

Fundamentals, Selection, Installation and Maintenance of Gearboxes (Gear Drives) - Part 2

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The information contained in this booklet represents a significant collection of technical information on fundamentals, selection, installation and maintenance of gearboxes (gear drives). This information will help to achieve increased reliability at a decreased cost. Assemblage of this information will provide a single point of reference that might otherwise be time consuming to obtain. Most of information given in this booklet is taken from various sources for which information is given in each chapter and in the references list given at end of this booklet. For more information, please refer them. All information contained in this booklet has been assembled with great care. However, the information is given for guidance purposes only and the ultimate responsibility for its use and any subsequent liability rests with the end user. Please see the disclaimer uploaded on <http://www.practicalmaintenance.net>.

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Notes:

Drives using gears should be called gear drives because a gear drive may not have a box (enclosure). However, as gear drives are commonly called gearboxes, I have used gearboxes instead of gear drives in chapters 1 & 7 of Part 2 and in the title.

For information on the following, please view Part 1

- Use of Gear Drives
- Gear Nomenclature and Tooth Proportions
- Types of Gears
- Gear Trains
- Selection of Enclosed Gear Drives
- Gear Materials and Heat Treatment

Storage, Installation and Commissioning of Gearboxes

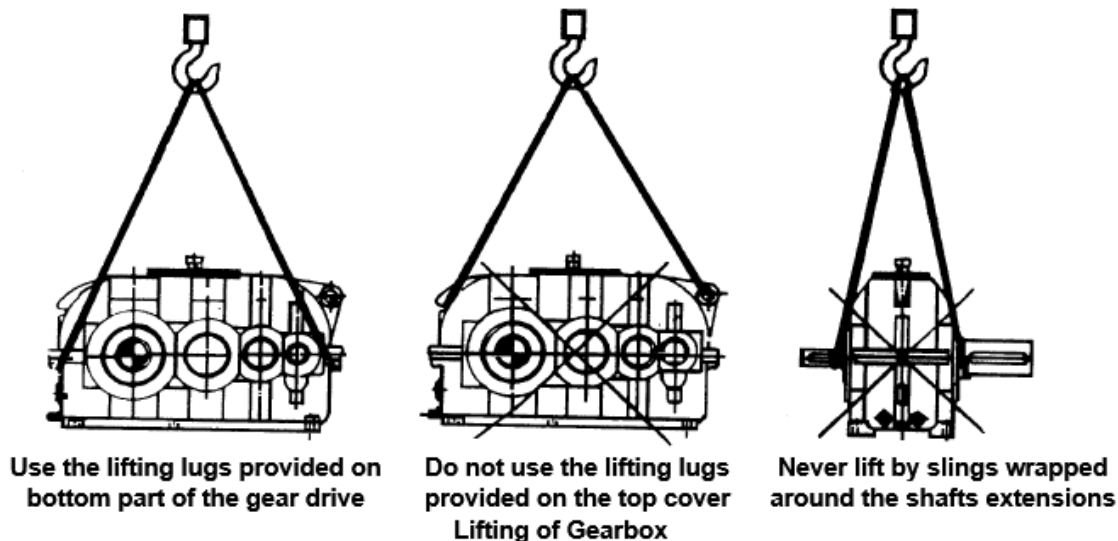
General information on handling, storage, installation and commissioning of gearboxes (gear drives) is given in this chapter. As per me, technically gear drive is a better description than gearbox. However, I have used gearbox instead of gear drive in this chapter for the benefit of persons who may search for the information on gear drive on internet using gearbox.

Handling

A gearbox should always be moved by rolling on bars or by lifting it with slings or chains through the designated lifting lugs, holes or eye bolts provided on the gearbox.

Never drag the gearbox. It will damage the machined mounting surfaces and may overstress the housing.

For equal load sharing make use of all available lifting points. Slings or chains should be adjusted to distribute the load evenly.



Never use the lifting lugs or eye bolts of the gearbox components (e.g.: top cover) to lift the entire gearbox.

If the gearbox is part of a drive package, the lifting points of the gearbox may not be used to lift the complete drive package (i.e. drive with its base frame).

Never lift by slings wrapped around the shafts extensions as serious damage may result.

Pay attention that the external components will not be damaged when handling or transporting the gearbox.

Storage

When storage is necessary it should be in-doors and preferably in a dry, vibration free area having a relatively constant temperature. When outdoor storage is unavoidable, gearboxes should be raised off the ground on skids and covered by a tarpaulin or equivalent covering.

Normal Storage

During manufacture and for intervals of storage up to four months, internal components of a gearbox should be coated with a suitable oil-based rust preventative (such as Houghton Cosmoline 1102 or equivalent). This rust preventative should contain water displacement and fingerprint suppressant additives. This is applied either by filling the unit completely, rotating the shafts and draining, or by spraying. Special care must be taken to reach all internal areas when applying preservation by spraying. External machined surfaces should be coated with a similar rust preventative during manufacture.

Units that have been factory preserved using the "Cosmoline" type preservative may require flushing with a solvent prior to filling them with the recommended operating lubricant - if hardening as a result of solvent evaporation has occurred or lubricant foams due to reaction with rust preventatives.

External and internal inspection of the gearbox should be made monthly. Any moisture observed should be removed at this time and components and surfaces recoated with rust preventative as necessary.

Adverse Conditions or Long Term Storage

In conditions of long periods of storage or storage in environments subject to high humidity, extreme temperature change, or exposure to an oxidation enhancing atmosphere, gearbox should be completely filled to the top (using a stand pipe if necessary, to assure that oil reaches the upper most parts of the gearbox) with a high quality, oil based rust inhibiting lubricant. In cases where it is impractical to fill the gearbox to the top, the lubricant should be circulated to coat all internal components and the gearbox sealed to help prevent condensation. All openings should be sealed with plugs, vapor inhibitor paper, or caps. Gearbox equipped with a breather must have the breather replaced with a plug during the storage period. RTV may be used to seal labyrinth seals. All external machined surfaces should be coated with solvent based rust preventative.

Inspection of the gearbox should be performed periodically. The sump drain should be opened and a small amount of oil removed along with any condensate that might be present. The gearbox should be refilled to the top and resealed. Oil lost through leakage should be replaced immediately to prevent any possibility of atmospheric vapor condensation within the gearbox. Outside surfaces should be recoated as necessary.

Another alternative for long term storage preservation, normally utilized where completely filling to the top with lubricant is not feasible or practical, is the use of vapor space inhibited oil or the recommended lubricant containing a mixture of vapor space inhibited oil concentrate (for example, NOX-RUST® 1100 VCI-10 OIL by Daubert Chemical Company, Inc., Website: www.daubert.com or equivalent). When using this type of protection, the lubricant vendor's instructions should be followed. Normally these instructions require that a percentage of the oil volume be drained at approximately three month intervals and a new charge of vapor inhibitor installed. It is extremely important when this procedure is used, that all openings be sealed to limit dissipation of the rust preventative vapors to atmosphere.

Before installation of a gearbox after a long term storage period, check the shaft seals and replace them if found damaged.

Note:

Most manufacturers' gearboxes are test run with break-in oil that contains rust preventive which will protect the internal parts for six months under normal storage conditions after they leave the factory.

Drive Not in Action

When the internal parts are not continually bathed by the lubricant as during operation, the gearbox is particularly vulnerable to attacks by rust and corrosion. Hence when a gearbox is installed in the system but is going to be out of action/service in excess of one week (for example - during a long outage), following is recommended.

For best results (with highly viscous lubricant), let the gearbox cool for approximately four hours after shutdown, then start the drive and let it run for approximately five minutes. This will coat the internal parts of the gearbox with cool oil. Thereafter, run the drive for five minutes or turn the gears slowly by hand once a week throughout the shutdown period to maintain the oil film on all the internal parts of the gearbox.

Installation

Proper alignment of a gearbox is absolutely essential for long and trouble-free operation. Hence to ensure long service and reliable performance, a gearbox must be rigidly supported and the shafts accurately aligned. The following describes the minimum precautions required to accomplish it.

Foundation

The foundation must be adequate to withstand normal operating loads and possible overloads. It should also maintain alignment of the gearbox with connected machinery under such loads.

Unless a gearbox is specifically ordered for inclined mounting, the foundation must be level and flat. The lubrication system may not operate properly if the gearbox is not mounted in the position for which it is designed. It may be desirable to elevate the foundation to facilitate oil drainage.

If a concrete foundation is used, steel mounting pads and bolts of sufficient size to distribute the stress into the concrete should be grouted into the foundation.

If a structural steel foundation is used (i.e., wide flange beams or channels), a base plate or sole plate of suitable thickness should be used and should extend under the entire unit. The supporting structure should be sufficiently strong so that no undue vibrations are generated. Excess vibrations are dangerous to the gearbox and may cause premature failure of the gearbox.

Shaft Mounted Units

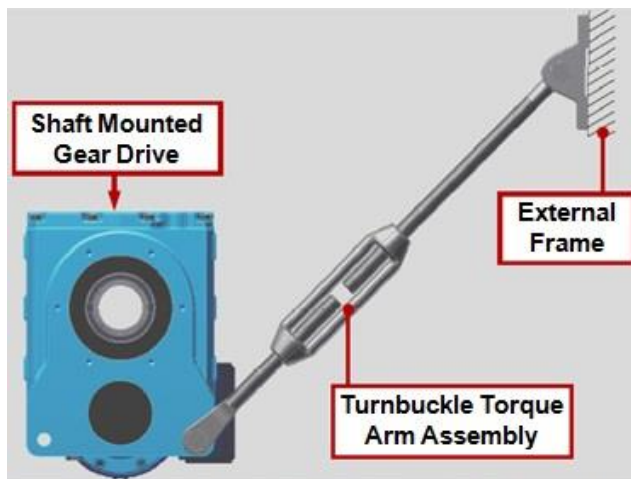
Shaft mounted gearboxes should be mounted as close to the driven equipment bearing support as possible to minimize bearing loads due to overhung load.

Assembly and subsequent dismantling is aided by applying an anti-corrosive lubricant to both the hollow shaft and the driven shaft before assembly (lubricant should not be applied if a shrink disc is used to secure the hollow shaft to the driven shaft). The unit must slide freely onto the driven shaft. Do not hammer or force the gearbox into place.

If a shrink disc is used to secure the hollow shaft to the driven shaft, follow the shrink disc manufacturer's assembly procedure.

If the shrink disc manufacturer's procedure is not available, the following may be used:

- De-grease the hollow shaft of the gearbox and the driven shaft (both shafts must be completely free of grease).
- Slide the shrink disc over the hollow shaft.
- Tighten the tensioning bolts of the shrink disc in gradual increments (approximately $\frac{1}{4}$ turn) following a circumferential pattern (e.g.: clockwise) - do not tighten in a diametrically opposite (crosswise) sequence - until all bolts are tightened to the specified torque (torque indicated on the shrink disc). When the tensioning bolts have been tightened, there must be an even gap between the clamping flanges of the shrink disc. If this is not the case, the shrink disc connection should be checked for correct fit.



Shaft Mounted Drive with a Turnbuckle Torque Arm

When a shaft mounted gearbox is mounted directly onto a driven shaft with no other external support, it must have a torque arm attached to it. A torque arm is a pivoted connecting link between the gearbox and a fixed anchor point intended to resist the torque developed by the gearbox. Quite simply, a torque arm transmits the reaction torque produced by the gearbox into the structure of the machine, thereby preventing the counter rotation of the gearbox during operation.

The torque arm should always be mounted within the angular limits specified by the manufacturer. The preferred mounting position is perpendicular to a line through the output shaft center and the point of attachment of the torque arm to the unit housing. In this position the minimum load on the torque arm will be experienced.

Despite a high degree of accuracy in manufacturing processes, shaft runout may have negative consequences on a shaft mounted gearboxes. Such runout (on the driven shaft) may cause the gearbox to “wobble” on the shaft during operation. This could be particularly troublesome if the torque arm is rigidly mounted to the anchor point. A rigidly mounted torque arm attached to a gearbox driving a shaft with a high degree of runout may result in (among other things): a broken driven shaft, cracks at the anchor point, a decrease in the life of the output bearings of the gearbox and/or the connected equipment, and lubrication leakage. Introducing some “float” in the torque arm at the anchor point will counteract these problems. The float may be introduced by providing rubber bushing or disc springs at the anchor point.

Foot Mounted Units

In case of foot mounted units, assembly of shaft connections is generally carried out before their installation.

Assembly of Shaft Connections (Couplings)

Remove the antirust compound from the shaft extensions and outside machined surfaces. This compound can be removed by a solvent. Do not use files or sand papers to remove it.

Before attempting to mount the coupling, inspect the coupling bore and shaft diameter with a micrometer to determine that the coupling bore and shaft are correct. Fit key to the coupling keyway to check for proper fit. Install key into shaft keyway. Check height of key to assure clearance at the top of the key when coupling is mounted.

Clearance or interference fits for coupling hubs should be in accordance with ANSI/AGMA 9002. As per ANSI/AGMA 9002-B04, clearance bore and interference bore shall be as under.

