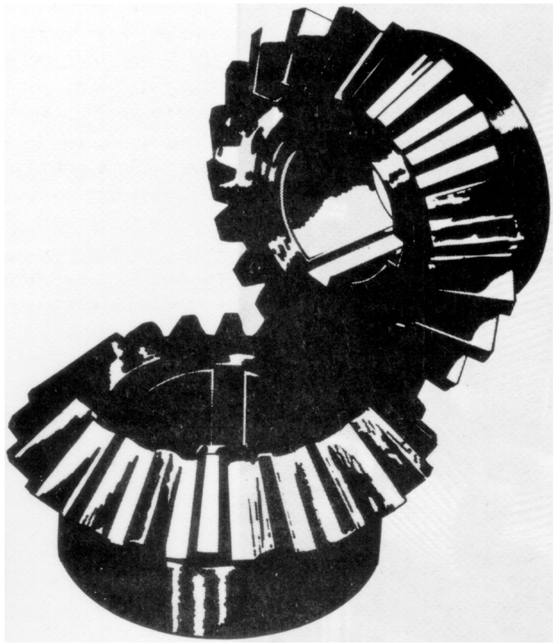

Fig. 7. Bevel gears.

When shafts are at right angles and both shafts turn at the same speed (ratio 1:1), the two bevel gears can be alike and are called miter gears. Figure 7 illustrates one application.

C.1.4.2 Spiral Bevel (Fig. 8B).

Spiral bevel gears carry higher loads and run more quietly than straight bevel gears. They are more expensive and have greater thrust loads. As teeth are at an angle to the axis of rotation, loading may be distributed over two or more teeth at any instant, depending on spiral angle. Such gears are suitable for pitchline velocities up to 8,000 ft/min (40 m/s). Hardened and ground teeth extend this limit to 18,000 ft/min (90 m/s). Figure 9 illustrates the use of these gears in a right angle gear drive for vertical fire pumps.

C.1.4.3 Zerol (Fig. 8A).

Zerol gears are similar to spiral bevel gears, but have curved teeth arranged in such a way that the effective spiral angle is zero. Thrust leads, therefore, are equivalent to these produced by straight bevel gears. Tooth bearing is localized so that stress concentration at the tips is eliminated.

C.1.5 Skewed Shafts

Skewed shafts utilize the following types of gears: crossed helical, worm, and hypoid.

C.1.5.1 Crossed Helical (Fig. 10).

Crossed helical gears, also known as spiral gears, are similar in appearance to conventional crossed helical gears. They have a low load capacity due to low tooth contact area. They are principally used to provide a wide variety of speed ratios without change in center distance or gear size. They can be used with nonintersecting, nonparallel shafts.

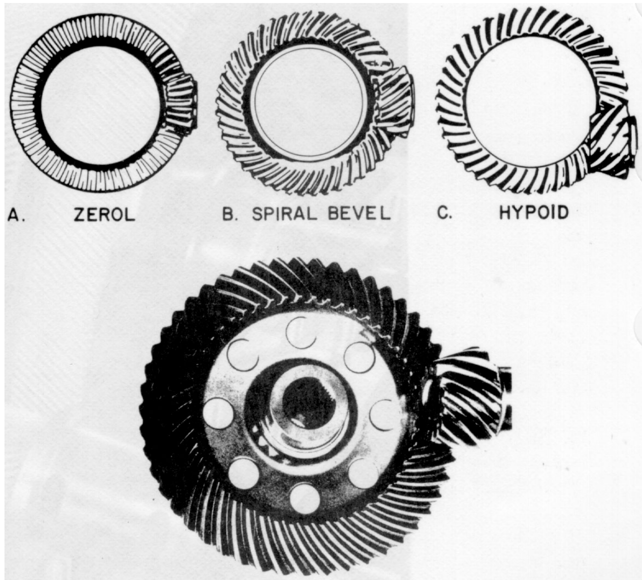

Fig. 8. Bevel gears.

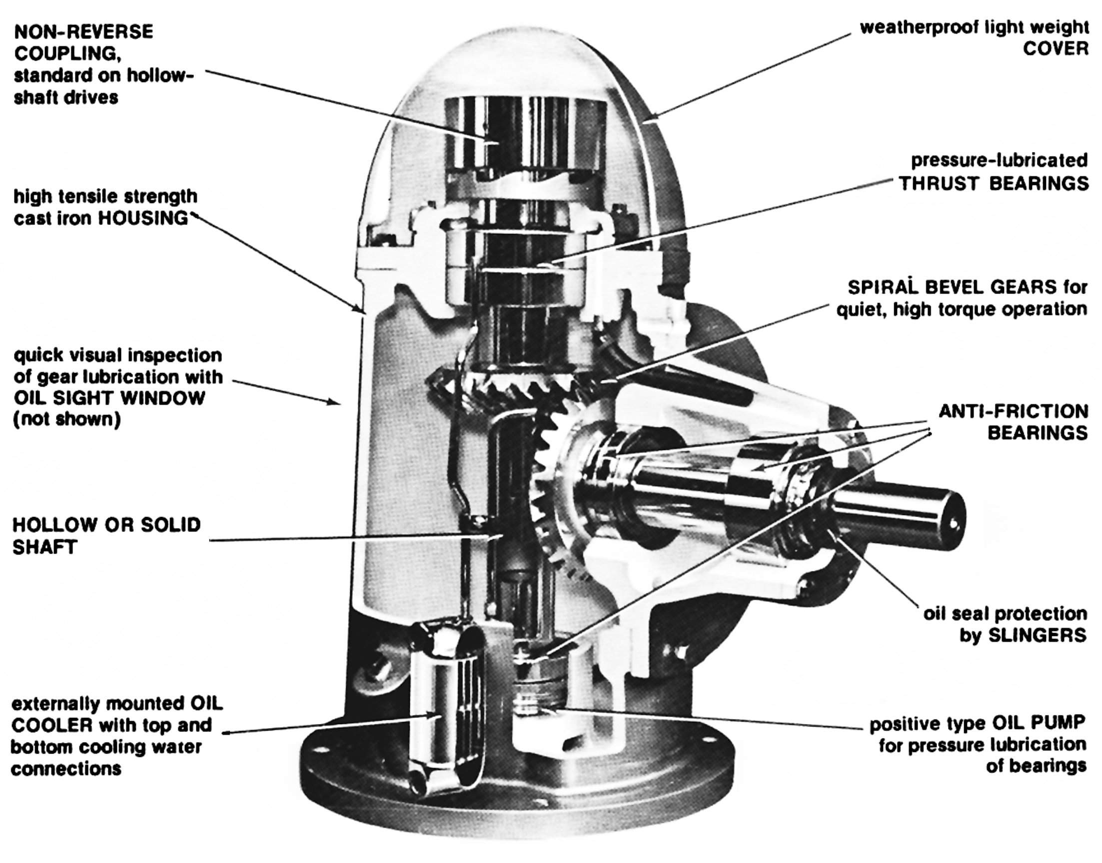

Fig. 9. Right angle gear drive. (Johnson Gear, div. Of Arrow Gear Co.)

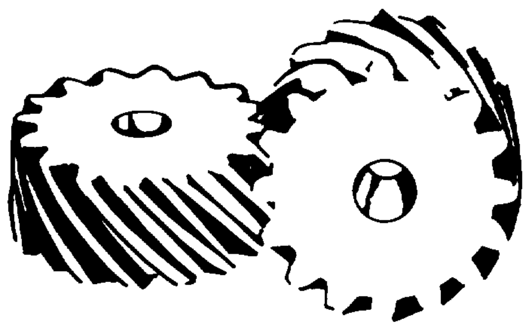

Fig. 10. Crossed helical gears.

C.1.5.2 Worm Gears (Fig. 11).

Worm gears are used to transmit power between nonintersecting shafts, always at right angles to each other. A hardened and ground integral worm gear drives a heavy forged bronze gear. Comparatively high velocity ratios may be obtained satisfactorily in a small space. They are used for shaft direction change and speed reduction. They generally cannot backdrive at ratios greater than 20:1. Contact area is large, thus load capacity is high in spite of the sliding action. Because of side slide, the contacting surfaces tend to break through the lubricating film, requiring the use of properly designed lubricant.

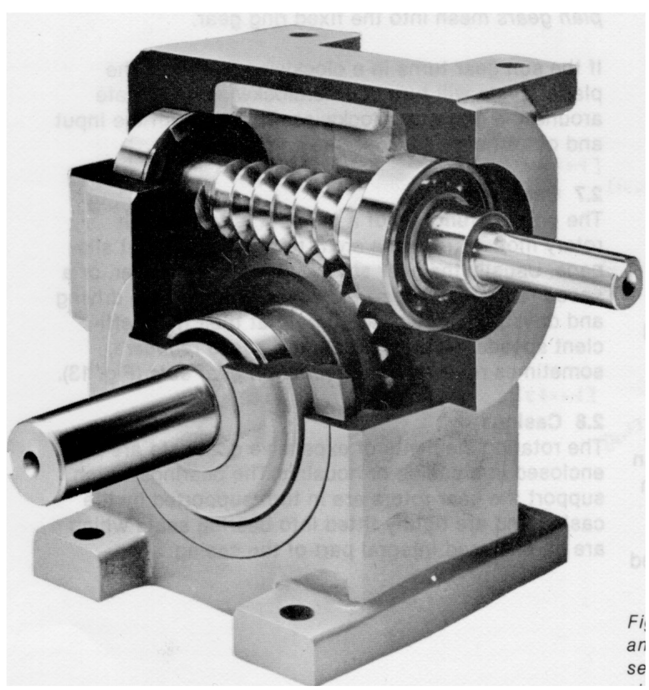

Fig. 11. Worm gears. (Winsmith Div., UMC Industries)

C.1.5.3 Hypoid Gears (Fig. 8C).

Hypoid gears are similar to spiral bevel gears except that the shafts do not intersect. They also operate more smoothly and quietly and are stronger for a given ratio. Because the shafts do not intersect, bearings can be mounted on both sides for high rigidity. The shaft angles are usually 90°, but other angles are possible. They have a higher load capacity than spiral bevel gears of the same size and ratio.

C.1.6 Internal Gears

Internal gears or annular gears have teeth cut on the inside of the rim. The shape of the teeth is the same as the shape of the space on an external gear of the same pitch diameter. These gears have the advantage of being more compact, operate more smoothly, and their teeth are stronger than in comparable external gears. When used with a single pinion, both gears operate in the same direction, where external gears run in opposite directions.

Internal gears are often made in the form of open rings, which, after having their teeth cut, may be concentrically attached to the side of a flywheel, sprocket, pulley, or other power transmission device. In other cases, gear blanks for internal gears are made in the form of a modified shallow cup, so that teeth can be cut on the inside of the rim, and the solid back of the gear may be provided with a hub, and thus be attached directly to a shaft

Figure 12 illustrates an internal gear in use as part of a planetary gear set. The center or *sun gear* is keyed to the motor shaft and driven by it. Supported in bearings in a cage that connects to the output shaft, the *plan gears* mesh into the fixed ring gear.

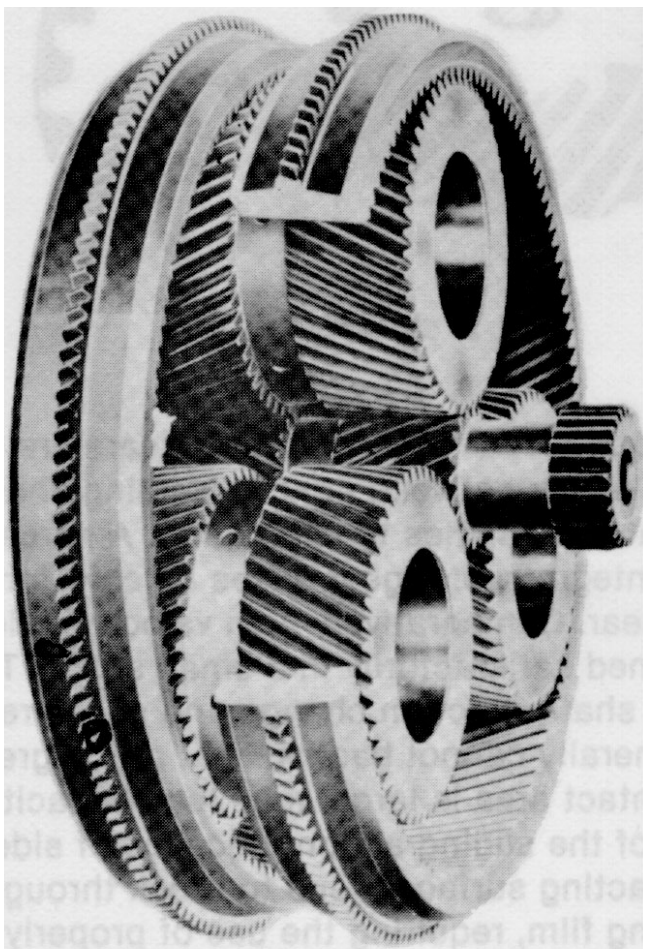

Fig. 12. The outside or ring gear shown in this assembly is an internal gear. The small spur gears form a planetary gear set, with the one in the center known as the sun gear. (Cincinnati Gear Co.)

If the sun gear turns in a clockwise direction, the planet gears will turn counterclockwise and rotate around the ring gear clockwise. That is, both the input and output shafts turn in the same direction.

C.1.7 Gear Sets

The primary function of a gear set is to transmit rotary motion from one shaft to another without slippage. Usually the gear set is a direction changer, or a speed reducer or speed increaser, enabling the driving and driven equipment to operate at their most efficient speeds. Most gear sets are *speed reducers*, sometimes referred to as *reduction gear sets* (Fig. 13).

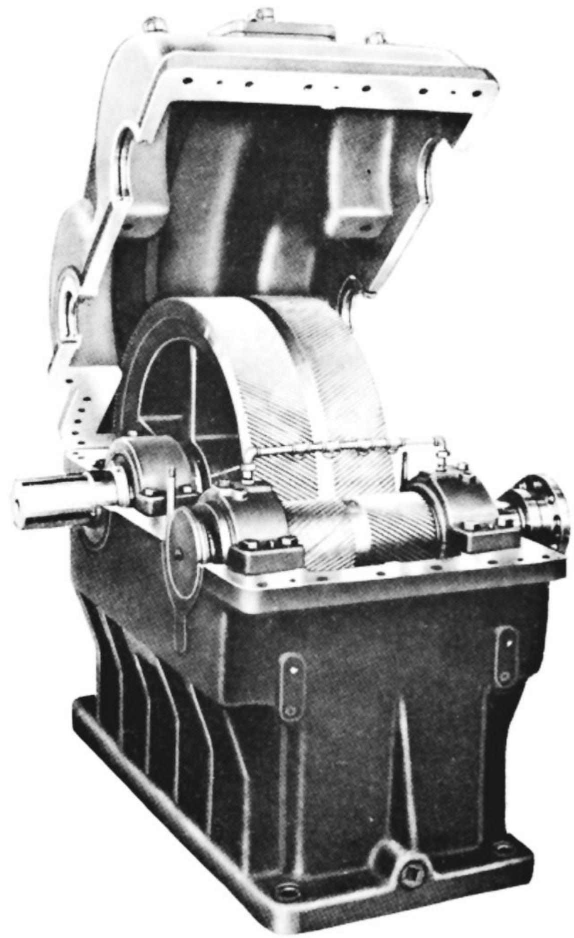

Fig. 13. Reduction gear set. (Worthington Corp.)

C.1.8 Casings

The rotating elements of expensive gear sets are enclosed in a casing or housing. The bearings which support the gear rotors are in turn supported by the casing and are tightly fitted into bearing seats which are an enclosed integral part of the casing.

Casings for totally enclosed gears of any type, whether parallel or crossed axis, are very carefully designed and sturdily built. The stiffness factor, of necessity, is high, to avoid any deflection of the bearing seats and consequent deflection of the rotating element shafts. Any deflections under load will change the alignment of the rotating elements with respect to the driving and driven objects and with respect to each other, and will seriously affect bearing wear and tooth contact. Casings must also provide a clean oily atmosphere for the bearings and the rotating elements and must also be effective radiators of the heat generated by operation of the gears.

Casings are manufactured by casting, fabrication by welding wrought steel, and/or a combination of the two methods. If cast they are made of close grained cast iron or of steel. Frequently, they are annealed or given suitable heat treatment for grain refinement or for stress relieving purposes.

Regardless of how the rough casing is produced, whether in the foundry or in the fabricating shop or both, extensive machining in the machine shop is necessary to refine the dimensions and produce acceptable smooth surface finishes.

The use of all-welded casings is increasing, but there are still many cast casings in service, especially in the small and moderate size ranges. Cast casings have relatively thick walls and flanges with additional thickness in ribbed areas so located to provide support for the bearings. The inside areas of the casing castings are usually free of ribs and pockets and must be thoroughly cleaned of debris from the fabrication shop. In service, hot lubricating oil very efficiently detaches foreign material imbedded in or attached to metal surfaces.

Welded construction is lightweight, quickly built and economical. Another big advantage is that it frees the designer from pattern and casting restrictions. There is great latitude for sizing thickness of sections, location of bearing housings, and size and location of reinforcing ribs. Machine welding and other fabrication shop practices accelerate this method of construction, particularly for the larger sizes.

Casings for parallel axis gears (excluding marine propulsion reduction gears) are usually in two main components which in turn may be subdivided into smaller parts. The two main components, upper and lower, are divided horizontally by bolted flanges at the pitch line of the gears which is also the center line of the shafts (Fig. 13). The lower halves of the bearing shells (for split sleeve bearings) are located in the bottom half of the casings. The bottom flange of the bottom casing is built for bolted attachment to the foundation. That part of the bottom casing between the lower and upper flanges is very rigid because the primary function of the casing is to support the gears and absorb the load they transmit to the bearings, as well as holding the rotating elements in their correct axis of rotation without deflection.

Casings for nonparallel axis gears may be either welded or cast, or a combination of both. Generally the smaller casings are cast and the larger casings are fabricated.

Depending on the type of construction, the upper casing halves may or may not be as rigid as the lower halves. The lower halves of split sleeve bearings rest in the bottom casing and are secured by the upper bearing halves. The upper halves of the bearings in turn may be secured by independent strongbacks which are bolted and doweled to the lower casings. In this type of construction there is no excessive load to be carried by the upper casing; all loads are transferred to the lower casing.

In some types of construction, the upper-half bearing housings are integral with or restrained by the upper casing. In these instances the upper casings must be rigid enough to share the bearing load. In some gears the journals run in the upper halves of the bearings when fully loaded.

Bearing alignment and therefore gear alignment is determined by the accuracy of the machine work in the casing. The center lines of the bearing bores in the casing determine the center lines of the shaft axis. There is no room for error because in most instances there is no means of bearing adjustment within a gear set. The casings must be line bored to extreme accuracy of location, axis direction, and diameter. The usual tolerances for this boring operation are approximately 0.0002 in. to 0.0001 in. (0.00508 mm to 0.00254 mm) depending on size.

Casing joints have to be perfectly parallel and lie in the proper planes to prevent distortion or cracking when the joints are made up. Because of the copious internal oil sprays necessary to lubricate the teeth, the joints also have to be oil tight. This requires a fine face finish and close bolt spacing because the joints are metal-to-metal without gaskets to compensate for imperfections.

Oil seals must be provided for the shafts at the exit points. Metallic oil deflectors are frequently used for this service.

Casings must be designed to be good radiators of heat. Gears operating under load generate great quantities of heat which must be dissipated. This does not present a problem with the larger gear sets because the casing surface areas are inherently large enough to be good radiators. The radiation capacity of more compact units can be improved by including cooling fans attached to the casing and driven directly from a gear shaft

High speed helical and herringbone gear sets are used extensively for increasing speeds from electric motors, turbines, or diesel engines on such applications as centrifugal pipeline pumps, centrifugal gas compressors, and wind tunnel fans. Figure 14 illustrates one type of gear set used in conjunction with a 27,000-hp (36-kW) motor for the main air compressors in an air separation plant serving a steel mill.

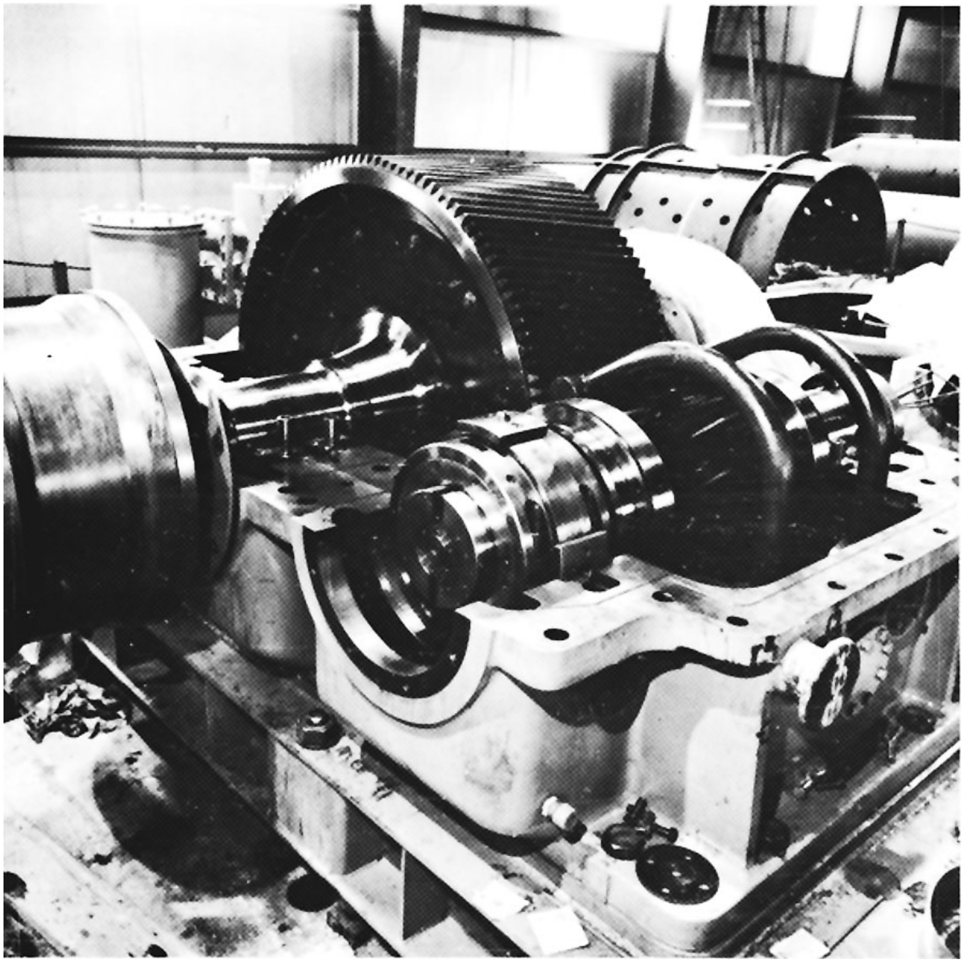

Fig. 14. Gear unit.

Gear sets are rated on (a) mechanical power, based on wear (durability) and strength (load) capacity, and (b) thermal power, that is the average power that can be transmitted continuously without overheating at normal ambient temperature and without special cooling. This rating varies with speed and ratio. American Gear Manufacturers Association ratings are based on a 100°F (38°C) rise above ambient temperature with a maximum operating temperature of 200°F (93°C).

C.1.9 Bearings and Oil Seals

Two different types of bearings are used in gear sets: the sliding bearing or the antifriction bearing.

C.1.9.1 Sliding Bearing

The oldest and most common is the sliding bearing (Fig. 15). Practically all open gearing such as that used to drive paper machine dryer rolls was, until recently, run in sleeve bearings. Antifriction bearings are now being used more for this service as paper machine speeds increase.

Two similar metals in sliding contact with each other are not a good combination as regards durability; therefore, bronze was adapted many years ago as a suitable bearing for iron and steel shafts. Besides the

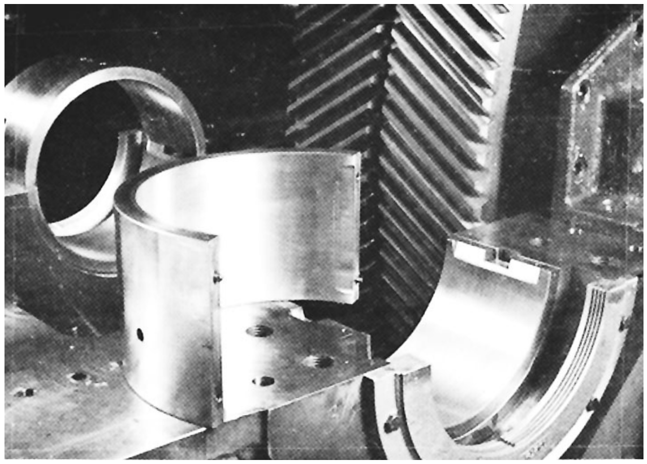

Fig. 15. Straight sleeve bearing.

fact that it has a low coefficient of friction, is a dissimilar metal, and is a very durable metal, bronze can be readily machined and an excellent surface finish produced. Bronze sleeve bearings are still used for relatively low-speed open gearing, but they are not used in high speed enclosed gear sets or standard enclosed speed reducers.

With the exception of a few specialized operations, sleeve bearing lining material is babbitt. Table 3, extracted from ASTM Specifications B23-26, shows white metal alloy Nos. I, II, and III. There are actually 12 specification numbers in the full table ranging from high tin alloy to high lead alloy. Babbitt is the commercial name usually associated with bearing material within the group. Navy Composition W is also shown, but it is not part of the ASTM table.