Nominal Bore Diameter (in)	Clearance Bore (in)	Interference Bore (in)
$\frac{3}{8}$ to $1 \frac{1}{2}$	+ 0.001 / - 0.000	- 0.0005 / - 0.0010
$1 \frac{9}{16}$ to 2	+ 0.001 / - 0.000	- 0.0010 / - 0.0020
$2 \frac{1}{16}$ to 3	+ 0.0015 / - 0.000	- 0.0010 / - 0.0020
$3 \frac{1}{16}$ to 4	+ 0.0015 / - 0.000	- 0.0015 / - 0.0030
$4 \frac{1}{16}$ to 5	+ 0.0015 / - 0.000	- 0.0020 / - 0.0035
$5 \frac{1}{16}$ to $6 \frac{1}{2}$	+ 0.0015 / - 0.000	- 0.0020 / - 0.0040
$6 \frac{9}{16}$ to 7	No Standard Tolerance	- 0.0020 / - 0.0040
$7 \frac{1}{8}$ to 8	No Standard Tolerance	- 0.0030 / - 0.0050
$8 \frac{1}{8}$ to 9	No Standard Tolerance	- 0.0035 / - 0.0055
$9 \frac{1}{8}$ to 10	No Standard Tolerance	- 0.0040 / - 0.0060
$10 \frac{1}{8}$ to 11	No Standard Tolerance	- 0.0045 / - 0.0065
$11 \frac{1}{4}$ to 12	No Standard Tolerance	- 0.0050 / - 0.0070
$12 \frac{1}{4}$ to 13	No Standard Tolerance	- 0.0055 / - 0.0075
$13 \frac{1}{4}$ to 14	No Standard Tolerance	- 0.0065 / - 0.0085
$14 \frac{1}{4}$ to 15	No Standard Tolerance	- 0.0070 / - 0.0090
$15 \frac{1}{4}$ to 16	No Standard Tolerance	- 0.0075 / - 0.0100
$16 \frac{1}{4}$ to 17	No Standard Tolerance	- 0.0080 / - 0.0105
$17 \frac{1}{4}$ to 18	No Standard Tolerance	- 0.0085 / - 0.0110

Notes:

- Class 1 clearance fits assumed; no standards for clearance fit above $6 \frac{1}{2}$ inches.
- Clearance fit bore tolerance is + 0.002/ - 0.000 for cast iron components on bores above $4 \frac{1}{2}$ inches.

Outboard pinion and sprocket fits should be as recommended by the pinion or sprocket manufacturer.

Under no circumstances, couplings or similar items should be mounted by giving heavy blows or impacts. This may cause damage to shafts, bearings, oil seals, etc.

Coupling hubs, pinions and sprockets with interference fits should be heated according to the manufacturer's recommendations, generally 250°F to 300°F, before assembling to the shaft.

Coupling hubs should be mounted flush with the shaft ends, unless specifically ordered for overhung mounting. Pinions, sprockets and sheaves should be mounted as close as possible to the gearbox housing to minimize bearing loads and shaft deflection.

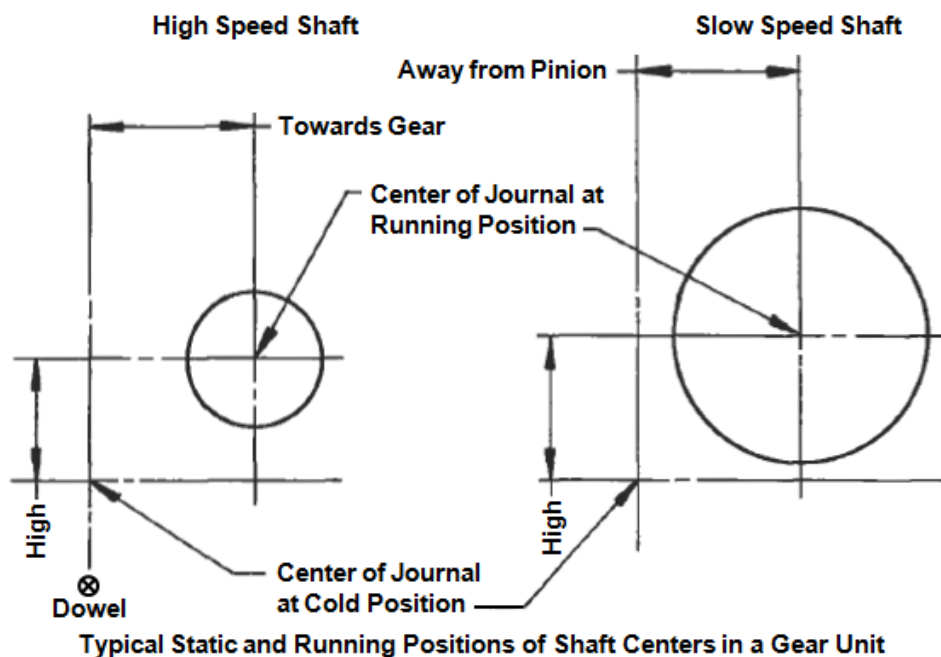
Installation and Alignment

After a suitable foundation has been laid and coupling hubs mounted, an installation procedure should start with securing the driving machine (prime mover) or driven machine (whichever is more permanently settled) and aligning the gearbox to it.

Before alignment, check run out of each coupling hub.

Ensure correct gap between two coupling halves and carry out angular and parallel (radial) alignment as per gearbox manufacturer's recommendation. While aligning, take care of the operating positions of driven and driving shafts, which will be different from the cold static positions. At operating load and temperature, the final positions of the shafts of a gearbox will differ from their positions under no load and ambient temperature due to thermal expansion of the gear housing and the mechanical loading imposed by the gear action.

Following figure shows typical static and running positions of shaft centers in a gear unit.



If there is much difference between the temperature of the gearbox and that of the driven machine, or driving machine, the resulting difference in level may be appreciable. The coefficient of linear expansion of cast iron is 0.000011 mm per mm per °C. Hence if the input/output shaft of a gearbox is located at height of 500 mm above the base plate and difference in temperature is 30°C between the gearbox and the driven machine or driving machine, the error of alignment, if correct when cold, will be 0.165 mm.

The driven and driving machines can also have thermal movements which must be either added to or subtracted from the gear movements, depending upon the directions of those movements.

Misalignment takes up the clearance space within the bearings with a corresponding axial misalignment of the gear tooth surfaces. The result could be premature wear caused by abnormally high bearing loads and irregular tooth contact.

When shims are used to level or align the gearbox, make sure that all feet are supported so that the housing will not distort when it is bolted down. Use a feeler gauge to determine thickness of required shims. After tightening the bolts, check for distortion by placing a dial indicator on the gearbox foot near the bolt to be checked. If the gearbox foot moves when the bolt is loosened, then distortion is present, and the gearbox needs more shims around the bolt. The gearbox should be checked for distortion using this method at each foundation bolt. Improper shimming will reduce the life of the gearbox and may cause failure. All bolts should be tightened to the torque value recommended by the manufacturer.

After the gearbox is properly aligned to the first component and bolted down free of distortion, a soft blue tooth contact check should be performed. If the contact pattern is satisfactory, then proceed with the installation. However, in the event that the soft blue check indicates poor tooth contact, alignment and housing distortion should be rechecked before proceeding with installation.

After completing the alignment, install dowel pins as specified by the manufacturer to prevent misalignment and ensure proper realignment if gearbox is removed for service. Leave room to redowel if necessary. Doweling a gearbox before operating it is very important since, as a gearbox is heated and cooled, it will crawl on the mounting surface and an alignment change may result.

Most manufacturers furnish starter holes for the dowel pins, and they have selected what they feel is the best location for the zero movement point on the gearbox. If manufacturer's recommendations are not available, dowel the gearbox under the shaft for which alignment is the most critical. A good rule to follow is that on high speed units, the dowel pins should be located as near as possible to the high speed pinion since this shaft has the most critical coupling alignment. Many large, heavy duty, low speed and high torque gearboxes will require the dowel pins to be placed under the low speed output shaft since this shaft has the most critical alignment point. On a gearbox which has rigid couplings on one shaft, this rigid shaft alignment will be the most critical, and dowels should be installed under the rigid shaft.

After installation of the gearbox, the installation and alignment of the third machine element should be carried out along the same basic guide lines.

After doweling the gearbox, take the trial run to check hard blue tooth contact and hot alignment. Before trial run, ensure that all work is completed as per startup / commissioning check list provided by the manufacturer.

Apply layout blue to the teeth as directed for a hard blue tooth contact check, and then start the gearbox up and operate it for a short period of time. After shutting down the gearbox, check the tooth contact pattern as evidenced by the wear-off of the bluing. If a satisfactory contact pattern is obtained, proceed for hot alignment check. However, should the contact pattern be poor, corrective measures should be taken as recommended by the manufacturer.

If hard blue tooth contact is satisfactory, run the gearbox for hot alignment check. After the gearbox has run long enough at stabilized operating temperature, a hot alignment check should be made. For hot alignment check, the gearbox should be stopped, the coupling halves separated, the alignment checked, indicator readings recorded, and any necessary corrections made before the shafts and drive cools down. After this initial hot alignment check

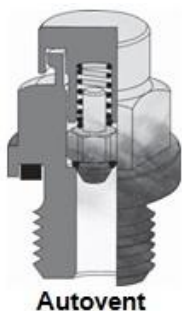
is made, the gearbox should be redoweled to the foundation if any corrections were carried out before the drive cools off. Then an additional hot alignment check should be made to ensure that gearbox is correctly aligned.

After completion of precise alignment (positions of mating shafts are such that during normal operation no flexing takes place at coupling) is carried out as explained above, each member (prime mover, gearbox and driven member) must be securely bolted and doweled in place.

Breather

Gearboxes normally are shipped with the breather vent plugged and the breather shipped loose in a bag or box attached to the gearbox. Install breather before putting the gearbox into operation. Efficient lubrication is hindered when dirt, water vapor, or reactive chemical fumes are allowed to enter the gearbox. The breather is designed (for example, with filter) to inhibit such contaminants from entering the gear case/housing while at the same time releasing pressure caused by heat generated within the gearbox. Without proper venting, the increased internal pressure might force lubricant through the oil seals thus damaging the seals. Installing a desiccant breather to the gearbox assures that the air ingested is dry and clean.

When the gearbox is to be used in moist areas such as wet cooling towers, paper mills, or similar installations, it is recommended that the breather plug be removed from its location in the upper housing - and an elbow and breather line be installed from the housing to a location that will minimize the entrance of moist or polluted air.



For humid conditions, wash down applications and dusty environments, Autovent™ may be used as shown in above figure. The Autovent helps prevent bearing and gear damage by blocking entry of foreign material (water, dust, corrosives, etc.) through the breather that acts like a check valve. The typical design utilizes a spring pressing a ball or a sealing plunger, which presses against a machined orifice. The check valve opens at approximately 2 psi during operation and closes tightly when the gearbox cools, producing a slightly negative pressure to ensure a tight seal. This keeps contaminants out of the oil. It helps maintain proper oil cleanliness, reduces oil foaming and oxidation.

Be careful when applying paint near the breather (do not plug the breather with paint).

In extreme harsh environments, the gearbox can be completely sealed, and the pressure variation can be accommodated by an expansion chamber with a flexible diaphragm.

Forced Lubrication System

If forced/pressure lubrication system is provided for the gearbox, install various components (oil pumps, coolers, filters, etc.) and instruments like pressure switches, flow switches, RTDs, etc. as per manufacturing drawing. Commission the system and check set points of various instruments and interlocking logic.

Startup/Commissioning Check List

Before startup, certain checks should be made to protect personnel and equipment. If a checklist written by the manufacturer is not available, the following list can be used as a guideline.

- Grease couplings, seals and bearings (if grease lubricated).
- Make sure that gears have backlash.
- Check for free rotation of the shafts.
- Check that prime mover will drive the gearbox's input shaft in the correct direction.
- Fill the gearbox up to the proper oil level with the correct type and grade of specified lubricating oil.
- Ensure that breather vent (ventilator) is installed.
- In case of forced lubrication system, check completion of required piping and tightness of pipe connections.
- Ensure that switches, alarms, heaters, coolers and other safety and protection devices are installed and operational for their intended purposes.
- Check installation of inspection covers.
- Check installation of coupling guards.
- Check installation of adequate machinery guards to protect operating personnel as required by the applicable standards of the Occupational Safety and Health Administration (OSHA), and by other applicable safety regulations.

When the entire machinery package has been completely and satisfactorily installed, it should be started up and operated. During the first day or two of operation, special attention should be focused on bearing and lubricating oil temperatures and on housing and shaft vibration to detect problem if any.

Startup Procedure

If the gearbox is equipped with heaters for low temperature operation, turn on heaters and allow oil temperature to rise at least 40° F.

If a shaft driven lubricating oil pump is used, be sure that it is primed to minimize the time necessary to build up a positive pressure. Pressure should build up in 10 to 15 seconds. If pressure does not develop, stop the gearbox and rectify the problem. When an electric driven lubricating oil pump or some other remote pump provides lubrication for the gear, run that pump a few minutes prior to starting the gearbox to provide initial lubrication during startup.

Check for correct water flow and temperature through the lubricating oil cooler if provided.

Check for lubricating oil and cooling water leaks.

Start the gearbox under as light a load as possible and operate it until temperatures stabilize.

Checked the gearbox constantly for unusual sounds, excessive vibration or excessive heat. If any of these problems develop, the gearbox must be stopped immediately and the cause determined and corrected. The operating temperature must not exceed 200°F at any location in the gearbox.

If possible, the gearbox should be operated under a light load (approximately half load) for one (1) or two (2) days to allow final break-in of gears. After this period, the gearbox can be operated under normal load.

It is recommended that the lubricating oil in the gearbox should be changed after the initial two weeks of operation.

Running-In a New Gearbox

In contrast to pitting and bending fatigue, which only occur after a period of running time (initial start-up period), scuffing may occur immediately upon start-up. In fact, gears are most vulnerable to scuffing when they are new and their tooth surfaces have not yet been smoothed by running-in (also called breaking-in or wearing-in). For this reason, it is wise to run-in a new gearbox under one-half load for at least 10 h to reduce the surface roughness of the teeth before applying full load.

One of the gearbox manufacturer cautions/recommends as under.

If a gearbox is to be run for a long period at low speed during running-in period for the connected equipment, the oil level should be raised sufficiently to allow bearings to ride in the oil since the splash lubrication is relatively ineffective at low speeds. Reduce the oil level to normal after regular speeds are resumed.

Tooth Contact Checking

Information about why one should check gear tooth contact, how the actual check is made, and how to interpret the tooth contact checks on involute double helical gears, spiral bevel gears and worm gears is given in this chapter.

Why Check Tooth Contact

Gear teeth must have an even load across the entire face width to minimize stress on the teeth. The contact between gear teeth is line contact; therefore, the alignment between the rotating elements (pinion and gear) is critical. Tooth alignment is controlled by the accuracy of the rotating elements, the housing, and the bearing assembly.

When to Check Tooth Contact

Twisting of the housing either during shipment or because of poor foundation conditions will cause poor tooth contact, incorrectly installed rotating elements or bearings will cause poor contact, and of course poorly manufactured parts will also cause poor contact. Therefore, tooth contact should be checked on all new installations, after any disassembly of the gear unit, and after any housing-to-foundation change. It may also be checked as part of routine annual maintenance or when a problem related to alignment is suspected. Contact must be checked on the job foundation to be sure that the unit will operate properly.

Checking Tooth Contact

Tooth contact can be checked two ways:

- Soft Blue - Apply soft machinist's bluing or transfer bluing to the teeth of one gear and roll that gear by hand through mesh with its mating gear. (The terms "blue" or "bluing" are used for convenience; the dye is available in other colors.) The transfer of the blue from one gear to the other gear is read as the contact.
- Hard Blue - Paint the gear teeth with hard or layout blue, run the gear unit, and observe the pattern of 'wear-off' of the bluing.

Contact checking may usually be accomplished through the inspection cover port. Occasionally, soft blue checking is done with the housing cover removed, such as during the reassembly process.

Soft Blue Method

The soft blue method is usually done first. Since the unit is not running, this check does not give true contact. It does give a good indication of what contact will be. If it indicates inadequate contact, maintenance personnel may choose not to start the unit until contact is corrected. If the unit has been disassembled, then a soft blue check before the housing cover is installed may save a tear-down to correct contact. This is especially important if a new set of rotating elements or bearings is installed.

Soft blue is usually applied to three or four teeth on the pinion in two places 180° apart.

Clean the teeth thoroughly with solvent, and brush on the blue in a very thin and even layer. One-inch-wide good quality paint brush with the bristles cut off to a length of about one inch makes a good blue application brush. In case of a double helical gear drive, center the gear

set. Now hold a drag on the gear and roll the pinion through mesh with the gear. In order to produce a clear, readable contact pattern you must apply a load on the gears when you rotate them. Rotation direction is not important, but the contact must be checked on the loaded flank, not the unloaded tooth flank. Observe the blue that transferred from the pinion to the gear. This is the contact pattern.

A piece of cellophane tape can be used to remove the blue pattern from the gear and save it for maintenance records. After the check, place a piece of tape on the gear tooth flank and press it firmly on the tooth. Remove the tape, and place it on a clean sheet of white paper and label it with the location where the check was made. Or pattern may be photographed.

The contact should be checked at three places around the gear, approximately 120° apart. However, the blue must be reapplied and smoothed on the pinion after each meshing.

Hard Blue Method

Thoroughly clean the area where hard blue is to be applied. The teeth must be absolutely free of oil, or the blue will not adhere properly and large flakes will chip off, making the contact check inaccurate. In view of this, just cleaning with a solvent such as naphtha is not sufficient, since this procedure will not completely remove the lubricant. Additional cleaning with a volatile solvent such as electrical contact cleaner, or lacquer thinner is necessary.

Apply the blue to an area three or four tooth wide at four places on the gear and at two on the pinion. Run the unit (usually at full speed). Running conditions may vary from no load to full load. The best way is to run the unit at very light load (up to 20%) for two hours or so, and then shut it down and check the contact. With higher loads the unit should run a shorter time before checking contact. The trick is to run the unit just long enough to wear the blue off the areas of higher contact stress. High loads can mask poor contact and give a false reading.

Information for interpretation of tooth contact for different types of gears is given in the following sections.

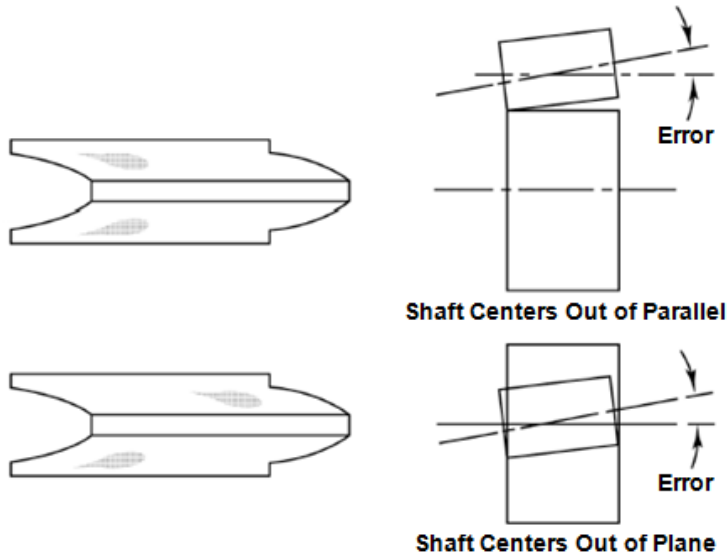
Contact Patterns for Spur and Helical Gears

In case of spur and helical gears (parallel axis gears), a check of contact is typically only done to verify the accuracy of the installation because they usually offer very little opportunity for change. Assuming properly manufactured parts, minor corrections can be made to the tooth contact by shimming the gear housing. Most of the time, the two supporting shafts are fixed in their relationship to each other by the location of support bearings in the gear housing. Hence, all you can do is to move the shafts axially in relation to each other. If the gears are having crowning, the axial movement can be used to center the contact pattern.

Remember that many wide face width single and double helical gears have helix angle modifications to correct for torsional twist and bending deflections of the teeth due to operating loads. When these modifications are made, the helix angles on two mating gears are purposely cut differently so that they will be the same when the gear teeth deflect as the design load is applied. The tip or root relief modifications are also designed to improve load distribution when a unit is operating under load. However, these modifications can make the contact appear quite bad under no load, as in a soft blue check.

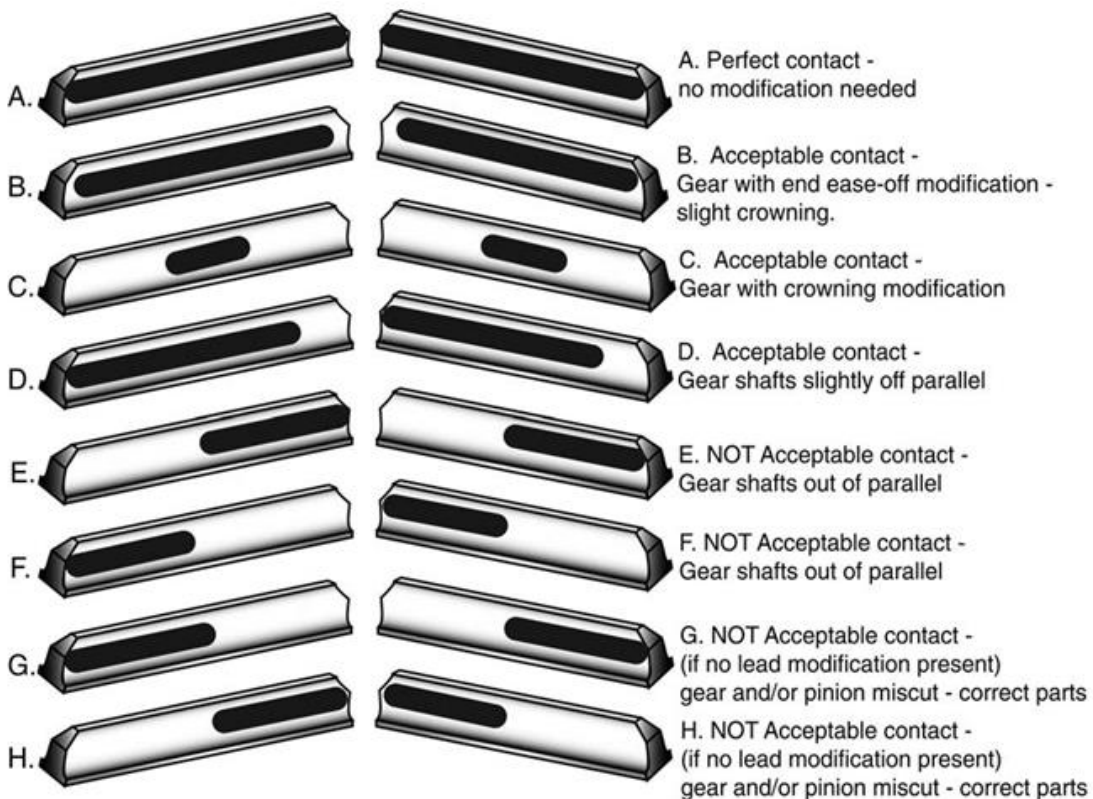
The following information is to be used only for guidance in deciding if tooth contact is adequate. Exactly what contact should be acceptable should be based on the manufacturer's recommendations and experience.

Generally, with a soft blue check you are looking for some blue to transfer, usually in a line that covers at least 80% of the face width. Do not be alarmed by a lack of blue covering the face of the tooth. Face contact should normally not extend entirely to the tip of the tooth. The following figure shows typical contact patterns for spur and helical gears with shaft center errors (shafts out of parallel or out of plane).



Typical Contact Patterns for Spur and Helical Gears

The following figure shows typical contact patterns for double helical gears. Keep in mind that a soft blue contact will not produce such dark impressions - look for the same pattern in a “sketchy” impression.



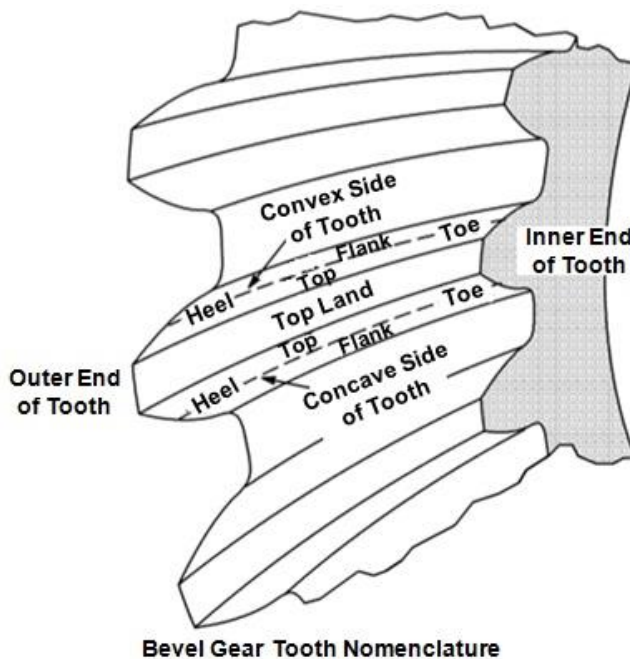
Typical Contact Patterns for Double Helical Gears

The hard blue check can be done from no load to full load, and the results will vary with the load condition. If the unit is run at no load the test will usually appear similar to a soft blue check. More blue will wear off the pinion than the gear due to the higher number of cycles the pinion sees. As the load increases, blue will wear off more of the tooth flank. Look for evidence of even load across as much of the gear tooth, both flank and face width, as possible.

In case of non-parallel axis gears, though the shafts are still fixed in their relationship to each other, axial movement can greatly affect the contact pattern and the backlash. This is true for bevel, hypoid and worm gears. One can get proper contact and backlash by mounting them at the position the manufacturer made them at.

Contact Patterns for Bevel Gears

The quality of performance that is designed and manufactured into a set of bevel gears (gear and pinion) can only be achieved by the correct mounting of the gears during assembly. Hence these gears are usually mounted on tapered roller bearings with shims to make it possible to position them axially and thereby obtain proper tooth contact and backlash. In view of this, it is important to check the tooth contact pattern of bevel gears after assembly to verify that they are assembled correctly.



Above figure shows bevel gear tooth nomenclature (as per ANSI/AGMA 2008).

Tooth Contact Pattern is the area of a tooth surface that contacts the corresponding tooth surface of a mating gear.

Heel refers to the outer end of the tooth.

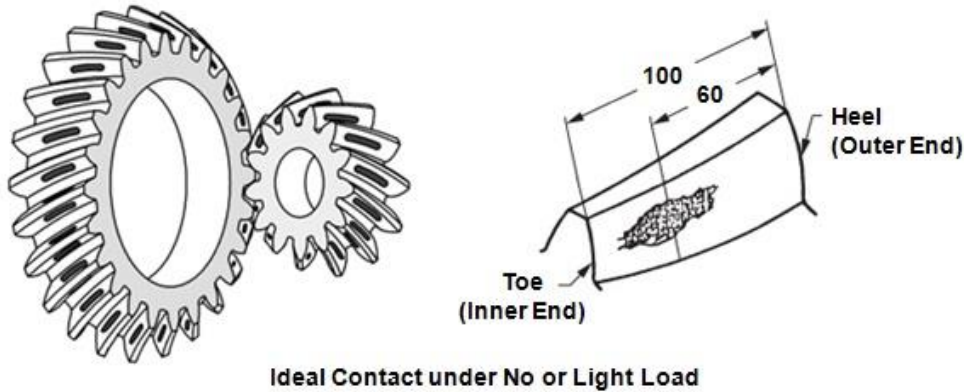
Toe refers to the inner end of the tooth.

Top of the Tooth is the upper portion of the tooth surface.

Flank of the Tooth is the lower portion of the tooth surface.

Top Land of the Tooth is the surface of the top of the tooth between the sides.