Table 3. White Metal Alloys.

White Metal, Alloy Number	*Tin, %*	*Antimony, %*	*Lead, max. %*	*Copper, %*	*Iron, max. %*	*Arsenic, max. %*
I	91	4.5	0.35	4.5	0.08	0.10
II	89	7.5	0.35	3.5	0.08	0.10
III	83.33	8.33	0.35	8.33	0.08	0.10
Navy Comp. W	87.5 to 8.5	7.0 to 8.0	0.35	3.5 to 4.5		
Original babbitt	89.3	8.9		1.8		

The advantages of babbitt are: easily cast and machined; low coefficient of friction with the ability to retain a substantial film of oil; and the bearing surface becomes hard enough to resist wear yet the underlying matrix is soft enough to adjust to load conditions. It is long wearing with almost an infinite fatigue life. To a degree, babbitt can absorb hard particles of foreign matter and thus spare the shaft surfaces; it can be dressed by hand if necessary. Most importantly, in operation, minor high spots or imperfections will melt and wipe clear with no further trouble. In a similar manner, it can handle transient thermal surges of a minor nature. Babbitted bearings are easy to repair and repairs can be made very quickly.

The mechanics of how some of these advantages develop is that, several types of metallic crystals are formed in the microstructure, one of which is the primary, relatively hard, antimony-tin type. This is well dispersed throughout the matrix by alloying with the copper. These hard crystals are wear resistant and consequently retain their shape, while the adjacent softer crystals wear very quickly and form an infinite number of natural oil pockets which contribute greatly to the makeup of an oil film. The soft areas also absorb small foreign bodies.

Under a microscope the surface of a babbitted bearing looks anything but smooth; it has many high and low spots even after it has been run in. These high points are called *asperities* and are the load carriers, being the closest points to the shaft but separated from it by the lubricating oil film.

As a structural material, babbitt is very weak and must be supported by a backing of a much stronger material formed to the shape of a sleeve or shell. The sleeves or shells into which the babbitt lining is cast are made of mild steel in most modern designs, but some are made of cast iron and others of nonferrous compositions.

In the manufacturing process this bimetal component is made by casting molten babbitt into the steel (or iron or bronze) shell. The interface of the two metals has to be perfectly homogeneous if a sound metallurgical bond is to be reached. This was difficult in the past, and mechanical locks were employed as a means of more firmly anchoring the babbitt in the liner. Most locks were formed by cutting dovetail grooves into the shell, allowing the molten babbitt to flow into the grooves where it held tightly upon solidification.

Most modern bearings are babbitted by a centrifugal casting process wherein the molten babbitt is introduced while the shell is rotated on its horizontal axis at very high speeds. Centrifugal force pushes the liquid babbitt tightly against the liner forming a good dense babbitt layer free of voids. The success of this process is dependent on thorough preparation to produce very clean liner surfaces.

Some bearings are still poured by hand, especially in repair work. When properly done with clean warm shells, fluxed and tinned, and with correct babbitt pouring temperature, results can be excellent. After thorough cleaning, the bearing halves are placed together with the axis vertical around a central mandrel (dummy shaft) dammed up against leakage with fire clay. The shells and mandrel are heated, and the babbitt poured. The inside diameter after pouring must allow enough excess material for boring out in the machine shop. If the shells are of iron, they must be degraphitized; otherwise they cannot be tinned properly.

Tin-base babbitt is poured at 880°F (471°C) but should be approximately 20° higher if it is to be carried more than a few steps to the bearing. Temperature control is important. If instruments are not available, one traditional indicator is still dependable; the babbitt is ready to pour when it will quickly ignite a clean dry pine stick dipped into the pot. High lead babbitt is ready to pour when it chars but does not ignite the stick.

Babbitt thickness is the main determinant in the capacity of babbitted bearings. The earliest bearings had thick linings and were not highly resistant to fatigue (cyclic loading); approximately 1000 psi (6894 kPa) (69 bar) at steady load was the limit. As bearing technology improved, babbitt thickness was reduced and today's light babbitt liner on steel or bronze backing is good for approximately 5000 psi (34.5 MPA) (345 bar) load and 4000 ft/min (1220 m/min) journal speed. Some thin babbitt linings are down to 0.004 in. (0.0122 mm).

Good quality, high speed bearings are vital to the smooth operation of a gear, because a worn bearing shifts the distribution of load on the gear teeth and causes undue wear.

In large double helical drives, both gear and pinion are supported by heavy duty, horizontally-split babbitted sleeve bearings. Although there is no thrust due to load, the ends of the gear bearings are babbitted to act as thrust bearings. This limits axial movement of the gear, while the pinion is free to move and position itself with the gear to equalize tooth load.

C.1.9.2 Antifriction Bearing

Antifriction bearings (Fig. 16) offer several advantages, the most important being that they automatically set and maintain precisely correct center distances between gear rotors. This is due to the very small clearances in the bearings. Bearing efficiency is very high. They are available in widely standardized ranges as complete individual bearing units with known load characteristics available from catalogue data. Antifriction bearings have low starting friction and can be run at high speeds.

Some disadvantages are high first cost and lower shock resistance than many sleeve bearings. Quick or temporary repairs are impossible, in most cases repairs would cost more than new bearings. Antifriction bearings cannot tolerate impurities such as grit or dirt. They must be handled with extreme care and only by experienced personnel. They have a finite fatigue life because the loading contact with the races is cyclic which is conducive to fatigue.

The balls in a ball bearing make only point contact between the inner and outer rings, frequently referred to as the inner and outer races. The rollers in the roller bearing make line contact between the inner and outer rings.

Generally, the inner ring is tightly fitted to the journal of the rotating shaft and rotates with the shaft This requires perfect interference fit onto the journal to avoid distortion of the inner ring. The outer ring is fitted in the bearing housing of the gear case and generally remains stationary with allowance for drift in some cases. The journal and housing fits are critical and can be determined only by the bearing manufacturer.

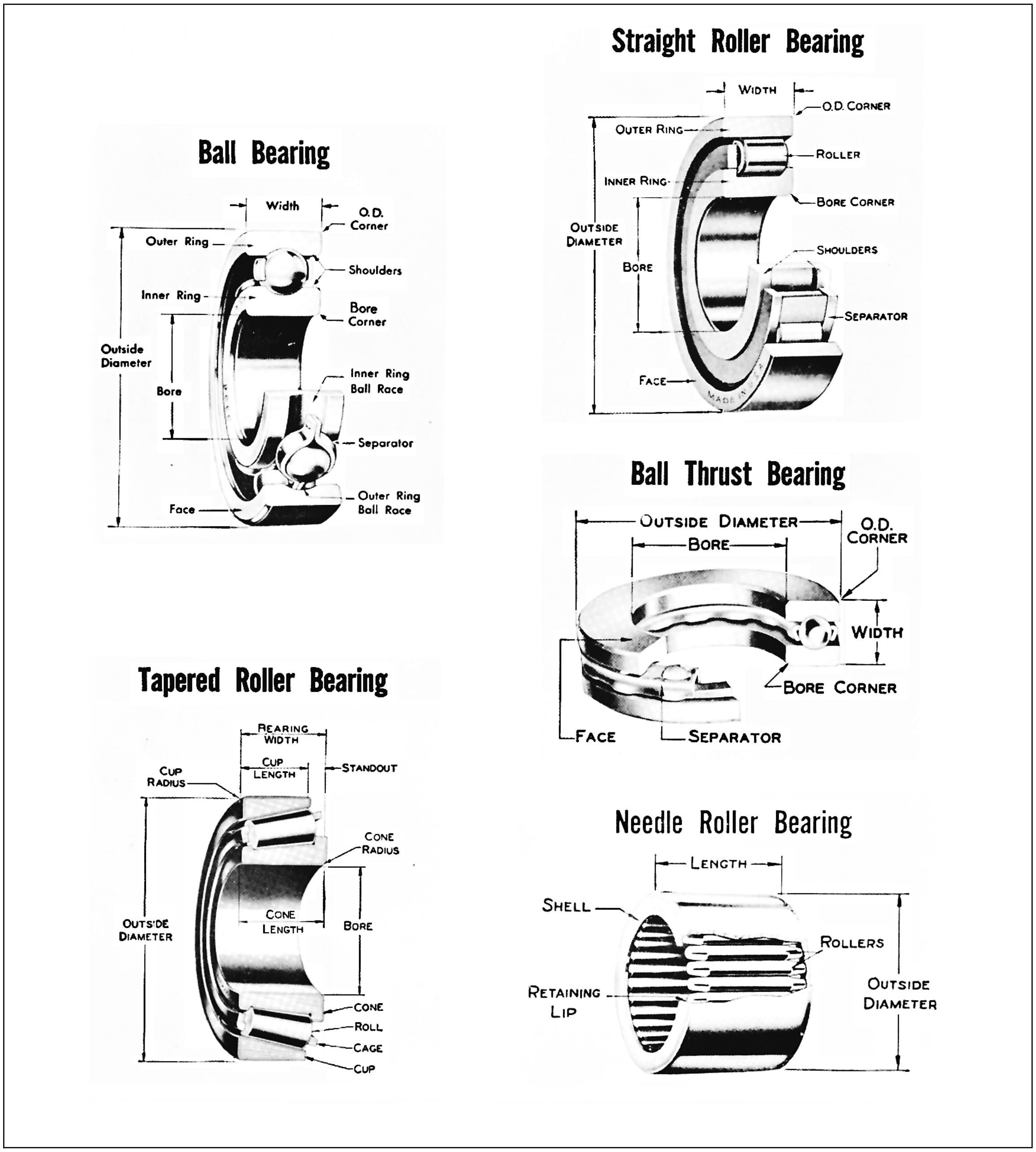

Fig. 16. Bearing parts and their names. (Anti-friction Bearing Manufacturers Assoc. Inc.)

These minimum contact areas between the rings and rolling elements are conducive to high unit stresses in the bearing components. The high stresses dictate that the bearing parts be hard and have great strength in compression to retain their designed geometry under load. At the same time they must be capable of distortion under load but within the elastic limit. The contact area of a single ball under load is in the shape of an ellipse instead of a single round point, while the contact area of a roller bearing is a small rectangle rather than a narrow straight line.

Gear teeth and the balls or rollers in an antifriction bearing are loaded in a similar sequence. As a ball or roller approaches the point of maximum loading, it gradually assumes some of the load up to the maximum and then gradually unloads as it rotates out of the loaded area. Loading and stress application is cyclic as it is in a gear. For this type of loading, extreme accuracy and extremely smooth surface finish is necessary to resist fatigue.

Because of these requirements, antifriction bearing material must be homogeneous, free of slag and stringers, and hardenable. High carbon steel, carburizing low carbon steel, or special alloys are used because they are hardenable and tough. Raw materials are in the form of seamless high carbon steel tubing, forgings, wire stock and special alloys.

C.1.9.3 Seals

Metallic oil deflectors are used with sleeve bearings. These deflectors are carefully machined and located for positive shaft clearance at all times. They are attached to the casings, run as close to the shaft as possible, and operate on the principle of throttling down pressure in each annular chamber. These deflectors may be made of material such as steel, brass, bronze or aluminum. They are split along the same plane as the casing. The insides which run against the shaft are ''knife edged'' so that if there is an accidental rub, it will quickly rub the sharp edge clear of the shaft, with minimum damage.

Seals used with antifriction bearings are frequently a part of the bearing and as such may be proprietary items. Antifriction bearing seals are of all types from plain felt seals and leather cup seals through labyrinths, rings, and slingers.

C.2 Service Factors

Gears in actual service, like other machine elements, are subject to a variety of operating conditions. Rather than conducting an individual analysis in every situation, a practical alternative is to introduce an experience factor. This factor, known as the *service factor*, is an indication of the gear rating or mechanical horsepower rating (MHP).

In order to safeguard the gear teeth from breakage, the maximum load rating must be greater than, or equal to, the known dynamic load multiplied by the service factor. The base service factor is 1.00 and other standard factors are 1.25, 1.50, 1.75, 2.00.

There are three considerations that determine the proper service factor.

1. Type of driving unit. Electric motors run smooth and engines pulsate. Engine-driven gears require a higher service factor because of the increased stresses on the gear teeth.

2. Type of driven unit. Shock loading and load peaks require an increased service factor.

3. Duty cycle. The number of hours of operation also affect service factor. Standard available duty cycles are ½, 3, 8 to 10 and 24 hours per day.

Table 4 shows the equivalent gear unit service factors which correspond to the AGMA gearmotor classifications.

Table 5 lists recommended service factors for various loads and duty cycles. For details of application classification, refer to AGMA Standard 151-02.

Table 4. Equivalent Service Factor of AGMA Classifications.

AGMA Gearmotor Class	*Equivalent Service Factor*
I	1.00
II	1.41
III	2.00

C.3 Horsepower

Two formulas are given for calculating horsepower. One indicates the horsepower with reference to tooth strength; the other, the horsepower as limited by tooth wear. The speed factor is to be based on revolutions per minute. The power capacity of both gear and pinion should be checked for tooth wear and tooth strength. The smallest of the power ratings thus obtained should be used.

$$\text{Horsepower for Wear} = \frac{(S_C)\ (X_C)\ (Z)\ (F)\ (N)\ (T)}{126{,}000\ (K)\ (P)}$$

$$\text{Horsepower for Strength} = \frac{(S_b\ (X_b)\ (Y)\ (F)\ (N)\ (T)}{126{,}000\ P^2}$$

Where: S_c = Surface stress factor (Table 6)
X_c = Speed factor for wear (Table 7)
Z = Zone factor (Fig. 17)
F = Face width, inches
N = Revolutions per minute
T = Number of teeth
K = Pitch factor – $P^{0.8}$ (Table 9)
P = Diametral pitch
S_b = Bending stress (Table 6)
X_b = Speed factor for strength (Table 8)
Y = Strength factor (Fig. 18)

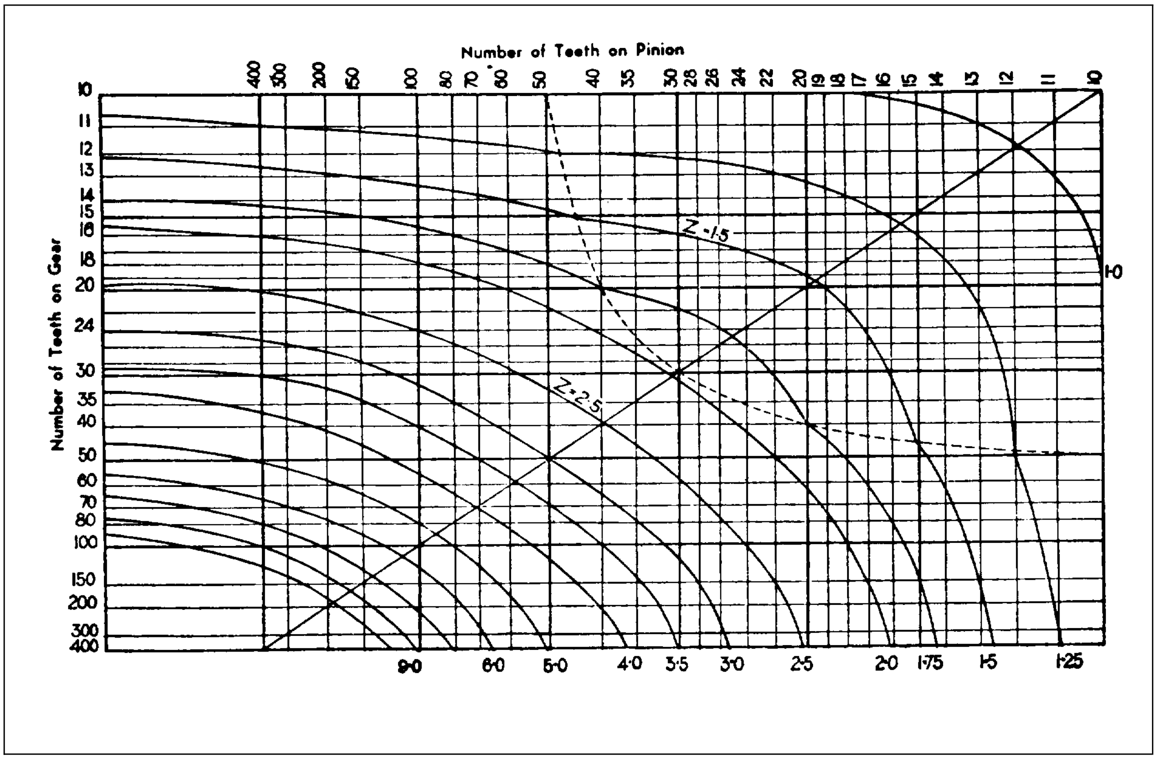

Fig. 17. Zone Factor Z for Spur Gears; 20-Degree Pressure Angle.

Example: Find the allowable horsepower for the following spur gears. The pinion and gear speeds are 500 and 100 revolutions per minute; continuous operation 12 hours per day; diametral pitch — 3; face width — 4 inches; pressure angle 20 degrees; pinion 20 teeth; pinion material 0.40% carbon steel normalized; gear 100 teeth; gear material, cast iron of ordinary grade.

$$\text{Pinion hp for wear} = \frac{(1600)(0.305)(2.20)(4)(500)(20)}{(126{,}000)(2.40)(3)} = 47\ (35\ \text{kW})$$

$$\text{Gear hp for wear} = \frac{(1000)(0.410)(2.20)(4)(100)(100)}{(126{,}000)(2.40)(3)} = 40\ (30\ \text{kW})$$

$$\text{Pinion hp for strength} = \frac{(19{,}000)(0.305)(0.72)(4)(500)(20)}{(126{,}000)(3)(3)} = 147\ (110\ \text{kW})$$

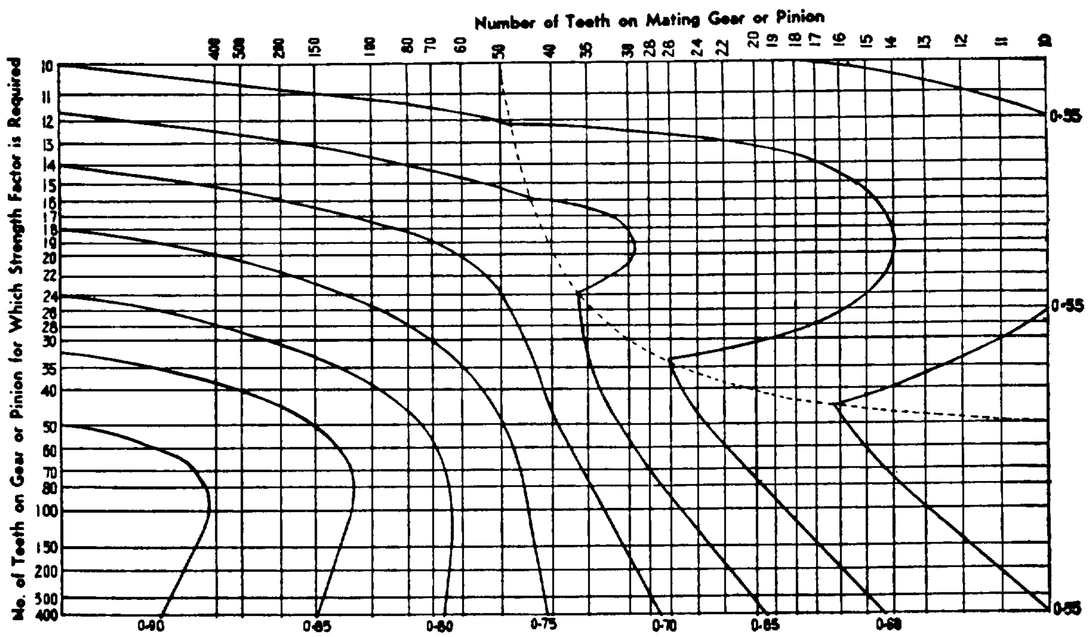

Fig. 18. Strength Factor Y for Spur Gears; also Helical Gears of 30 Degree Helix Angle; 20 Degrees Normal Pressure Angle.

$$\text{Gear hp for strength} = \frac{(5800)(0.410)(0.61)(4)(100)(100)}{(126{,}000)(3)(3)} = 51 \text{ (38 kW)}$$

In this case, the smallest horsepower rating 40 is used. Conversion to metric: hp × 0.7457 = kW

Table 5. Recommended Service Factors for Character of Load on Driven Machine.

	Electric Motor			*Multicylinder Gas or Steam Engine or Turbine*			*Single-Cylinder Gas Engine*		
	Intermit-tent 3 hr per day	*8-10 hr per day*	*24 hr per day*	*Intermit-tent 3 hr per day*	*8-10 hr per day*	*24 hr per day*	*Intermit-tent 3 hr per day*	*8-10 hr per day*	*24 hr per day*
Uniform	0.8	1. 0	1.25	1. 0	1.25	1.25	1.25	1.50	1.75
Moderate Shock	1.0	1.25	1.50	1.25	1.50	1.75	1.50	1.75	1.00
Heavy Shock	1.5	1.75	2.00	1.75	2.00	2.25	2.00	2.25	2.50

C.4 Gear Materials

Many kinds of materials are used for gears. The cheapest is ordinary gray iron, for example ASTM 20, which is relatively good from a standpoint of wear. ASTM Classes 30 and 40 are frequently used; and the high strength irons, Class 60 or higher, are also appropriate for gears with the proper heat treatment.

Untreated wrought-steel gears are inexpensive and generally have a carbon content of 0.30-0.50% with low wear capacity. Cast steel should be annealed to avoid excessive distortion. Both the wrought and cast steels may be heat treated for improved mechanical properties. Oil quenching is not so drastic as water quenching and is generally used because the consequent distortion is minimized. Also, because of low distortion, alloy steels are favored when teeth are to be heat treated. Blanks, however, can be heat treated before the teeth are cut. Ordinarily, the maximum hardness for machining is about 250 BHN, but successful machining is often done on alloy steels of 350 BHN.

When gears are to be through hardened, the carbon content should be 0.35% to about 0.6%. Frequently used steels in order of approximate cost, are 1335, 5140, 4037, 4140, 8640, 8740, 3135. Hardness depends on the carbon content, hardenability, and heat treatment.

Table 6. Basic Surface and Bending-stress Factor of Spur and Helical Gears.
Factors for use in British Standard Horsepower Formula

Type of Material (Numbers in Parentheses Indicate Footnotes)		*Minimum Tensile Strength Tons per Sq. In.*	*Minimum Brinell Hardness Number*	*Surface Stress Factor Sc*	*Bending Stress Lb. per Sq. In. Sb*
Fabric...		—	—	560	4,500
Cast Iron, Ordinary Grade..................................		12	165	1,000	5,800
“ ” , Medium Grade		16	210	1,350	7,600
“ ” , High Grade, as Cast		22	220	1,450	10,400
Castings, Malleable		20	140	850	11,000
Phosphor Bronze, Sand Cast.............................		12	69	700	7,000
“ ” , Chill Cast...		15	82	850	8,500
“ ” , Centrifugally Cast..		17	90	1,000	10,000
Cast Steel, 0.35% to 0.45% Carbon		35	145	1,400	19,000
“ ” , 0.50% to 0.55% Carbon	(1)	38	160 (3)	3,100	13,000
Forged Carbon Steel, 0.15% Carbon	(2)	32	140 (4)	9,000	28,000
“ ” “ , 0.40% Carbon	(6)	35	145	1,400	17,000
” “ ” , 0.40% Carbon	(7)	35	145	1,600	19,000
“ ” “ , 0.40% Carbon	(1)	35	145 (8)	2,800	12,000
” “ ” , 0.40% Carbon	(9)	40	175	1,800	20,000
“ ” “ , 0.40% Carbon	(10)	40	175	2,000	22,000
” “ ” , 0.55% Carbon	(6)	45	200	2,000	21,600
“ ” “ , 0.55% Carbon	(7)	45	200	2,300	24,000
” “ ” , 0.55% Carbon	(1)	45	200 (11)	4,000	15,000
Nickel Steel, 1% nickel	(12)	40	175	2,000	22,000
“ ” , 3% nickel	(12)	45	200	2,300	24,000
“ ” , 3% nickel	(2)	45	200 (13)	10,200	40,000
“ ” , 3½% nickel	(1)	55	250 (14)	5,100	18,500
“ ” , 3½% nickel	(12)	55	250	3,000	30,000
“ ” , 3½% nickel	(2)	45	200 (15)	10,200	40,000
“ ” , 5% nickel	(2)	55	250 (16)	11,200	47,000
Nickel-chromium, 1½% Ni, 1 Cr	(17)	55	250	3,000	30,000
“ ” , 1½% Ni, 1 Cr	(1)	55	250 (13)	5,100	18,500
“ ” , 1½%	(17)	100	440	5,500	40,000
“ ” , 3½%	(17)	55	250	3,000	30,000
“ ” , 3½%	(1)	55	250 (14)	5,100	18,500
“ ” , 3½%	(2)	55	250 (16)	11,200	47,000
Carbon-chromium, 0.55% carbon	(18)	55	250	3,000	30,000
“ ” , 0.55% carbon	(18)	65	290	3,500	36,000
“ ” , 0.55% carbon	(1)	55	250 (14)	5,100	18,500

(1) Surface Hardened;
(2) Casehardened;
(3) Core, I60; case, 530;
(4) Hardness of core;
(5) Core, I40; case, 640;
(6) normalized; for sections thicker than 5 inches;
(7) normalized; for sections less than 5 inches thick;
(8) Core, I45; case, 460;
(9) Heattreated; for sections thicker than 5 inches;
(10) Heat-treated; for sections less 5 inches thick;
(11) Core, 200; case, 520;
(12) Heat-treated;
(13) Core;
(14) Core, 250; case, 500;
(15) Core, 200; case, 620;
(16) Core, 250; case, 600;
(17) Oil hardened and tempered to strength given in second col.;
(18) Heat-treated to strength given in second column.