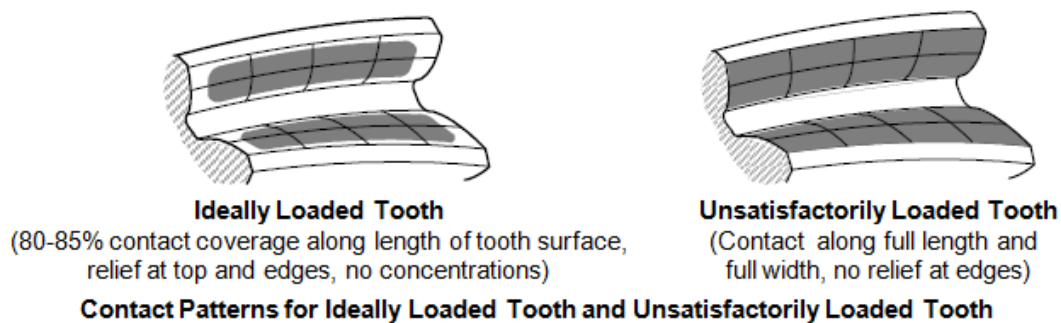
The tooth contact checking procedure for bevel gears is the same as that described for helical gears except the expected contact pattern is different.



In general, ideal contact favors the toe end under no or light load, as shown in above figure. As load is increased to full load, contact shifts to the central part of the tooth width.

Depending on the amount of load applied to bevel gears, deflections occur, and changes appear in the tooth contact pattern. On certain applications it is desirable to modify the tooth contact pattern in the generating operation to allow for stresses which are present under operating conditions. It is important to note that in such cases, light load patterns will differ markedly from the typical pattern shown in above figure as a result of the particular deflections under operating conditions. In these cases, the applicable light load pattern must be specified by the gear drive manufacturer.

Under full load, the tooth contact pattern should utilize virtually the total tooth length without heavy concentration at the ends or tops of the teeth of either member. The contact pattern should have a slight relief at the ends, along the flank and top of the tooth profile.



Above figure shows contact patterns under full load for the ideally loaded tooth and unsatisfactorily loaded tooth.

For more information on typical satisfactorily loaded tooth contact patterns and typical unsatisfactorily loaded tooth contact patterns, please see Annex F of ANSI/AGMA 2005 - Design Manual for Bevel Gears.

Even when a gear is ideally manufactured, it may reveal poor surface contact due to lack of precision in housing or improper mounting position, or both. Usual major faults are:

1. Shafts are not intersecting, but are skew (offset error).
2. Shaft angle error of gear housing.
3. Mounting distance error.

Errors 1 and 2 can be corrected only by reprocessing the housing/mounting. Error 3 can be corrected by adjusting the gears in an axial direction. All three errors may be the cause of improper backlash.

Offset Error of Shafts

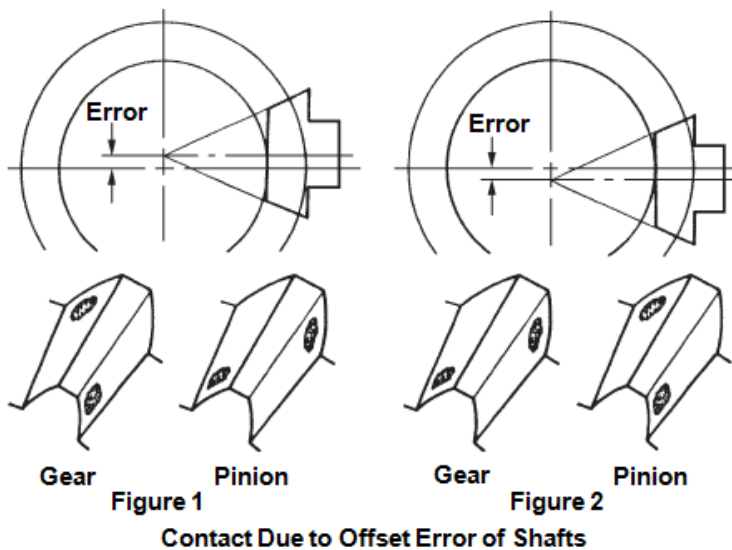
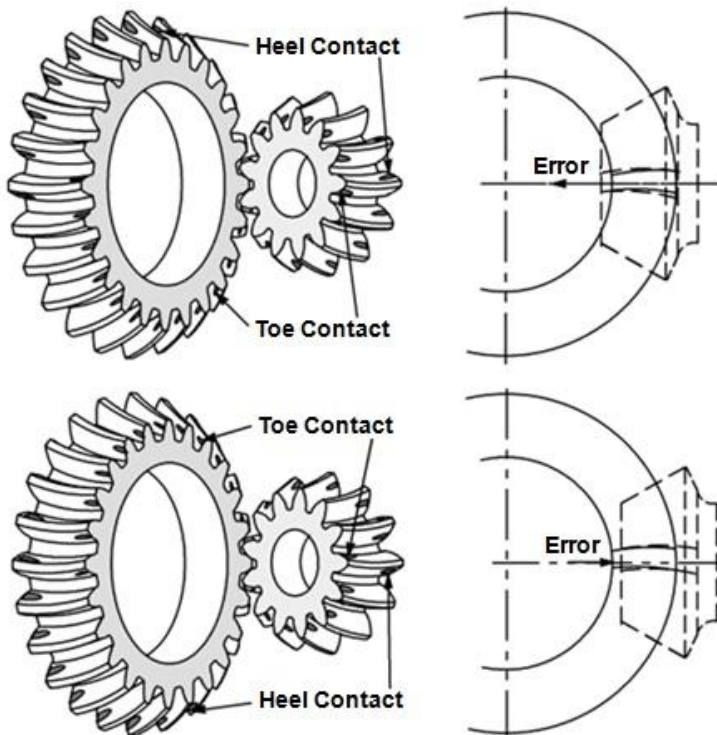


Figure 1 Figure 2
Contact Due to Offset Error of Shafts

As shown in above figure if gear housing has an offset error, then it will produce cross contact pattern. In a cross contact pattern, the contact pattern appears at the toe on one side of the tooth and at the heel on the other.

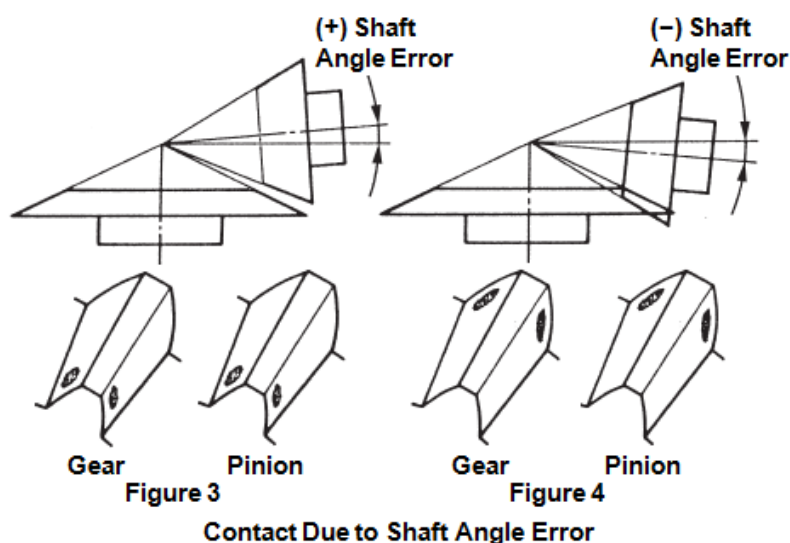


Cross Contact Pattern in Zerol Bevel Gears

Zerol bevel gears are curved tooth bevel gears with zero spiral angle. Gears with spiral angles less than 10 degrees are also sometimes referred to by the term zerol bevel gears.

As shown in above figure for zerol bevel gears, a contact pattern at the toe of one side of the tooth and at the heel of the other side may be indicative of improper positioning of one of the members. If the contact is at the toe on the concave side of the tooth of a member and at the heel of its convex side, that member should be moved axially into mesh. If the contact is opposite, that member should be moved axially out of mesh. If this movement does not correct the cross contact pattern, offset error exists.

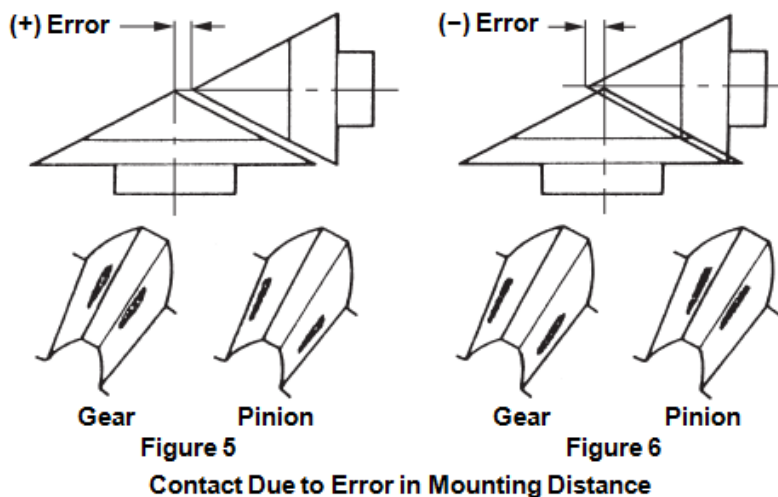
Shaft Angle Error



As shown in above figure, the contact pattern will be toward the toe end (towards the toe end on both sides of the teeth for both members) if the shaft angle error is positive (shaft angle is too large) and the contact pattern will be toward the heel end (towards the heel end on both sides of the teeth for both members) if the shaft angle error is negative (shaft angle is too small).

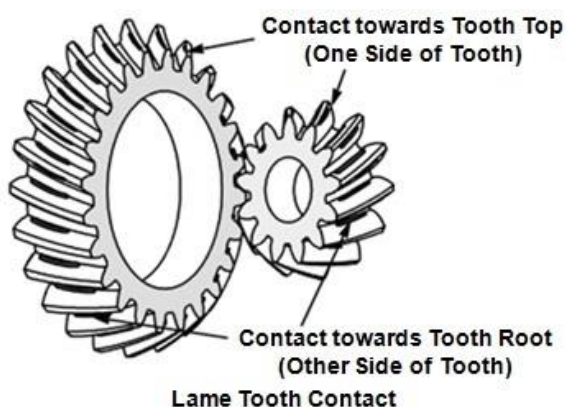
Mounting Distance Error

Note: Mounting distance error is also called profile error by some manufacturers.



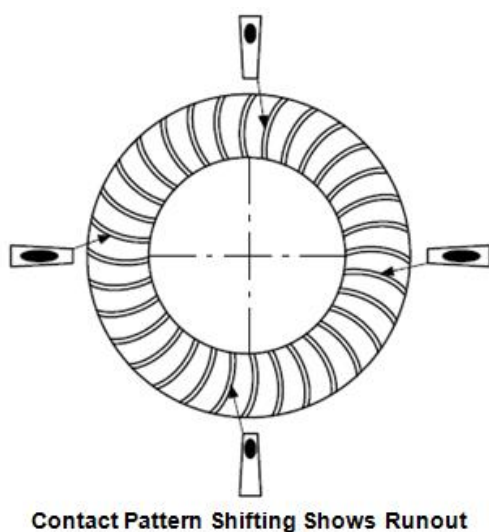
As shown in above figure when the mounting distance of the pinion is a positive error, the contact pattern of the pinion will be towards the tooth root (on both sides of the tooth), while the contact pattern of the mating gear will be toward the top of the tooth (on both sides of the tooth). On the other hand, if the mounting distance of the pinion has a negative error, the contact pattern of the pinion will be toward the top (on both sides of the tooth) and that of the gear will be toward the root (on both sides of the tooth). These errors may be diminished by axial adjustment with a backing shim.

Mounting distance error will cause a change of backlash. Positive error will increase backlash while negative, decrease. Since the mounting distance error of the pinion affects the surface contact greatly, it is customary to adjust the gear rather than the pinion in its axial direction to get the desired backlash.



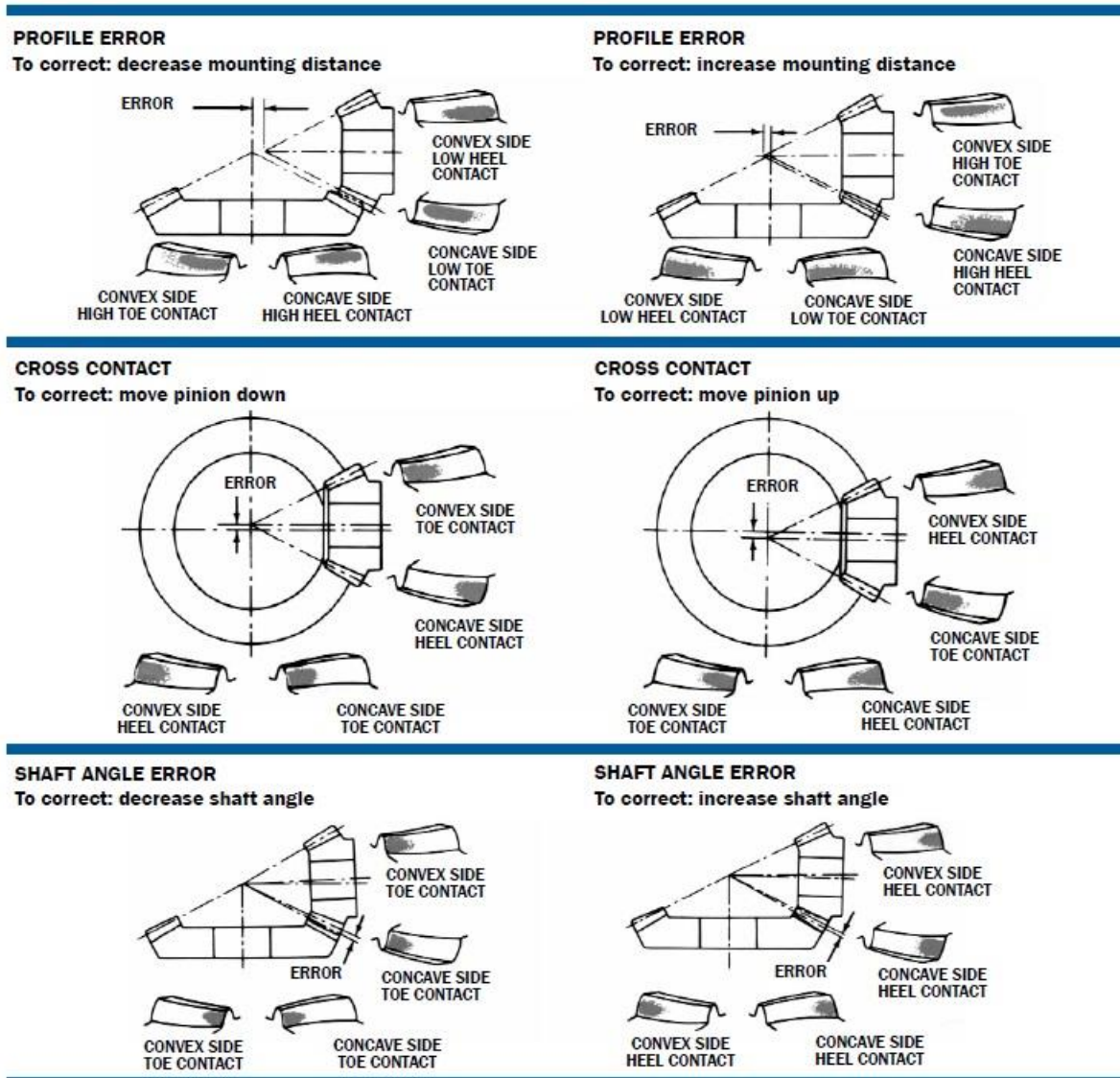
As shown in above figure when the contact pattern appears to be towards tooth top on one side of the tooth and towards tooth root on the other side (on both gear and pinion), the pattern is known as a lame tooth contact. If the pinion is positioned to the proper mounting distance and the gear set has the proper amount of backlash, then the teeth of the pinion, gear or both have been improperly manufactured and the gear set should be replaced. However, the set of gears is to operate in only one direction it may be possible to axially reposition the pinion to correct the tooth contact pattern on the drive side and to ignore the pattern on the opposite side.

Contact Runout



As shown in above figure a gear runout is characterized by the tooth contact pattern shifting progressively around the gear from heel to toe and toe to heel.

As a ready reference, following figure shows reasons for various types of contact patterns and corrective action.

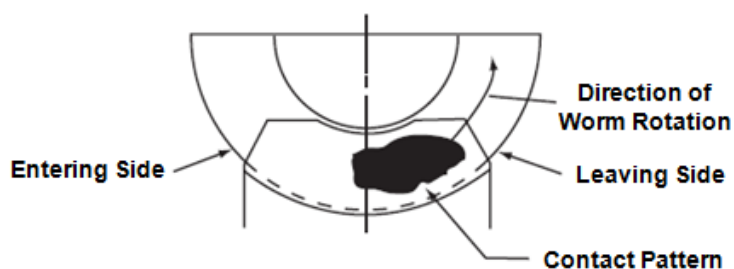


Types of Contact Patterns and Corrective Action

(Note: Pinion member is left hand in all illustrations.)

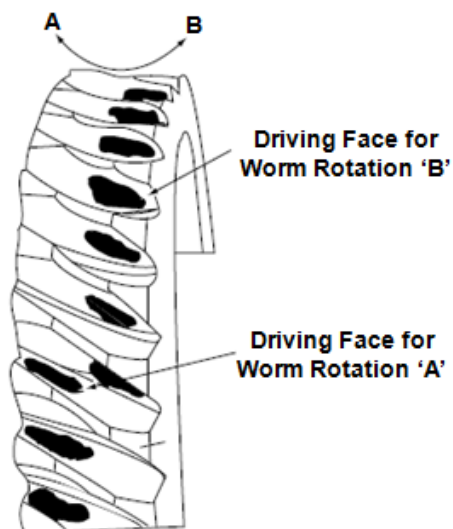
All illustrations: Courtesy of The Gleason Works.

Contact Patterns for Worm and Worm Gear



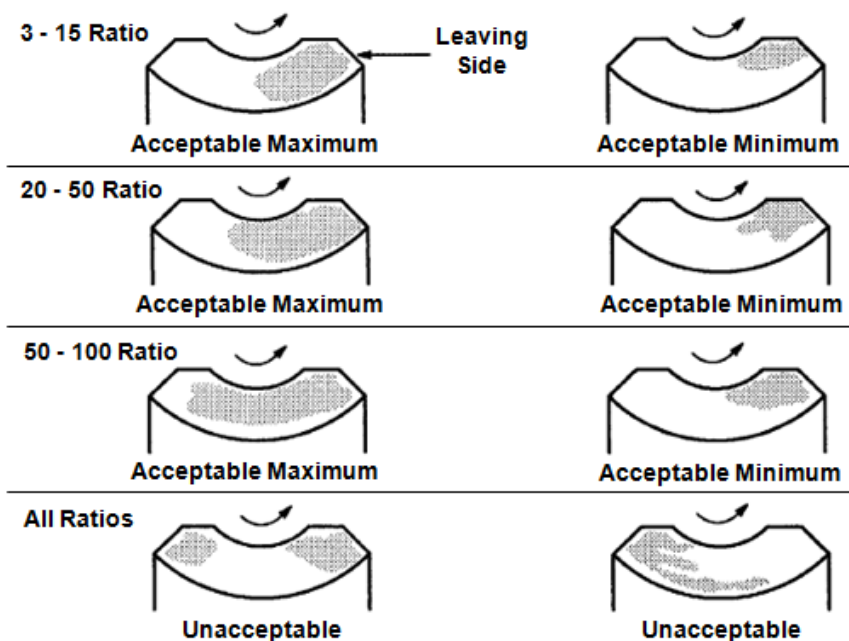
Typical Initial No-load Contact Pattern for Worm Gear

Above figure shows typical acceptable initial no-load contact pattern for a worm gear with a “leaving side” contact. The acceptable contact pattern leaves an entry gap where the worm enters the worm gear tooth. The entry gap allows for easy entry of lubricant during initial running of the gear set, and when the worm gear deflects axially and the worm bends under load, the contact tends to become more central while still leaving some entry gap for the lubricant.



Contact Patterns for Rotation in Both Directions

When the worm gear is to transmit load in both directions of rotation, the other drive flank should have a similar “leaving side” contact which is diagonally opposite as shown in above figure (diagonally opposite because the worm is rotating in the opposite direction of rotation).



Initial No-load Contact Patterns for Worm Gears

All Illustrations: Courtesy of ANSI/AGMA 6022-C93

Above figure shows acceptable and unacceptable initial no-load contact patterns for various ratios of worm gear drives.

In case of unacceptable initial no-load contact patterns, there is no entry gap for lubricant and deflection may aggravate the condition further. Hence worm gear mounted in this manner may result in a temperature rise in the lubricant and the worm gearing as much as 20% higher than a properly hobbled and mounted worm gear.

An excellent worm gear in conjunction with a defective gear housing or improper mounting position will decrease the level of contact pattern and the performance.

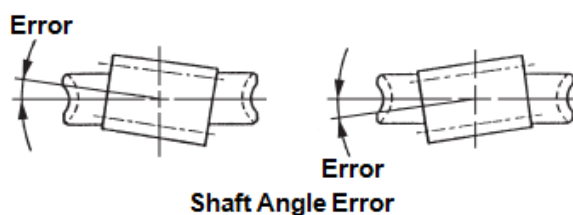
Even when a gear is ideally manufactured, it may reveal poor surface contact due to lack of precision in housing or improper mounting position, or both.

The three factors which may influence the contact pattern are:

1. Shaft angle error.
2. Center distance error.
3. Mounting distance error of worm gear.

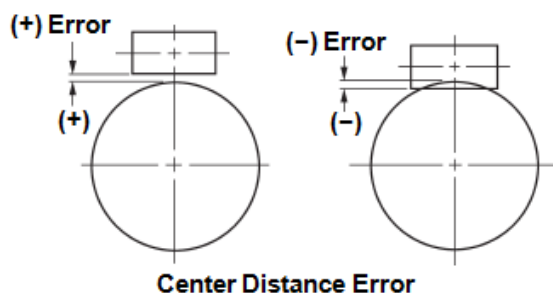
Errors number 1 and number 2 can only be corrected by remaking the housing. Error number 3 may be decreased by adjusting the worm gear along the axial direction. These three errors introduce varying degrees of backlash.

Shaft Angle Error



If the gear housing has a shaft angle error, then it will produce crossed contact. It may be noted that a helix angle error will also produce a crossed contact.

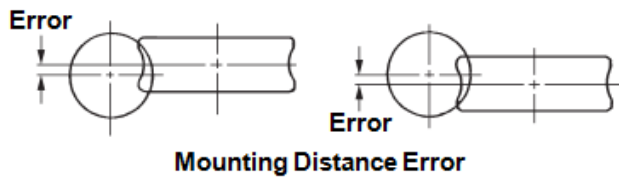
Center Distance Error



A gear housing having a center distance error will also produce crossed contact.

Such errors not only cause bad contact but also greatly influence backlash. A positive center distance error causes increased backlash. A negative error will decrease backlash and may result in a tight mesh, or even make it impossible to assemble.

Mounting Distance Error



The mounting distance error can be corrected by adjustment of the worm gear in the axial direction. As shown in the following figure, axial adjustment of the worm gear (by means of shims) to one side by a small amount can result in an acceptable “leaving side” pattern. This error affects backlash, which tends to decrease as the error increases.



Note:

To shift the contact pattern on the worm gears that are straddled mounted (mounted between bearings), interchange worm gear shaft bearing retainer shims from one end to the other (move the worm gear to the right or left of the worm), as required. Shims should not be added or deleted after axial end play has been established.

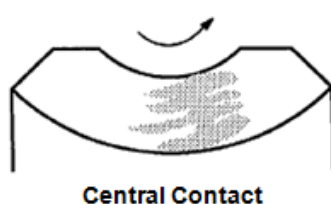
If the worm gear is mounted on a shaft with a radial runout or against a shaft shoulder or flange with axial runout, the initial worm gear contact can vary around the worm gear. To find out this type of mounting problem, the initial worm gear contact should be checked at three places at 120° around the worm gear.

Worm Gear Contact under Load

While initial worm gear contact under no-load should be on the “leaving side”, when the worm gear is operated under load for several days the contact should become almost full-face contact to keep the contact stresses low.

In the first 200 to 500 hours of operation, usually some worm gear pitting and wear occurs while the worm gear teeth are “worn-in” to suit the actual loads being transmitted.

Special Initial Worm Gear Contact



For certain applications of power drives, it is advantageous to have initially central contact on the worm gear teeth as shown in above figure. It can be produced by cutting the worm gear with over size hobs.

It is advantageous to have central contact for applications where the worm gear drives the worm (speed increasers) or for reversing drives (for example, cranes, elevators or other lifting applications where loaded flank does not change with changes in direction of rotation). Central contact is required where the worm gear drives the worm because in such case the normal “leaving side” contact becomes “entering side” contact.

Contact Checking Dyes

Following dyes may be used for soft blue check and hard blue check.

Soft Blue Check

Dykem: Hi-Spot Paste 83307 (Website: www.itwprofessionalbrands.com)
Permatex: Permatex® Prussian Blue 80038 (Website: www.permatex.com)

Hard Blue Check

Dykem: Steel Blue® (Product Code: 80300 for Brush-In-Cap 4 oz., 80600 for 930ml bottle)

References

This chapter is written based on information from the following sources.

Installation, Operation, & Maintenance Manual by Lufkin Industries, Inc. U. S. A.

ANSI/AGMA 2008-C01, Assembling Bevel Gears, published by American Gear Manufacturers Association 1500 King Street, Suite 201, Alexandria, Virginia 22314.

ANSI/AGMA 6022-C93, Design Manual for Cylindrical Wormgearing, published by American Gear Manufacturers Association 1500 King Street, Alexandria, Virginia 22314.

Gear Technical Reference by Kohara Gear Industry Co., Ltd. | Website: www.khkgears.co.jp

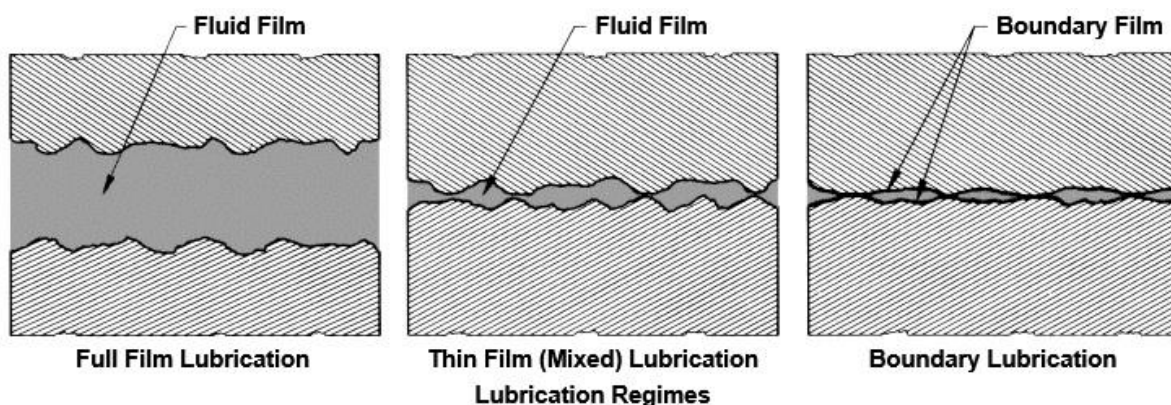
Lubrication of Gear Drives

In gear applications a lubricant is used to control friction and wear between the mating surfaces. In enclosed gear drive applications, it is also used to transfer heat away from the contact area to a point of lower temperature where it can be dissipated. It also serves as a medium to carry the additives that may be required for special functions and works as a conduit in which it transports contaminants to filters to be removed. Information about lubrication regimes, methods of lubrication, types of lubricants and selection of lubricant for gear drives is given in this chapter.