Table 7. Speed Factors X_c for Wear.

Rev. per Minute	Running Time – Hours per Day							
	1	2	4	6	8	12	18	24
	Speed Factors X_c for Wear							
100	0.935	0.735	0.585	0.515	0.470	0.410	0.350	0.320
150	0.865	0.685	0.540	0.475	0.435	0.370	0.330	0.300
200	0.825	0.650	0.520	0.460	0.415	0.360	0.310	0.280
300	0.775	0.615	0.485	0.425	0.380	0.330	0.290	0.270
400	0.730	0.580	0.460	0.400	0.360	0.320	0.270	0.250
500	0.700	0.550	0.440	0.380	0.350	0.305	0.260	0.240
600	0.680	0.530	0.425	0.370	0.340	0.290	0.250	0.230
800	0.635	0.500	0.400	0.350	0.320	0.270	0.240	0.220
1,000	0.610	0.480	0.380	0.335	0.305	0.260	0.230	0.210
1,500	0.550	0.440	0.345	0.310	0.275	0.240	0.210	0.190
2,000	0.520	0.415	0.325	0.290	0.260	0.220	0.200	0.180
2,500	0.480	0.380	0.305	0.265	0.240	0.210	0.185	0.165
3,000	0.450	0.355	0.280	0.250	0.225	0.195	0.170	0.155
4,000	0.415	0.325	0.260	0.225	0.207	0.180	0.155	0.145
5,000	0.380	0.305	0.240	0.210	0.190	0.165	0.145	0.132
6,000	0.355	0.285	0.225	0.200	0.180	0.155	0.135	0.125
7,000	0.340	0.270	0.215	0.190	0.170	0.150	0.130	0.118
8,000	0.325	0.260	0.205	0.180	0.165	0.142	0.125	0.113
9,000	0.315	0.250	0.200	0.175	0.157	0.135	0.120	0.108
10,000	0.305	0.240	0.190	0.165	0.152	0.130	0.115	0.105

Table 8. Speed Factors X_b for Strength.

Running Time, Hours per Day	Revolutions per Minute									
	100	150	200	300	400	500	600	800	1000	1500
	Speed Factors X_b for Strength									
1	0.600	0.550	0.525	0.445	0.435	0.420	0.415	0.410	0.385	0.350
3	0.510	0.435	0.425	0.410	0.400	0.380	0.370	0.345	0.330	0.300
6	0.430	0.415	0.405	0.380	0.360	0.345	0.330	0.310	0.295	0.275
12	0.410	0.380	0.360	0.340	0.320	0.310	0.300	0.285	0.270	0.245
24	0.375	0.350	0.330	0.310	0.295	0.285	0.275	0.255	0.245	0.225
Running Time, Hours per Day	Revolutions per Minute									
	2000	2500	3000	4000	5000	6000	7000	8000	9000	10,000
	Speed Factors X_b for Strength									
1	0.325	0.305	0.285	0.260	0.240	0.225	0.215	0.208	0.200	0.192
3	0.285	0.260	0.245	0.225	0.208	0.195	0.185	0.178	0.170	0.165
6	0.255	0.235	0.220	0.200	0.185	0.175	0.165	0.160	0.153	0.148
12	0.230	0.215	0.200	0.182	0.168	0.158	0.150	0.145	0.140	0.135
24	0.210	0.195	0.180	0.165	0.152	0.143	0.138	0.130	0.126	0.120

Table 9. Pitch Factors K.

Diametral Pitch	Factor K	Diametral Pitch	Factor K	Diametral Pitch	Factor K	Diametral Pitch	Factor K
1	1.00	2¼	1.90	4	3.05	9	5.80
1¼	1.20	2½	2.10	5	3.65	10	6.40
1½	1.40	2¾	2.25	6	4.25	12	7.40
1¾	1.55	3	2.40	7	4.80	14	8.30
2	1.75	3½	2.70	8	5.40	16	9.25

C.5 Fabrication

C.5.1 Hardening

The key to gear strength lies in providing extra strength in the carburized and hardened case at the positions of maximum stress. Fig. 19 illustrates a plastic model of a gear tooth undergoing simulated service showing that the points of maximum stress occur where the teeth meet, and in the fillets. To withstand these stresses, cases of carburized gears should be deep — about 1/6 to 1/5 of the tooth width at the pitch line.

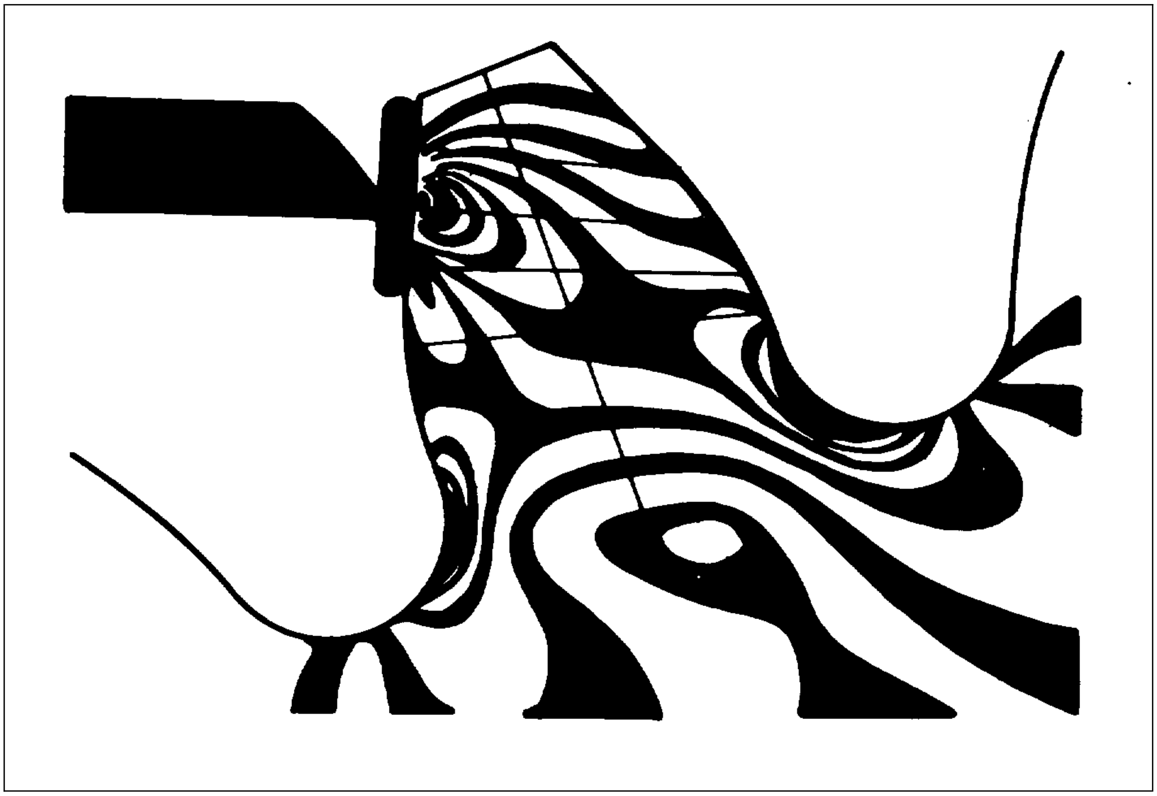

Fig. 19. Points of maximum stress.

This hardening of the surface regions can be effected by making the whole part first of low-carbon steel, which is tough and then ‘‘carburizing’’ the surface region. This is done by exposing the piece to the action of an agent which introduces carbon into the surface, heating it in such an agent at an elevated temperature for a number of hours, whereupon the piece is hardened by quenching in the usual manner.

C.5.1.2 Carburizing

When a piece of low-carbon steel is packed in a material consisting largely of carbon, as for example charcoal, and is heated in this material to an elevated temperature such as 1700°F (925°C), carbon will penetrate from the packing material into the surface regions of the steel. (That is, A low carbon steel which contained originally 0.15% carbon will eventually contain in its surface layers up to 1.20% carbon.) Therefore, by carburizing, a gear now has 1.20% carbon at the surface while remaining at 0.15% carbon in the interior. If this piece is hardened, the gear has an extremely hard surface with a tough core.

Hardening is effected by quenching, water quench for plain carbon and oil quench for alloy. Tempering (drawing) is done only at low temperatures.

C.5.1.3 Quenching

A carburized piece quenched from slightly above 1390°F (755°C) will be fully hardened. One practice consists of allowing the carburized piece to slow cool completely from the carburizing temperature, then reheating it to perhaps 1450°F (785°C) and quenching it. However, when reheated to only this temperature, there is still a substantial amount of undissolved free cementite, if present in the form of large plates, the piece is more brittle than if the cementite is all dissolved or is present in globular form.

This possible shortcoming due to the carbide is responsible for two alternative practices that may be followed. The piece may be quickly removed from the carburizing box at the end of the carburizing period, and quenched at once from this temperature, followed by the low temperature draw only. Or, it may be quenched from the carburizing temperature, 1700°F (925°C) and then reheated to say 1450°F (785°C) and again quenched, and finally tempered. Both of these latter two methods, termed respectively ‘‘Quench from the Box’’ and ‘‘Double Quench’’, avoid the presnce of grain-boundary cementite. In the case of plain carbon steels given a double quench, the first quench (high temperature) is commonly done in oil to avoid distortion and strains, and the second quench in water, to attain the required hardness.

The first method (a single quench after an air cool and reheating) may be effected if grain-boundary carbide is not objectionable, or if the steel is of such nature that the free cementite formed globules in cooling. The quench from the box may be employed if the steel can be quenched from the high temperatures to final hardness without harm. The double quench (undesirable only because of cost) is employed where other conditions make it advisable, such as the nature of the part, the kind of steel, or special properties desired.

Torch hardening, also called flame hardening, involves heating by means of a torch and immediately cooling with water. This method is particularly adaptable to the teeth on large gears, where it would be cumbersome to heat the whole piece for quenching. It is also advantageous in avoiding distortion due to quenching, because only the surface is heated by the flame, while the body of the gear remains relatively cool. Heat is applied rapidly by means of an oxyacetylene torch. The method has the unique, though not necessarily advantageous, feature that only a small area is hardened at a time. The torch moves slowly along the surface which is to be hardened, and so heats it to the quenching temperature, and is followed immediately by a stream of water which quenches the heated area.

Induction hardening, an electrical method in which somewhat similar metallurgical principles are involved, heats the surfaces by high-frequency electrical induction. This process applies a copper yoke or other shape of inductor to the region whose surface portion is to be hardened. Through this yoke flows a current of high frequency (range 2000 to 100,000 cycles) thereby heating by induction the surface layer which is to be hardened. The heating time is extremely fast, being completed in a matter of 2 to 3 seconds, thus incidentally providing an interesting sidelight on the speed with which carbide may adequately go into solution. The current is then shut off, and the part immediately flushed with a powerful water spray.

A relatively modern development in surface hardening is the process of nitrogenizing, or nitriding by ammonia. Steels of certain compositions, when heated in a stream of ammonia gas at approximately 900°F (480°C), acquire an extremely hard case. Steels of special composition have been developed which are particularly suitable for nitriding. The most effective alloying element is aluminum, which is added to the steel in amounts approaching 1.0%. Chromium is also effective, as is also molybdenum. Other elements also contribute to nitrogen hardening, though less markedly, for example, silicon and tungsten. In the common nitriding steels, the elements used are aluminum, chromium, and molybdenum.

A great advantage of the nitriding process is that the hardening operation consists merely in heating in ammonia gas at 900°F (480°C) and cooling slowly in the furnace, without subsequent heat treatment. The scaling common to ordinary treatment is thus avoided, so that parts may be finished to size and then nitrided as the final operation. Further, due to the low temperature and slow subsequent cooling, distortion is extremely slight. Finally, the hardness obtainable exceeds anything possible by any other known treatment of steel. Hardnesses are commonly in the range 900 to 1100 Brinell, whereas the hardness obtained by ordinary treatment is approximately 750 Brinell. The high hardness of nitrided steel is believed to be due to the presence of fine, well-dispersed particles of aluminum nitride (or other alloy nitride) of submicroscopic size. There is much experimental evidence to support this belief.

This process is costly and involves not only the cost of the steel but also the lengthy time of nitriding and precision in the treatment.

C.5.2 Grinding and Polishing

For precision gears, tooth profiles may be refinished particularly after hardening, by special grinding or lapping machines, or by shaving. In the lapping process the rotating elements are mounted in a lapping stand and run together at low speed in a viscous abrasive compound, resulting in a highly polished wearing surface. The gear shaving machine runs the cutter and work gear together under pressure with their axes slightly crossed, similar to the arrangement for lapping. The hardened steel cutter is provided with multiple radial slots to form cutting edges on the tooth face. These have a side-sliding motion on the gear tooth to remove thin chips. This machine is so accurate and the cut is so light that minute high spots are cut off, producing a super finish.