Lubrication Regimes

Three different lubrication regimes that can be present between the teeth of two meshing gear elements are boundary lubrication, thin film (mixed) lubrication and full film lubrication. Full film lubrication can be broken down into two forms: hydrodynamic and elastohydrodynamic. Hydrodynamic lubrication occurs when two surfaces in sliding motion (relative to each other) are fully separated by a film of fluid. Elastohydrodynamic lubrication is similar but occurs when the highly loaded surfaces are in a rolling motion (relative to each other). The film layer in elastohydrodynamic conditions is much thinner than that of hydrodynamic lubrication, and the pressure on the film is greater. It is called elastohydrodynamic because the film elastically deforms the rolling surface. Depending on load, speed, temperature, tooth design, and tooth finish, any or all of these lubrication modes could exist in the same gear drive. The goal is to have full film lubrication present between meshing gear teeth. Unfortunately, often boundary lubrication is present and damage to the gear teeth results.

Even the most polished and smooth surfaces have microscopic rough spots called asperities. When two surfaces are in contact with each other, the load is carried by the asperities on the surfaces. During sliding, the total tangential force required to shear these asperity junctions is usually high, causing unacceptable friction, wear, and surface damage. To reduce the frictional force and thus allow easier sliding, a lubricant is deliberately introduced to separate the asperities either totally or partially. The use of liquid or gas lubricants is known as fluid-film lubrication. It may be noted that smoother or more highly finished the surfaces, the thinner the fluid film required to keep them separated.



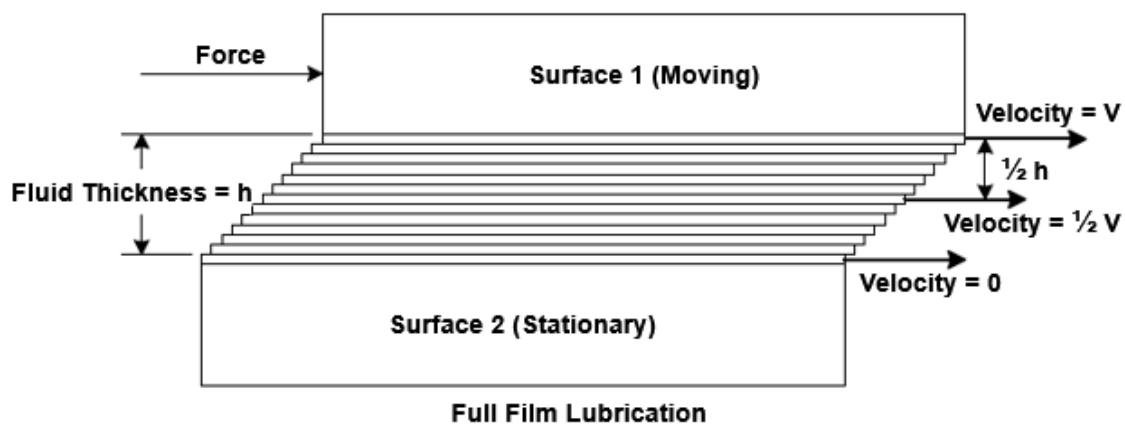
Full film lubrication, also known as thick film lubrication, refers to the total separation of asperities by a lubricant film thickness many times larger than the size of the lubricant molecules. If this condition exists only partially - that is, if part of the load is carried by the fluid pressure and the rest is borne by contacting asperities separated by a molecularly thin lubricant film - the term thin film (mixed) lubrication is used. In the most severe form of thin film lubrication, the entire load is carried by asperities lubricated by surface films of molecularly thin liquids, gases, or solids; this condition is known as boundary lubrication. Above figure

schematically shows the three lubrication regimes. The exclusive use of solid lubricants (graphite, molybdenum disulfide, etc.) is called solid lubrication.

Boundary Lubrication

Boundary lubrication most often is found at slow to moderate speeds, on heavily loaded gears, or on gears subject to high shock loads or where there are frequent starts and stops. This mode of lubrication exists when the oil film is not thick enough to prevent some metal to metal contact. This condition usually shows up as early wear and pitting on the teeth due to irregularities in the tooth surfaces. When boundary lubrication is encountered, oils having extreme pressure (EP) or anti wear (AW) additives should be used to minimize wear and possible scuffing. These additives cling to metal surfaces and form a sacrificial layer that protects the metal from wear.

Hydrodynamic Lubrication



Full film lubrication reduces friction between moving surfaces by substituting fluid friction for mechanical friction. Full film lubrication is illustrated in above figure. Surface 1 moves over surface 2 at velocity V , separated by a film of fluid with a thickness h . The oil film can be considered to be made up of many layers. The layer in contact with moving surface 1 clings to that surface and moves at the same velocity. Similarly, the layer in contact with surface 2 is stationary. The layers in between move at velocities directly proportional to their distance from the moving surface. For example, at a distance of $\frac{1}{2} h$ from surface 1, the velocity would be $\frac{1}{2} V$. The force required to move surface 1 across surface 2 is simply the force required to overcome the friction between the layers of fluid. This internal friction, or resistance to flow, is defined as the viscosity of the fluid.

To keep the surfaces separated, the fluid pressure must be high enough to support the load. In highly loaded bearings, like thrust bearings and horizontal journal bearings, relatively high fluid pressures are required to support the load. If this pressure is supplied by an outside source, it is called hydrostatic lubrication. If the pressure is generated internally (i.e., within the bearing by dynamic action), it is referred to as hydrodynamic lubrication.

Hydrodynamic lubrication usually only exists on higher speed gearing with very little shock loading.

Elastohydrodynamic Lubrication (EHL)

In instances of very high unit loads and high speeds, such as are experienced in antifriction bearings (ball and roller bearings), cams, and some gears, the lubricant is compressed and extremely high pressures are developed. The high pressures increase the lubricant's viscosity

and elastically deform the metal surfaces. This allows the load to be spread over a larger area and increases the load carrying capacity. This is referred to as elastohydrodynamic lubrication. In this regime of lubrication, an increase in load deforms the metal surfaces rather than affecting the oil film thickness because the oil film (the oil film thickness is usually less than 1.25 μm) is actually more rigid than the metal.

Elastohydrodynamic theory of lubrication is now accepted as very common in gear teeth. Elastohydrodynamic theory of lubrication makes possible the use of light oils in high speed drives and usually only occurs above 12,000 feet/minute (61 m/s) pitch line velocity.

Thin Film (Mixed) Lubrication

Thin film (mixed) lubrication is a cross between boundary lubrication and full film lubrication. While the bulk of the surfaces are separated by a lubricating layer, the asperities still make contact with each other. This is where the additives again come into play.

Methods of Lubrication

There are three gear lubrication methods in general use:

1. Grease lubrication
2. Splash lubrication (oil bath method)
3. Forced oil circulation method

There is no single best lubricant and method. Choice depends upon pitch line velocity (m/s) or rotating speed (rpm). At low speed, grease lubrication is a good choice. For medium and high speeds, splash lubrication and forced circulation lubrication are more appropriate. However, there are exceptions. Sometimes, for maintenance reasons, a grease lubricant is used even with high speed.

Ranges of Pitch Line Velocity (m/s) for Spur and Bevel Gears

No.	Lubrication	Range of Pitch Line Velocity (m/s)					
		0	5	10	15	20	25
1	Grease Lubrication	←-----→					
2	Splash Lubrication	←-----→					
3	Forced Circulation Lubrication	←-----→					

Ranges of Sliding Velocity (m/s) for Worm Gears

No.	Lubrication	Range of Sliding Velocity (m/s)					
		0	5	10	15	20	25
1	Grease Lubrication	←-----→					
2	Splash Lubrication	←-----→					
3	Forced Circulation Lubrication	←-----→					

Lubricants, Methods and Applicable Velocity Ranges

Above figure shows lubricants, methods and their applicable ranges of velocity.

Grease Lubrication

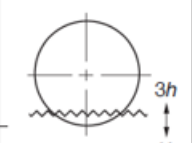
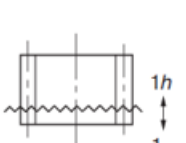
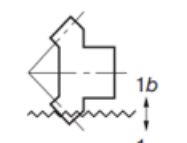
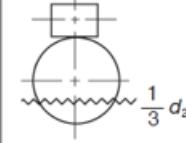
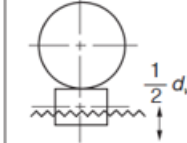
Grease lubrication is suitable for any gear system that is open or enclosed, so long as it runs at low speed.

The major points regarding use of grease are:

- Choosing grease with suitable viscosity is important. Grease with good fluidity is effective in an enclosed system.
- It is not suitable for use under high load and continuous operation. The cooling effect of grease is not as good as lubricating oil. So it may become a problem with temperature rise under high load and continuous operating conditions.
- There must be sufficient grease to do the job. However, too much grease can be harmful, particularly in an enclosed system. Excess grease will cause agitation, viscous drag and results in loss of power.

Splash Lubrication

Splash lubrication is used with an enclosed gear drive. In splash lubricated gear drives, the gears (sometimes an idler in mesh with the gear) are allowed to dip into the lubricant. The lubricant is picked up by fast moving gears and distributed to all gear tooth engagements and to troughs for lubrication of the shaft bearings. As the throwing action is quite violent, it distributes oil to every part of the housing. The housing contours are such that natural drainage for the oil is downward where troughs catch and distribute the oil to the bearings. Sometimes the oil is taken off the sides of gear by oil wipers (or scrapers) and delivered to the troughs.

Types of Gears	Spur Gears and Helical Gears		Bevel Gears	Worm Gears	
Gear Orientation	Horizontal Shaft	Vertical Shaft	Horizontal Shaft	Worm Above	Worm Below
Oil level					

h = Whole/Full tooth depth, b = Tooth width, d_2 = Pitch diameter of worm gear, d_w = Pitch diameter of worm
Adequate Oil Level

For splash lubrication proper oil level is important. Above figure shows the guide lines for proper oil level. Too low a level will result in poor or lack of lubrication. On the other hand, too high a level will result in unnecessary churning which wastes power, generates excessive heat, increases temperature and thins the oil. Increase in the temperature may accelerate degradation of the lubricant.

Splash systems are generally limited to pitch line velocities below 15 m/s for non-worm gears and 10 m/s sliding velocity for worm gears, since the lubricant may be thrown off at higher speeds. However, with the incorporation of appropriate design features (baffles and oil pans to reduce churning), splash systems have been successfully used in non-worm gear drives operating with pitch line velocities up to 25 m/s. To attain the desirable end result, the speed must be such that the centrifugal force exceeds gravity. Hence to be effective, splash lubrication needs at least 3 m/s pitch line velocity.

Forced Oil Circulation Method

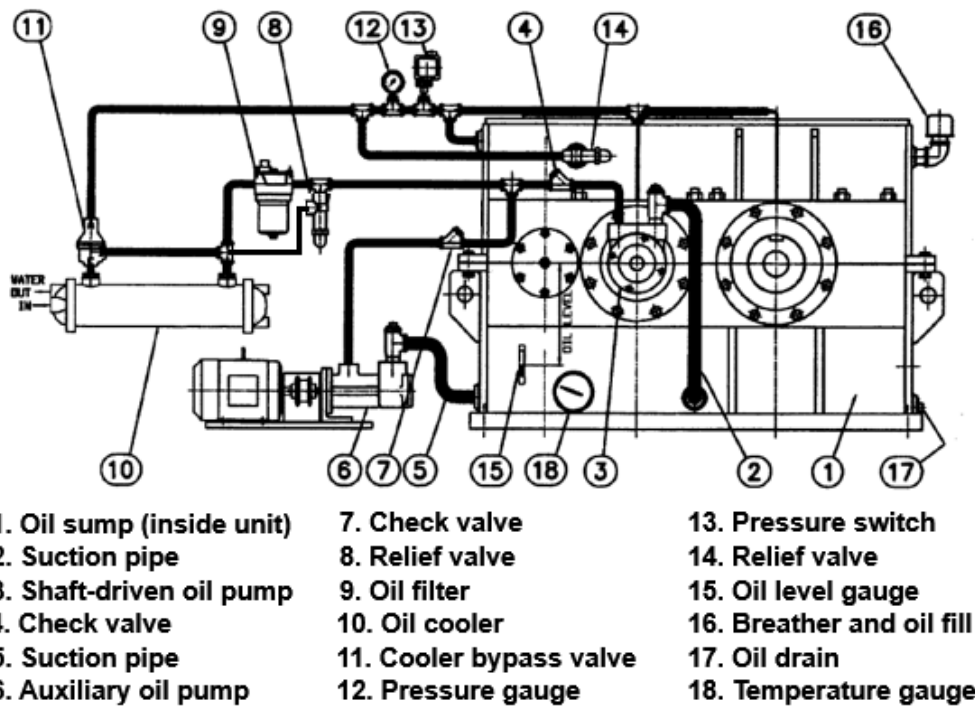
Forced/pressure oil circulation method applies lubricant to the contact portion of the teeth by means of an oil pump. There are drop, spray and oil mist methods of application.

In drop method, an oil pump is used to suck-up the lubricant and then directly drop it on the contact portion of the gears via a delivery pipe.

In spray method, an oil pump is used to spray the lubricant directly on the contact area of the gears.