C.5.3 Welding Techniques

Large gear blanks are mainly fabricated by welding precut and machined components followed by stress relieving at completion. The gear blanks can be either closed type or open type. (See Fig. 20 and 21.)

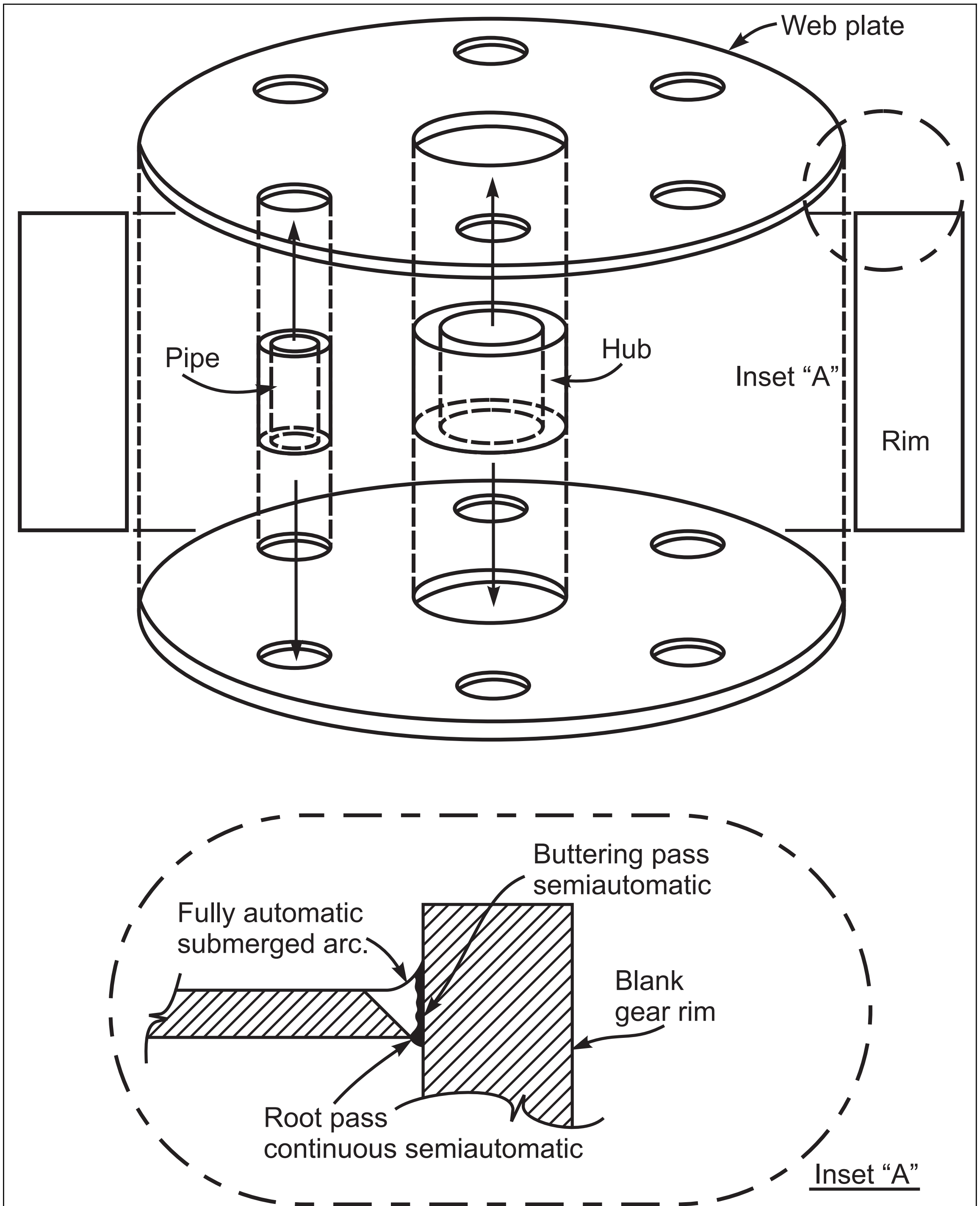

Fig. 20. Closed type gear blank.

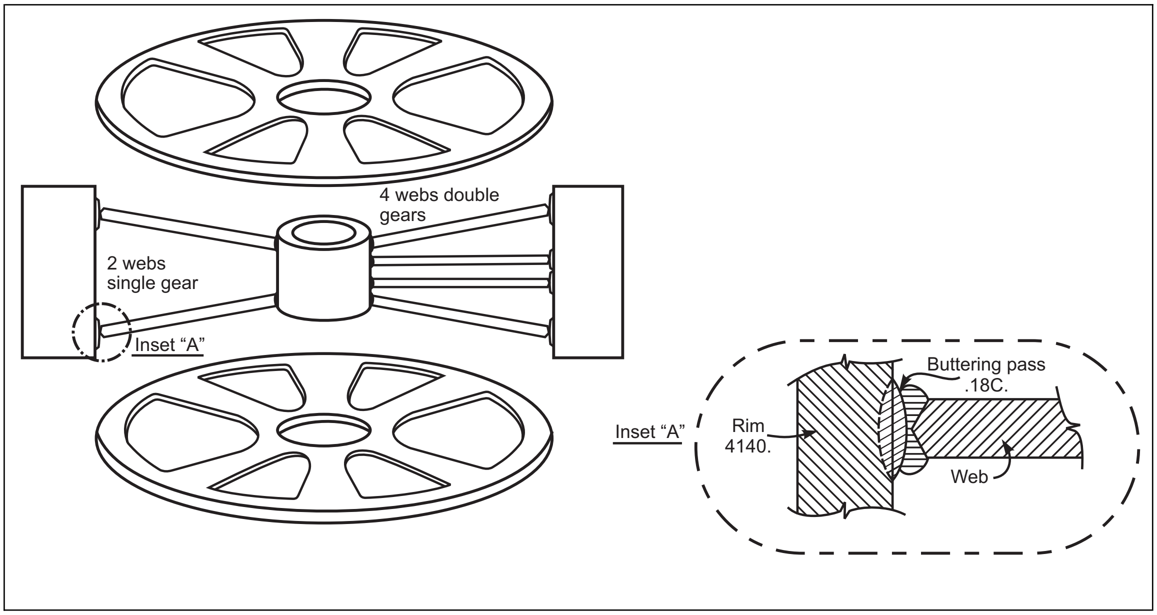

Fig. 21. Open type gear blank.

The main components are the webs, very accurately flame-cut from the solid carbon steel plate; the pipes, cut from A.53 pipe; the hub, of machined carbon steel; and the outer ring (rim). This outer ring starts with the blooming of an ingot and the cutting of it to size to assure accurate input weight. After piercing into a doughnut-shaped preform, it is ready for ring-mill expansion as a seamless rolled ring forging. It has a simple rectangular cross section and is rolled to the required dimension on a ring mill (Fig. 22).

The basic alloy is in the chrome-moly group. Heat treating is required to assure proper thorough hardening for good gear rim homogeneity and machinability. Hard spots could cause tooth cutting difficulties. The forger machines the ring to ⅛ in. (3.2 mm) tolerance.

The ring (rim) is subjected to an ultrasonic check before it is welded onto the gear blank. Structural integrity and brinnel hardness of the rim is maintained throughout the fabrication process.

After welding, the assembly is further heat-treated to relieve stress, approximately 1 hour per inch of rim thickness at 1100°F (593°C) or 50 to 100 degrees below the tempering temperature of the rim. It is then faced on a rotating turntable to assure pitch circle trueness to 0.001 in. (0.025 mm). Finally the teeth are cut to a tooth-to-tooth tolerance of 0.0002 in. (0.005 mm), and a contact matching is made of the gear and pinion to verify uniform tooth-load distribution for proper wear life. This process consumes considerable time and requires continuous quality control surveillance.

C.6 Backlash In Gears

In general, backlash in gears refers to play between mating teeth (Fig. 23). For purpose of measurement and calculation, backlash is defined as the amount by which a tooth space exceeds the thickness of an engaging tooth. Unless otherwise specified, numerical values of backlash are understood to be given on the pitch circles.

The general purpose of backlash is to prevent gears from jamming together and making contact on both sides of their teeth simultaneously. Lack of backlash may cause noise, overloading, overheating, and even seizing and failure of the gears and bearings.

Excessive backlash is objectionable, particularly if the drive is frequently reversing, or if there is a running load as in cam drives.

In setting up proper backlash amounts and tolerances for a pair of gears, many factors need to be considered. The most important factor is probably the maximum amount of runout expected in both gear and pinion. Next are the errors in profile, pitch, tooth thickness, and helix angle. Other important considerations are speed, space for lubricating film, and room for trapped oil to escape as the teeth pass through mesh.

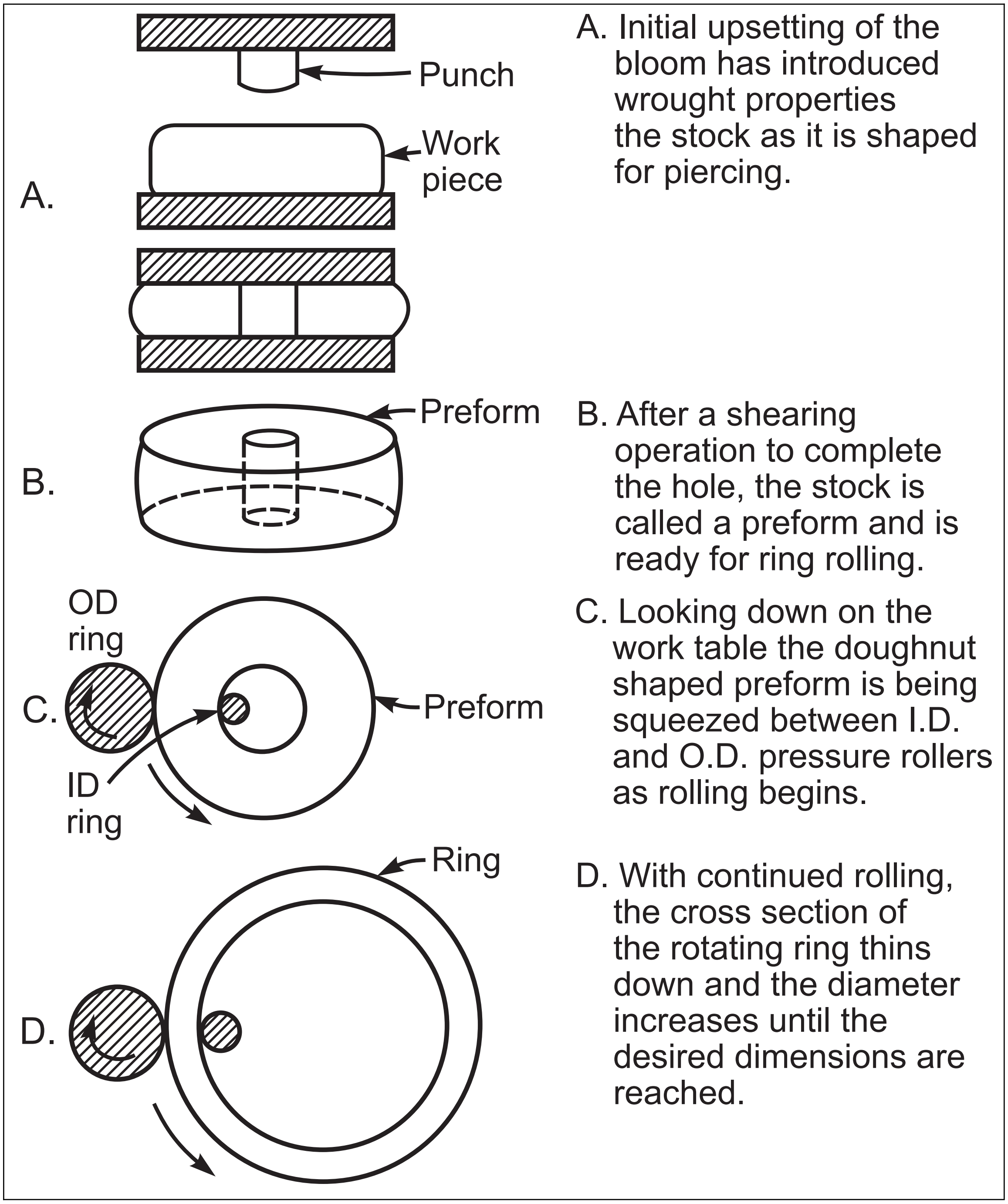

Fig. 22. The basic ring rolling process.

Slow-moving gears usually require the least backlash, although the lubricant may be a heavy oil or grease.

Fast-moving gears are usually lubricated with relatively light oil, but if there is insufficient clearance for an oil film, and particularly for oil trapped at the root of the teeth to escape, heat and excessive tooth loading will occur. At any given speed trapping of the oil is less of a problem with helical gears than with widefaced spur gears, because the gradual meshing of the teeth gives the oil an opportunity to escape. When helical gears are operated at extremely high speeds, the time for oil to escape may be very short, requiring careful regulation of oil directed into the teeth.

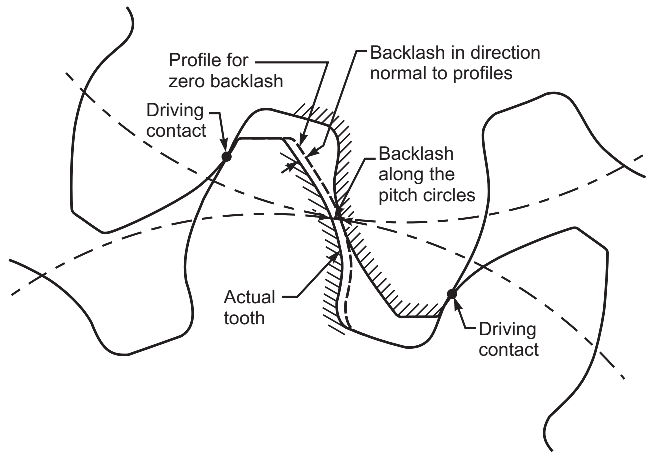

Fig. 23. Backlash between tooth pitches.

Heat is a factor because gears can operate warmer than their housings and therefore expand more. This heat may result from oil churning, or from frictional losses between the teeth, bearings, and oil seals, or from external causes (as from a driving turbine). Also, the material of the gear (for example, bronze) may expand more than the housing material for the same temperature rise.

Backlash is usually measured by holding one gear of a pair stationary, and rocking the other back and forth. The movement is registered by a dial indicator having its pointer or finger in a plane of rotation at or near the pitch diameter and in a direction parallel to a tangent to the pitch circle of the moving gear.

In spur gears, parallel helical gears, and bevel gears, it is immaterial whether the pinion or gear is held stationary for the test. In crossed helical and hyphoid gears, readings may vary according to which member is stationary; hence, it is customary to hold the pinion stationary and measure on the gear.

In some instances backlash is measured by thickness gages or feelers. A similar method utilizes lead wire inserted between the teeth as they pass through mesh, and final thickness of the wire measured with a micrometer.

Table 10 lists acceptable amounts of backlash, assuming average conditions, for general purpose gearing.

C.7 Lubrication

C.7.1 Function of Gear Lubricants

The correct lubricant properly applied will reduce friction, heat, wear, power loss, and noise. In gear lubrication, the contours of the teeth, the load on the teeth, and the speed of operation directly affect the formation of the oil film.

During the contact period of the gear teeth, considerable pressure is produced in a relatively small area, which the lubricating film must successfully withstand. Maintenance of the film and the efficiency of gear lubrication depend on the correct selection and application of the lubricant, adequate supply, proper alignment of components, and suitable operating conditions.

C.7.2 Selection of Gear Lubricants

When speeds are high, the time of contact is very short and the loads are usually high. A lubricating oil of comparatively light body can be used under such conditions. In case of low speeds and heavy load, when the time of contact is longer, a heavier oil should be used. Such oils can be retained more easily by oil seals, have less tendency to leak, cause less consumption, and reduce noise. But even under normal load conditions, the requirements of gear lubrication are exacting and demand a high grade of oil for efficient service.

Table 10. Suggested Backlash for the Following Gears When Assembled: Spur Gears Parallel and Crossed Helical Gears; Double-Helical or Herringbone Gears; Straight, Sprial, and Zerol Bevel.

		Backlash	
Diametrical		Inches	Millimeters
1		0.025-0.040	0.635-1.016
1½		0.018-0.027	0.457-0.686
2		0.014-0.010	0.356-0.508
2½		0.011-0.016	0.279-0.406
3		0.009-0.014	0.228-0.356
4		0.007-0.011	0.178-0.279
5		0.006-0.009	0.152-0.228
6		0.005-0.008	0.127-0.2
7		0.004-0.007	0.1-0.18
8 and 9		0.004-0.006	0.1-0.15
10 to 13		0.003-0.005	0.07-0.13
14 to 32		0.002-0.004	0.05-0.1
Circular pitch			
Inches	Millimeters		
4	102	0.032-0.050	0.8-1.3
3	76	0.024-0.038	0.6-0.9
2	51	0.017-0.025	0.4-0.6
1½	38	0.013-0.019	0.3-0.5
1	25	0.009-0.014	0.2-0.4
¾	19	0.007-0.011	0.17-0.3
½	13	0.005-0.007	0.13-0.17
¼	6	0.003-0.005	0.07-0.13
⅛	3	0.002-0.004	0.05-0.1

Room temperature or the temperature of the atmosphere surrounding the gear will also influence the selection of the oil. A gear drive operating in an ambient temperature of 0°F to 70°F (-18°C to 22°C) should be lubricated with an oil of lower viscosity and lower pour point than when the surrounding temperature varies between 70°F and 120°F (22°F and 49°C).

Gear units working under varying load conditions require an oil which will maintain its body and film strength at the higher temperatures caused by heavy loads and wil not be viscous under lighter loads.

The American Gear Manufacturers Association has established a recommended practice for industrial gear lubrication. Oils for gear lubrication, especially on enclosed gears, must be well-refined, high-quality oils. They must: (1) be non-corrosive to gears, (2) be neutral in reaction, (3) be free of grit and abrasive matter, (4) be resistant to oxidation and sludging, and (5) possess good demulsibility and defoaming characteristics.

For lubrication of enclosed spur, hellcal, herringbone, bevel and spiral bevel gears and worm gears, straight mineral oils are recommended. For certain worm gear operations, a compounded oil containing 3% to 10% acid-free tallow may be desirable.

The high pressures in hypold gears and, under certain conditions, in antifriction bearings, would cause film failure and metal-to-metal contact if regular lubricants were applied. It is therefore necessary to fortify the lubricant with a special extreme-pressure compound which effects a chemical reaction between the lubricant and the metal and thereby protects the metal surfaces.

C.7.3 Changing the Lubricant

In a new gear unit, the original oil should be changed as recommended by the manufacturer. The case should be flushed thoroughly with a flushing oil to remove metal particles and other foreign materials. The controlling factor here is not the oil, but the wearing in of the metal surfaces of the unit, plus any dirt that may have found its way into the unit prior to installation.

Frequency of these changes is governed by whether the equipment design and condition call for once-through lubrication or, on the other hand, application by means of a leaktight circulating system. Once-through application may require a small but continuous flow of oil or, at least, periodic lubrication at intervals of days or weeks. On the other hand, a circulating system equipped with oil purifiers and magnetic strainers, and installed on a machine operating at loads, speeds, and temperatures normal for the particular oil used can profit by an oil that will last for months or years.

C.7.4 Delivery Systems

Lubrication delivery systems supply the lubricant to the required points of application. They vary from the hand greasing of open or semi-enclosed low-speed gears to the pump-pressurized systems of enclosed high-speed gears. The most widely used is the splash system, wherein the oil sump level is high enough under all conditions to permit the larger gear or gears to pick up oil and carry it through the system. Channels in the gear case catch and drain oil to the bearings. Simple splash systems are suitable for practically all types of gears up to 2,500 ft/min (12.7 m/s) pitchline velocity.

Some gear sets operating up to pitchline velocities of 3,500 ft/min (17.8 m/s) have more than one set of gears in the same case. These types utilize oil pans under some of the upper gears as auxiliary sumps to maintain the supply for the upper gears.

High-speed gears above 3,500 ft/min (17.8 m/s) use a pump-pressurized system, because splash feed at these speeds would cause excessive churning with consequent overheating. Also, centrifugal force would act to throw the oil away from the mesh. Fig. 24 illustrates a system for intermittent application of lubricating oil to the teeth close to the mesh.

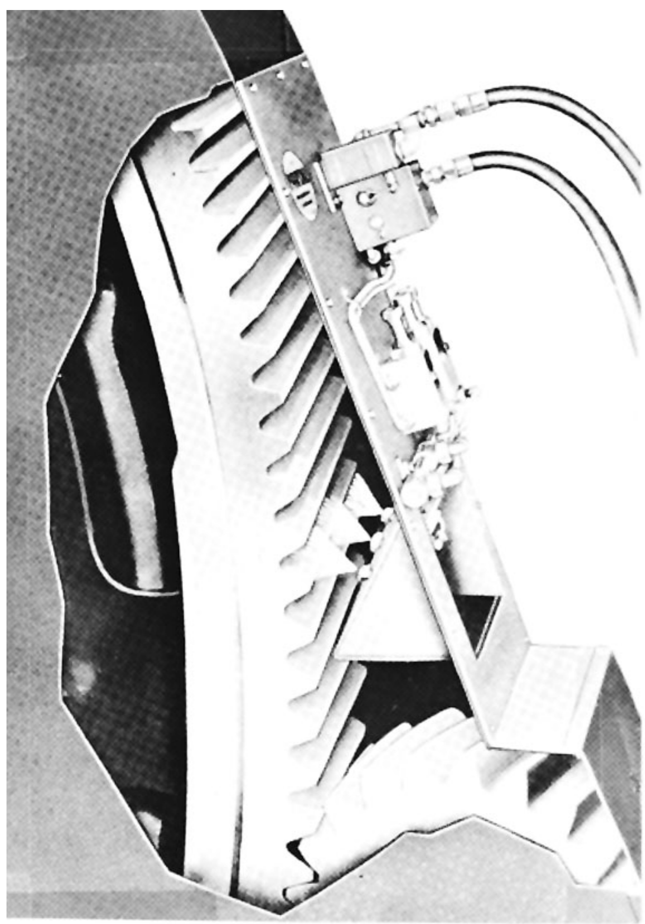

Fig. 24. An automatic spray system for gear lubrication. (Farval Div., Eaton Corp.)

C.8 Gearmotors

A gearmotor is a combination motor and enclosed speed reducing gear unit. It can be either vertically or horizontally mounted, with the motor either built in or bolted on (Fig. 25). The motor shaft extension usually carries the first reduction pinion which rotates in an oil atmosphere in the gear box. An oil seal prevents oil and oil vapors from the gear box from entering the motor enclosure. Motors used are usually 1800 rpm, 60 Hz induction type units. They are available in various enclosures such as open, completely enclosed fan cooled, and nonventilated.

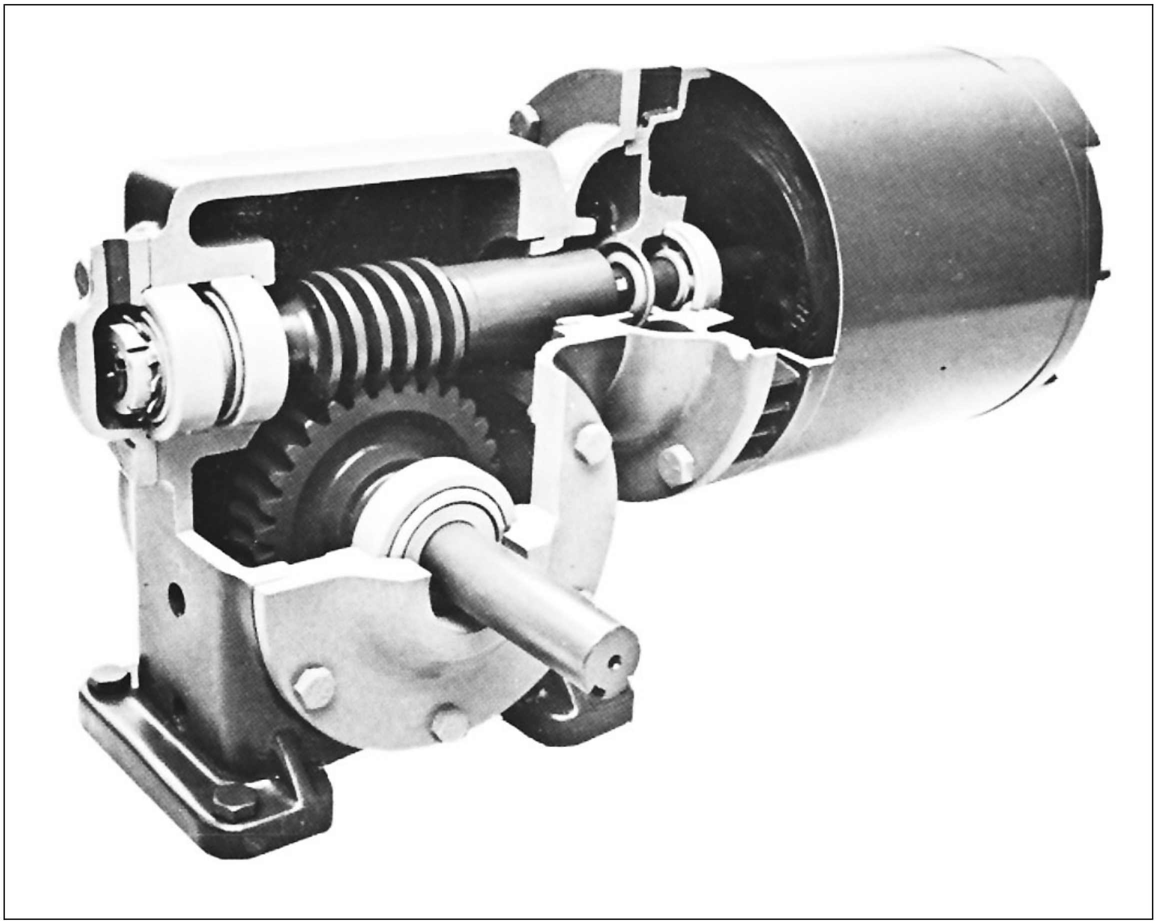

Fig. 25. Gear motor. (Winsmith Div., UMC Industries)

Since gearmotors are compact and very efficient, they are more popular than a motor and separate speed reducer. Typical applications are on conveyors, agitators, low speed fans and blowers, mixers, screens, and line shafts. Gearmotor AGMA Classifications are based on load conditions and service requirements as follows:

1. Class I. These are for use with steady loads of a rating no larger than that of the motor for eight hours per day. Moderate shock loads are tolerable when service is intermittent.