In oil mist method, lubricant is mixed with compressed air to form an oil mist that is sprayed against the contact region of the gears. It is especially suitable for high speed gearing.

As shown in the following figure, oil tank, pump (either motor driven or gear drive's shaft driven or both), filter, piping and other devices are needed in the forced oil circulation method. Therefore, it is used only for special high speed or large gear box applications.



Forced Oil Circulation Method

In a typical forced oil circulation method, a shaft-driven oil pump or a motor-driven oil pump (generally called auxiliary oil pump) draws oil from the gearbox sump through a suction pipe. The oil from the pressure side of the oil pump is directed to a filter to clean the oil and through a cooler to cool the oil. A pressure relief valve is typically located before this filter to protect the system from too high operating pressure. If the filter becomes clogged, the relief valve will permit the unfiltered oil to bypass the filter so the gearbox will continue to receive lubricating and cooling oil albeit unfiltered (unfiltered oil is better than no oil). Another relief valve is often located at the inlet to the gearbox to limit the oil feed pressure if the system contains both a shaft-driven and a motor-driven pump, and both are running at the same time.

A shaft-driven oil pump is a pump that is driven by one of the rotating gear shafts of the gear drive. Some lubrication systems will include both a motor-driven and a shaft-driven oil pump. The motor-driven pump can be activated prior to gear drive start up to supply full oil flow requirement to the gear drive prior to shaft rotation, until the shaft-driven lube oil pump is running at a speed sufficient to supply full lubricating oil flow to the gear drive, during coast down of the gear drive or as a backup in case of failure of the main shaft-driven oil pump.

Check valves are located so that the main pump does not pump through the auxiliary system and that the auxiliary pump does not pump into the pressure side of the main oil pump.

A bypass is provided at the oil cooler, serving as both a pressure relief valve and/or a thermostatically controlled valve set so that the pressure drop across the cooler is limited during times when the oil is cold. Additionally, temperature and pressure sensors are located at various critical points throughout the system.

By filtering and cooling the circulating lubricant, the right viscosity and cleanliness can be maintained. This method is considered to be the best way to lubricate gears.

Design Considerations for Spray Method

(Source: Gear Materials, Properties, and Manufacture by ASM International®)

Above 1500 m/min (5000 ft/min), most gears are lubricated by a spray method. For gear drives with antifriction bearings, spraying the oil at the gear mesh only and relying on splash to lubricate the bearings is permissible up to a maximum pitch line velocity of 2100 m/min (7000 ft/min). Above this speed and for gear drives with journal bearings, both the gears and bearings should be force lubricated.

The oil jets should be placed on the incoming side of the gear mesh for pitch line velocities ≤ 2600 m/min (≤ 8000 ft/min). Above 2600 m/min (8000 ft/min), more oil is needed for cooling than for lubricating, and the oil flow removes heat best by being directed at the outgoing side of the gear mesh where the oil jets can strike the hot drive-side of the gear teeth. However, for very high-speed gears (> 5300 m/min, or 16,000 ft/min), there is a danger that the amount of oil carried to the incoming side of the gear mesh may be inadequate, and it is advised to add a supplementary flow at the incoming side of the gear mesh. Generally, about 2/3 of the oil flow should be supplied to the outgoing side of the mesh for cooling, and 1/3 of the flow directed at the incoming side for lubrication.

There are general guidelines, based on experience and experimentation, for specifying the quantity of flow, jet size, feed pressure and number of jets.

An empirical equation used to calculate the quantity of oil flow, q , is: $q = P / c$

where q is the oil flow rate (in gal/min), P is the transmitted power (in hp), and c is taken from the following table.

c, hp/(gal/min)	Flow Conditions	Applications
200	Copious	General industrial
400	Adequate	Typical aviation
800	Lean	Lightweight, high-efficiency aviation
1000	Starved	Only for unusual conditions

For a typical industrial application transmitting 150 kW (200 hp), where weight is not critical, the designer might choose a value of $c = 200$ hp/(gal/min), resulting in a copious flow of 4 L/min (1 gal/min). On the other hand, for a high-efficiency aviation application transmitting 150 kW (200 hp), where weight is critical, a value of $c = 800$ hp/(gal/min) might be chosen, resulting in a lean flow of 1 L/min (0.25 gal/min).

However, some applications may require different flow rates than those given in above table. For example, wide-face, high-speed gearing may require a higher flow rate to ensure uniform cooling and full-face coverage.

The proper jet size, feed pressure, and number of jets must be determined to maintain the proper flow rate, jet velocity, and full-face coverage.

The diameter of a jet can be calculated for a given flow rate and pressure based on the viscosity of the oil at the operating temperature. There are practical limitations on jet size, and the minimum recommended size is 0.8 mm (0.03 in.). If a jet smaller than this is used, contaminants in the oil may clog it. Typical jet diameters range from 0.8 to 3.0 mm (0.03 to 0.12 in.).

The feed pressure determines the jet velocity, which in turn determines the amount of oil that penetrates the gear mesh. Typical feed pressures range from 20 to 100 psig. Industrial application feed pressures are typically 30 psig, and high-speed aerospace applications are typically 100 psig. In general, the higher the pressure, the greater the cooling.

The number of jets should be sufficient to provide complete lubrication coverage of the face width. More than one jet for each gear mesh is advisable because of the possibility of clogging. The upper limit on the number of jets is determined by the flow rate and jet diameter.

Types of Lubricants

Generally, lubricants are classified into following three types/classes.

Inhibited Oils (RO)

These types of lubricants are commonly referred to as rust and oxidation inhibited, or R&O lubricants. They are formulated with highly refined petroleum or synthetic base oils and contain additives that enhance oxidation stability, provide corrosion protection, and suppress foam. Their superior oxidation stabilities typically set them apart from other gear oil types. However, their load carrying capabilities may be less than others. The low viscosity R&O oils, commonly called turbine oils, are generally associated with higher speed and lighter load applications.

Antiscuff/antiwear Oils (EP)

In addition to protection against corrosion and oxidation, antiscuff [extreme pressure (EP)] oils contain additives (sulfur, phosphorous, etc.) which provide protection against unacceptable wear and scuffing. These oils are formulated with refined petroleum or synthetic base oils. They are generally used in ISO VGs of 150 and above, and were developed to protect gears operating at high loads and severe impact or reversal conditions.

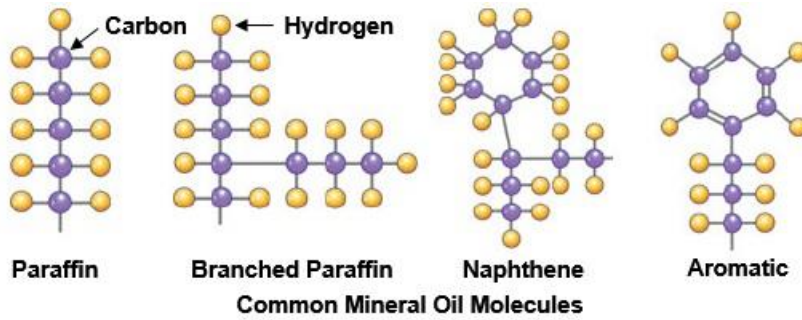
Compounded Oils (CP)

Compounded gear oils are a blend of petroleum base oils with three to ten percent of natural or synthetic fatty oils. These lubricants are frequently used in worm gear drives.

For minimum performance requirements of above types of oils, please see ANSI/AGMA 9005.

Mineral Oils and Synthetic Oils

A lubricating oil is composed of a base oil blended with various additives to enhance performance and maintain quality. The base oil may be a mineral oil or a synthetic oil.



Mineral oil comes from petroleum or crude oil and the quality depends on the refining process. Mineral oil is mainly made up of four different types of hydrocarbon molecules: paraffin, branched paraffin, naphthene and aromatic. Paraffinic oils have a long, straight chained structure, while branched paraffinic oils are the same with a branch off the side. Paraffin and branched paraffin are used mainly in engine oils, industrial lubricants, and processing oils. Naphthenic oils have a saturated ring structure and are most common in moderate temperature applications. Aromatic oils have a non-saturated ring structure and are used for manufacturing plastics, nylon fibers, epoxy resins, etc.

Synthetic oils do not originate from crude oil like conventional mineral oil. Synthetic oils are man-made fluids. Synthetic oils are formulated from derivatives of natural gas and other base materials. For instance, synthetic hydrocarbons [polyalphaolefins (PAOs)], which are among the most common synthetic base oils, are formulated from ethylene and decene (largely derived from natural gas). Synthetic oils have identical straight chained structures, much like the branched paraffinic oils. One of the benefits of synthetic oil is that the molecular size and weight are constant while in mineral oils they vary greatly; therefore, their properties are very predictable.

However, the risk with synthetic oils is the compatibility issues that come with using these lubricants. Some synthetics can dissolve seal materials, allowing leaks. Many have compatibility issues with other fluids. Polyalkylene glycol (PAG) base oils are notorious for their incompatibility with mineral oils, although oil-soluble PAGs (usually called OSPs) largely remedy this drawback. In the event that mineral oils and PAG oils form an incompatible mix, the result is a gelatinous mass that clogs lines and can lead to lubricant starvation and ultimately machine failure. Other incompatibilities for synthetics include paints, hose materials and some additives.

The best quality mineral oil is mostly made up of paraffinic oils. So, in many applications, mineral oil is just as good as synthetic, and in these applications is most likely the preferred base due to synthetic's high cost, toxicity, solubility, incompatibility, and hazardous disposal. However synthetic oil may be required in extreme applications where a high flash point, low pour point, fire resistance, thermal stability, high shear strength, or high viscosity index is needed.

Lubricant Selection

When one thinks of gear lubrication, the primary concern is usually about the gears. However, in addition to the gears themselves, there are many other components also to be served by the lubricant in the gearbox. Therefore, in selecting the correct lubricant for a gear drive, consideration should also be given to the bearings, seals, other associated components and auxiliary equipment, e.g., pumps and heat exchangers that may be affected by the choice of the lubricant.

To minimize the friction and wear between surfaces in relative motion, and to remove heat generated by the mechanical action of the system, the lubricant must have sufficient viscosity to separate the mating surfaces as much as possible, and also have the appropriate additives to minimize thermal and oxidative degradation, and provide antiwear and antiscaff performance for transient peak operating situations.

The gear lubricant should possess the following properties.

Properties	Description
Proper Viscosity	Lubricant should maintain a proper viscosity to form a stable oil film at the specified temperature and speed of operation.
Antiscuffing Property	Lubricant should have the property to prevent the scuffing failure of tooth surface under high pressure of load.
Oxidation Resistance and Thermal Stability	A good lubricant should not oxidize easily and must perform in moist and high temperature environment for long duration.
Demulsibility	Demulsibility, also known as water separation, is the ability of a lubricant to separate from water. Moisture tends to condense due to temperature change when the gear drives are stopped. The lubricant should have the property of separating moisture and water from the lubricant.
Antifoaming Property	Antifoaming property is a vital requirement. If the lubricant foams under agitation, it will not provide a good oil film.
Anticorrosion Property	Lubricant should be neutral and stable to protect steel and cupric metal parts from corrosive attack.

Note: For more information on lubricant properties, please see article on lubricant properties.

Following points may be considered for a lubricant selection.

- Type of gearing used: spur, helical, bevel gears or worm gears.
- Materials of construction of gear drive components, such as: gears, bearings, oil seals, etc.
- Operating conditions, such as: ambient temperature, operating oil temperature, minimum and maximum pitch line velocities.
- Special circumstances, such as: start up at low temperature, ambient temperatures above 50°C, high transient loads.

In general, the gears may be exposed to an ambient temperature range of – 40°C to + 55°C. The air temperature in the immediate vicinity of the gear drive is taken as the ambient temperature. The type of lubricant and its viscosity grade will be determined, in part, by the ambient temperature.

Gear drives operating in cold areas must be provided with oil that circulates freely and does not require high starting torques. It is recommended that the gear oil should have a pour point at least 5°C lower than expected minimum ambient start up temperature. In view of this, the viscosity of the lubricant must be low enough to allow the oil to flow freely at the start up temperature, but high enough to carry the load at the operating temperature. If suitable low temperature gear oil is not available, the gear drive must be provided with a sump heater to bring the oil up to a temperature at which it will circulate freely at starting. To avoid excessive localized heating, which could result in rapid degradation of the lubricant, a rating of 0.8 watts/cm² of heater surface area is recommended for most applications.

Provision should be made for cooling the lubricant where normal continuous operation of the gear drive would raise lubricant over its recommended temperature.

Based on above points, one can estimate the appropriate viscosity for the particular application based on the effective operating temperatures the gears will see in service.

Please note that the allowable maximum oil sump temperature for a given application is dependent on the choice of base oil type and additive chemistry. Consult the lubricant supplier for specifics on the oil being chosen. Many lubricants are unstable above their stated maximum temperature.

In general, the recommendations of the gear manufacturer should be followed, when available, in selecting a gear lubricant. In the absence of such information, for determining lubricant type and viscosity, following tables may be used.

Determining Lubricant Type Based On Application

The following table provides general guideline to choose type of lubricant to be used in a given application.

Operation of Driving Unit	Operation of Driven Unit			
	Uniform	Light Shocks	Moderate Shocks	Heavy Shocks
Uniform	RO	RO/EP	EP	EP
Light Shocks	RO	RO/EP	EP	EP
Moderate Shocks	EP	EP	EP	EP
Heavy Shocks	EP	EP	EP	EP

Note:

1. RO are inhibited oils and EP are antiscuff/antiwear oils.
2. Compounded oils are not included in this table because they are generally used for worm gear applications.

Following tables provide examples of operation of driving units and driven units.