2. Class II. These are for use with steady loads no larger than the motor rating for 24 hours per day continuous duty. They can tolerate moderate shock loads 8 hours per day.

3. Class III. These tolerate moderate shock loads 24 hours per day and heavy shock loads for 8 hours per day.

Cases where extremely heavy repetitive shock loads are applied or where high energy loads must be absorbed should be individually analyzed. Continuous overload capacity of gearmotors depends on the wear resistance of the gear teeth, whereas intermittent overload capacity depends on tooth shear strength.

C.9 Installation

Every gear set or pair of gears produced represents an expenditure of much time, effort and money in their design and manufacture. Corresponding care is required in the mounting and alignment of the gears in order to obtain maximum benefits of their inherent accuracy and workmanship. The best designed and most accurately machined gears will not perform satisfactorily, and may even be entirely unusuable, if not properly installed.

A satisfactory gear installation starts with the foundation. A soil analysis should precede the location of any large machinery foundation. As with any machinery foundation, it must be capable of withstanding the reactive forces for the horsepower involved, and also be capable of damping out some vibration. The top of the foundation must be flat, level within practical limits, and allow space to fit chocks or shims between it and the bottom of the gear case. Large foundation location problems can be eliminated by installing the foundation to a line wire, transit, or tripod level.

The gear case is placed on the foundation and supported on jacking bolts. These are bolts that are threaded into the bottom bolting flanges of the gear case for the sole purpose of positioning the gear case to suit the proper plane. It is good practice to set the jacking bolts to take the weight as indicated in Fig. 26. If the unit is very large and three jacking bolts cannot support it, they may be used in pairs in the same general location shown in the sketch.

Test arbors in the shop readily establish that all planes are normal. While gauges may also be used in field assembly, the final test in either case is suitable tooth contact and backlash with the rotating elements free to float. There is no compromise with either of these requirements. The surest way to install a gear base on the permanent foundation is to take tooth contact and backlash readings before and after bolting down.

Prior to final check and prior to bolting, the jacking bolts should be removed. It is best to remove them completely, but if desired, they can be run down into the holes to protect the threads as long as they do not contact the base.

C.10 Conversion To Metric Gears

In the English System, diametral pitch was created as a convenience for relating to center distance. Diametral pitch of a gear is defined as $P = N/D$, where N = number of teeth and D = pitch diameter in inches. In the metric system, the quantity used in place of diametral pitch is the module $m = D/N$, which is the reciprocal of diametral pitch.

Most of the design equations in gear texts are suitable for use with metric gear dimensions, providing that module m is substituted for pitch. For equations involving diametral pitch, P is replaced by 25.4/m and, since circular pitch $pc = \pi/P$, pc is replaced by $\pi m/25/4$.

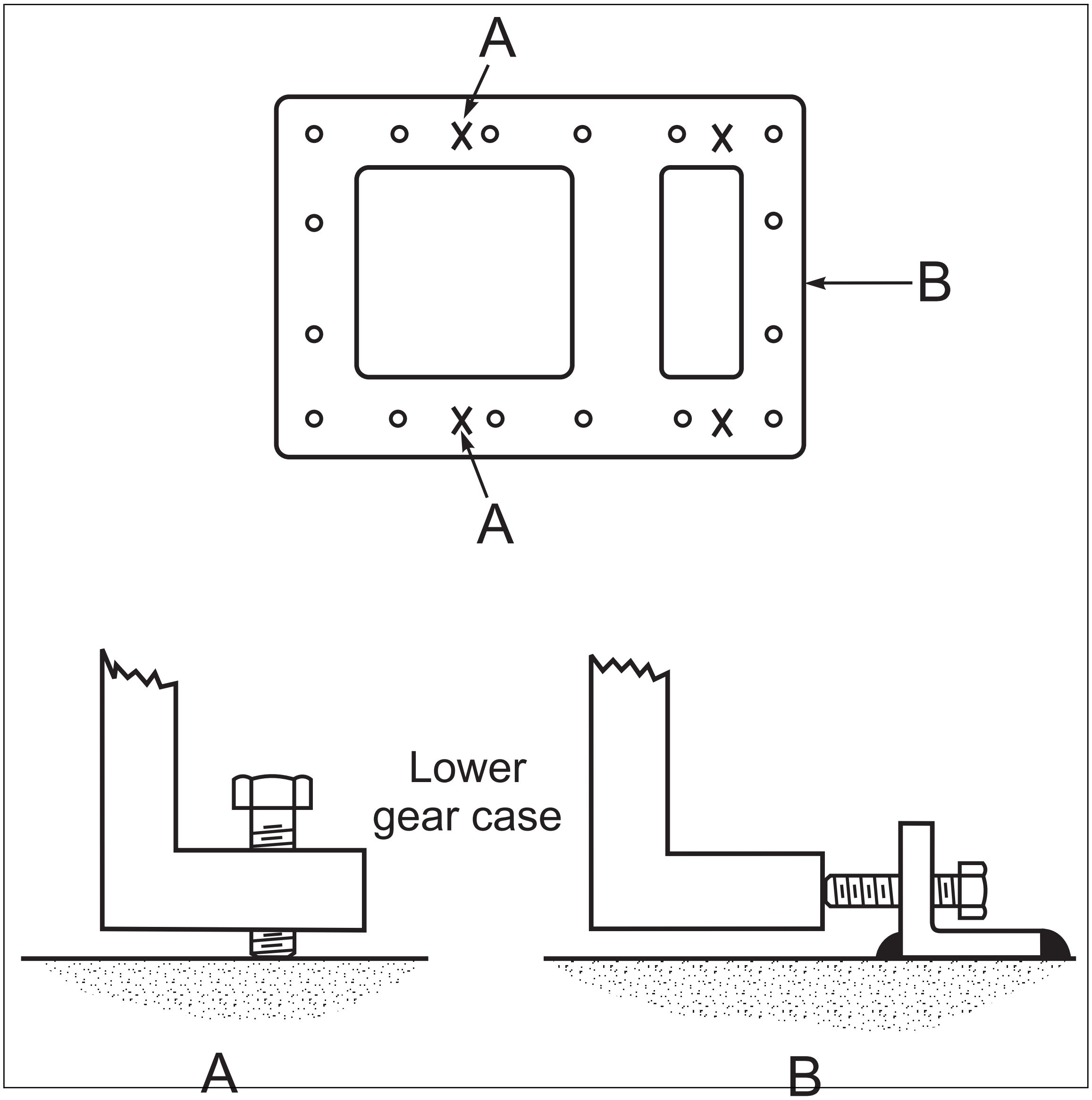

Fig. 26. Jacking bolts.

With these substitutions, all kinematic equations involving pitch parameters can be utilized for calculations. However, equations obtained by the above substitutions are still familiar ‘‘inch’’ equations. Thus, the conversions provide a way to adopt the metric module to the English System of kinematic design equations. Table 1 uses conventional formulas, producing answers in metric dimensions. For example, a pair of 24-tooth and 36-tooth gears are of module 1¼. What is the center distance? The converted equation is C = ½ [m (N1 + N2)]

Values substituted C = 0.5 [1.25 (24 + 36)]
C = 37.5 mm

Table 11. Conversion to Metric Gear Conversions.

To Obtain	*From Known*	*Use this Formula**
Pitch diameter, D	Module	$D = mN$
Circular pitch, P_c	Module	$P_c = m\pi = \frac{D\pi}{N}$
Module, m	Diametral pitch	$m = \frac{25.4}{P}$
No. of teeth, N	Module and pitch diameter	$N = \frac{D}{m}$
Addendum, $_a$	Module	$a = m$
Dedendum, $_b$	Module	$b = 1.25m$
Outside diameter, D_o	Module and pitch diameter or number of teeth	$D_o = D + 2m = m(N + 2)$
Root diameter, D_r	Pitch diameter and module	$D_r = D – 2.5\ m$
Base circle diameter, D_b	Pitch diameter and pressure angle, ϕ	$D_b = D \cos \phi$
Base pitch p_b	Module and pressure angle	$p_b = m\pi \cos \phi$
Tooth thickness at standard pitch diameter, T_{std}	Module	$T_{std} = \frac{\pi m}{z}$
Center distance, C	Module and number of teeth	$C = \frac{m(n_1 + N_2)}{2}$
Contact ratio, m_p	Outside radii, base circle radii, center distance, pressure angle	$m_p = \frac{\sqrt{{}_1R_o^2 - {}_1R_b^2} + {}_2R_o^2 - C \sin \phi}{m\pi \cos \phi}$
Backlash (linear), B	Change in center distance	$B = 2\ (\Delta C) \tan \phi$
Backlash (linear), B	Change in tooth thickness, T	$B = \Delta T$
Backlash (linear) along line of action, B_{LA}	Linear backlash above pitch circle	$B_{LA} = B \cos \phi$
Backlash (angular), B_a	Linear backlash	$B_a = 6{,}880\ \frac{B}{D}$ (arc minutes)
Min. number teeth for no undercutting, N_c	Pressure angle	$N_c = \frac{2}{\sin^2 \phi}$

*All linear dimensions in millimeters.
Courtesy Stock Drive Products, Div. Designatronics, Inc.

C.11 Troubleshooting Guide

Table 12 show useful remedies for common problems found in gears.

Table 12. Troubleshooting Guide

Type of Problem	*Possible Causes*	*Remedy*
Polishing	Metal-to-metal contact from lubricant of too low viscosity	Higher viscosity oil; Redesign gear.
Excessive wear	Failure to maintain, inspect, and record that wear was in progress	Improve maintenance program; Improve lubricant.
Abrasive wear (signs of lapped finish marks, or grooves)	Foreign material in lubricant	Use of filters; change after thorough cleaning of filters.
Corrosive wear (from chemical action)	Oil breakdown due to corrosive chemicals	Check lube oil, and change periodically.
Pitting: small	Surface irregularities; Incorrect alignment	Provide initially smooth gear-tooth surfaces, and even load distribution.
Pitting: large	Surface overload	Maintain load below endurance limit.
Spalling (larger than pitting, uneven and shallow)	High contact stresses that form voids	Redesign or lower endurance limits on gear.
Scoring (rapid wear due to failure of oil film, resulting in overheating of the mesh.)	Heat in mesh from marginal lubrication and excessive pressures	Need careful break-in period and correct loading.
Fractures (breakage of entire tooth or portion of a tooth)	Overload or cyclic stressing beyond endurance limit	Design must be within endurance limit, or material of greater strength needed.
Rib web failure	Stresses due to flexing; Development of high stress point	Increase thickness in rim to eliminate any possible stress risers.
Rippling and ridging (wave-like formations on metal surface)	Flow of surface due to high contact stresses, cyclic stresses, or poor lubrication	Increase hardness of material. Use lube oil of higher viscosity.
Heating	Overload; Oil level too low; Alignment of couplings; Adjustment of bearings	Check load, check coupling, bearing adjustment. Improve lubrication, proper function of oil pump.
Oil leakage	Bad oil seals, wrong packing; Oil drain plugs	Replace seal if worn. Check oil drains, plugs, and fittings.
Noise	Alignment, bearings loose or worn, insufficient lubrication	Check alignment, oil flow, and pressure.
Wear	Backlash of gears: may be insufficient; Possible misalignment due to worn bearings	Insufficient lubrication; Excessive temperature; Excessive speed; Excessive loads.
Bearing failure	Rust formation; Faulty lubrication; Excessive deflection of shafting; Adjustments may be too tight or too loose	Check oil lubrication, and bearing alignment.
Shaft failure	Alignment, couplings: flexible or rigid Overhung	Check alignment, coupling.