Mode of Operation	Driving Unit
Uniform	Electric motor, steam or gas turbine operating uniformly, i.e., low, infrequent starting torques
Light Shocks	Steam or gas turbine, hydraulic or electric motor with high, frequent starting torques
Moderate Shocks	Multi cylinder combustion engine
Heavy Shocks	Single cylinder combustion engine

Mode of Operation	Driven Unit
Uniform	Power generators, uniformly fed conveyors or apron feeders, lightweight elevators, packaging machines, feed drives of machine tools, fans, lightweight centrifuges, rotary pumps, agitators and mixers for light fluids or substances of uniform density, cutters, presses, punches, rotary units, drive units
Light Shocks	Intermittently fed conveyors or apron feeders, main drive of machine tools, heavy elevators, rotary units of cranes, industrial and mining fan systems, heavy centrifuges, rotary pumps, agitators and mixers for viscous fluids or substances of varying density, multi cylinder piston pumps, feeding pumps, extruders in general, calenders, rotary kilns, rolling mills
Moderate Shocks	Rubber extruders, intermittently operating mixers for rubber and synthetic materials, light weight ball mills, wood working machines, blooming mills, lifting units, single cylinder piston pumps
Heavy Shocks	Excavators, bucket wheel and chain drives, screen drives, dredging shovels, rubber kneaders, stone and ore crushers, mining machinery, heavy feed pumps, rotary drilling installations, brick presses, debarking drums, peeling machines, cold belt rolling mills, briquette presses, edge mills

Viscosity Grade Selection

Viscosity is one of the most important lubricant properties, and the higher the viscosity, the greater the protection against the various gear tooth failures. However, the viscosity must be limited to avoid excessive heat generation and power loss from churning and shearing of the lubricant by high speed gears or bearings. The operating temperature of the gear drive determines the operating viscosity of the lubricant. If the lubricant is too viscous, excessive heat is generated. The heat raises the lubricant temperature and reduces its viscosity, reaching a point of diminishing returns where increasing the starting viscosity of the lubricant leads to a higher operating temperature and a higher oxidation rate, without a significant gain in operating viscosity.

Gear teeth contact areas are relatively small; therefore, the contact unit pressures are comparatively high. It is therefore important to provide a film of lubricant of sufficient strength to withstand the localized pressure during the period of tooth contact. The pitch line velocity of the gear set governs the length of time the oil film must withstand the pressures. When speeds are high, the gear-tooth contact time is short and the loads usually not so heavy, and as a result, a lubricant of a relatively low viscosity may be used. At the lower speeds the loads are generally high and the tooth contact time longer, and consequently a lubricant, of a higher viscosity should be used. Where high speed and low speed gearing are contained in the same gearbox and lubricated from the same source of lubricant, the lubricant viscosity required for the lower speed gearing should be selected. Consideration should be given to the viscosity requirements of the bearings in these instances.

In the absence of a rigorous EHD analysis, the following tables may be used to select an appropriate viscosity grade for an application. The viscosity selection must be complemented with an appropriate performance additive to provide a finished lubricant with properties sufficient to meet the overall needs of the application.

Following table, for spur, helical and bevel gears, provides the estimated ISO viscosity grades for a given operating temperature – pitch line velocity combination for lubricant having viscosity index of 90 (This table shows viscosity grades at bulk oil operating temperatures from 60 to 90°C only. For viscosity grades at other temperatures, please see Table B-1, Annex B of ANSI/AGMA 9005-E02). The guidelines are applicable to gears which operate at or below 3600 revolutions per minute, or a pitch line velocity of not more than 40 meters per second, or both. For lubricant having viscosity index of 120, 160 and 240, please see Annex B of ANSI/AGMA 9005-E02. For the purpose of estimating, the operating temperatures were assumed to be nominally 45°C above the ambient temperature.

Viscosity grade at bulk oil operating temperature for oils having a viscosity index of 90 [Table B-1 (part), Annex B of ANSI/AGMA 9005-E02]								
Temp °C	Pitch line velocity, m/s							
	1.0 - 2.5	2.5	5.0	10.0	15.0	20.0	25.0	30.0
60	460	320	220	150	68	68	68	46
65	680	460	320	220	150	100	100	68
70	1000	680	320	220	150	100	100	68
75	1500	680	460	320	220	150	150	100
80	2200	1000	680	460	220	220	220	150
85	3200	1500	1000	460	320	220	220	150
90	3200	2200	1000	680	460	320	320	220

Note: Review anticipated cold start, peak and operating temperatures, service duty and range of loads when considering viscosity grades shown in the shaded cells.

Following table provides guidelines for cylindrical worm gearing. The guidelines are applicable to worm gears which operate at or below 2400 rpm (worm speed) or 10 meters per second sliding velocity. The table applies to lubricants with viscosity index of 100 or less. For lubricants with viscosity index greater than 100, wider temperature ranges may apply.

ISO viscosity grade guidelines for enclosed cylindrical worm gear drives (As per Table B-5, Annex B of ANSI/AGMA 9005-E02)			
Pitch line velocity of final reduction stage	Ambient temperature, °C		
	-40 to -10	-10 to +10	+10 to +55
Less than 2.25 m/s	220	460	680
Above 2.25 m/s	220	460	460

Note:

If a gear drive is to be used outdoors and the ambient temperature vary considerably between summer and winter, it is recommended that the oil be changed to the proper viscosity for each season.

Guidelines as per ANSI/AGMA 9005-D94

(Extracted from ANSI/AGMA 9005-D94, *Industrial Gear Lubrication* with the permission of the publisher, the American Gear Manufacturers Association, 1500 King Street, Suite 201, Alexandria, Virginia 22314)

In ANSI/AGMA 9005-D94, for ambient temperature - pitch line velocity combination, AGMA lubricant numbers guidelines were provided as given below.

AGMA lubricant number guidelines for enclosed helical, herringbone, straight bevel, spiral bevel and spur gear drives:

Pitch line velocity of final reduction stage	AGMA lubricant numbers, ambient temperature °C °(F)			
	-40 to -10 (-40 to +14)	-10 to +10 (14 to 50)	10 to 35 (50 to 95)	35 to 55 (95 to 131)
Less than 5 m/s (1000 ft/min)	3 S	4	6	8
5-15 m/s (1000-3000 ft/min)	3 S	3	5	7
15-25 m/s (3000-5000 ft/min)	2 S	2	4	6
Above 25 m/s (5000 ft/min)	0 S	0	2	3

Notes:

1. AGMA lubricant numbers listed above refer to R & O and synthetic gear oil. EP or synthetic gear lubricants in the corresponding viscosity grades may be substituted where deemed acceptable by the gear drive manufacturer.
2. Special considerations may be necessary at speeds above 40 meters per second (8000 feet per minute).
3. Variations in operating conditions such as surface roughness, temperature rise, loading, speed, etc., may necessitate use of a lubricant of one grader higher or lower. Contact gear drive manufacturer for specific recommendations.

AGMA Lubricant number guidelines for enclosed cylindrical worm gear drives:

Pitch line velocity of final reduction stage	AGMA lubricant numbers, ambient temperature °C °(F)			
	-40 to -10 (-40 to +14)	-10 to +10 (14 to 50)	10 to 35 (50 to 95)	35 to 55 (95 to 131)
Less than 2.25 m/s (450 ft/min)	5 S	7 Comp	8 Comp	8 S
Above 2.25 m/s (450 ft/min)	5 S	7 Comp	7 Comp	7 S

Note:

Worm gear applications involving temperatures outside the limits shown above, or speeds exceeding 2400 rpm or 10 m/s (2000 ft/min) sliding velocity, should be referred to the

manufacturer. In general, for higher speeds a pressurized lubrication system is required along with adjustments in recommended viscosity grade.

If the product is a mineral oil that contains only rust and oxidation (R&O) additives, it will be recognized with only the AGMA number. If it is a mineral oil with extreme pressure additives, it is recognized with the AGMA number followed by the “EP” designation. AGMA numbers followed by an “S” denote synthetic gear oils. Compounded gear oils contain 3% to 10% fatty or synthetic fatty oils and are noted by the AGMA number with “Comp” after it. Some gear oils contain residual compounds called diluent solvents that are used to temporarily reduce the viscosity making it easier to apply. In this case, the AGMA number is followed by an “R,” which describes product prior to addition of diluents solvent.

Following table (as per ANSI/AGMA 9005-E02) gives ISO viscosity grade classifications equivalents to former AGMA grade.

ISO Viscosity Grade	Mid-point Viscosity at 40°C, mm ² /s	Former AGMA Grade Equivalent
32	32	0
46	46	1
68	68	2
100	100	3
150	150	4
220	220	5
320	320	6
460	460	7
680	680	8
1000	1000	8A
1500	1500	9
2200	2200	10
3200	3200	11

Various tables given in this section were extracted from ANSI/AGMA 9005-E02, Industrial Gear Lubrication, with the permission of the publisher, the American Gear Manufacturers Association, 500 Montgomery Street, Suite 350, Alexandria, Virginia 22314.

Lubrication of Open Gearing

Open gearing is similar to enclosed gearing except, as the name implies, these gears are not enclosed within a housing. These gears can be of any type, but commonly are spur and helical gears.

Because open gears have a tendency to throw off conventional oils such as those used with most enclosed drives, much higher viscosity lubricants are used for their lubrication. The lubricant is sometimes applied by hand or brush to slow speed and coarse pitch unprotected gears. The frequency of application may be determined by observation.

Intermittent mechanical spray systems are generally used with open gearing and depend on the use of heavy oil, grease, or residual compounds which will remain on the gear teeth through several revolutions. The spray is activated automatically or by hand at certain timed intervals. The spraying time should equal the time for one or preferably two revolutions of the gear to ensure complete coverage. Periodic inspections should be made to ensure that sufficient lubricant is being applied to give proper protection. Generally, two hours is the maximum interval permitted between applications of lubricant. However more frequent application of small quantities is preferred.

Open gears can also be lubricated continuously by splash lubrication, pressure fed lubrication or idler immersion methods and intermittently by gravity feed or forced drip methods. For minimum viscosity recommendations for open gear lubrication methods, please see Annex D of ANSI/AGMA 9005-E02.

Mineral Oil Based EP Lubricating Oils

Following table lists brand names of mineral oil based antiscuff/antiwear (EP) lubricating oils for gears.

Company Name	Brand Name of Mineral Oil Based Antiscuff/antiwear Lube Oils (EP)
Indian Oil	Servomesh SP Servomesh Plus
Bharat Petroleum	MAK AMOCAM
Hindustan Petroleum	PARTHAN EP
Castrol	Alpha SP
Exxon Mobil	Mobilgear 600 XP Spartan EP
Shell	Omala

Caution

EP gear lubricants should never be used unless approved by the gear manufacturer or the backstop manufacturer in industrial gear drives that use internal backstops, such as those found on conveyor belts, or in the lubrication of cooling tower gear drives that employ ratchets. The EP chemistry will not allow the clutch mechanisms to properly engage, resulting in the mechanism slipping. This can cause serious safety hazard.

After Market Additives

There are a number of after-market lubricant additives that are being marketed as solutions to many lubricating problems. These additives may contain teflon or some other “secret ingredient” that supposedly imparts improved lubricating qualities to the lubricant. There may be cases where these additives improve performance in some way, or at least appear to improve performance, but in most cases their usefulness is questionable at best. These additives may actually reduce a lubricant’s effectiveness by reacting with some of the additives already in the oil.

The major lubricant manufacturers spend a great deal of time and money formulating their products to provide optimum performance for particular applications. If some additive is available that will improve a lubricant to the extent claimed by many of the after-market additive distributors, most lubricant manufacturers would have added it to their product.

If a lubricant is not performing as it should, a different lubricant may be required, or some problem may exist in the gear drive/system. Before adding anything to a lubricant, the lubricant’s manufacturer should be consulted. The lubricant manufacturer can provide information on the possible benefits or consequences of the additive and determine whether a different lubricant is required.

Lubrication of High Speed Gear Units

High speed gear units shall be designed with a pressurized lubrication supply system to provide lubrication and cooling to the gears and bearings.

For more information on lubrication of high speed gear units, please see ANSI/AGMA 6011-103, Specification for High Speed Helical Gear Units, published by the American Gear Manufacturers Association, 500 Montgomery Street, Suite 350, Alexandria, Virginia 22314, <http://www.agma.org>.

This standard includes design, lubrication, bearings, testing and rating for single and double helical external tooth, parallel shaft speed reducers or increasers. Units covered include those operating with at least one stage having a pitch line velocity equal to or greater than 35 meters per second or rotational speeds greater than 4500 rpm and other stages having pitch line velocities equal to or greater than 8 meters per second.

Lubricant Properties

The physical properties of a lubricant, such as viscosity and pour point, are largely derived from the base stock(s) from which they are produced. While viscosity is the most common property associated with a lubricant, there are many other properties that contribute to the characteristics of the finished product. The properties of finished gear lubricants result from a combination of base stock and additives. Information about properties a gear lubricant should have is given in this chapter.

A gear lubricant should have the following properties.

Viscosity

The main function of a lubrication oil is to create and maintain a lubrication film between two moving metal surfaces to avoid metal to metal contact, which leads to destructive wear. The oil property that governs the thickness of the lubrication film is its viscosity. Hence, for any oil lubrication system, viscosity is considered as its most important property.

Viscosity can be defined as measurement of fluid's internal resistance to flow at a specified temperature. The higher the viscosity of a fluid, the greater the internal resistance and the greater its load capacity. But with the higher internal resistance, temperatures can rise.

The viscosity of a liquid lubricant may be determined using various test methods. Historically, these have included the Saybolt, Kinematic, Engler and Redwood test methods. With these test methods, viscosity is reported as Saybolt Universal Seconds (SUS), mm²/s (cSt or centistoke), degrees Engler and Redwood, respectively.

There are two ways to measure a fluid's viscosity, namely Dynamic (Absolute) Viscosity and Kinematic Viscosity.

Dynamic (Absolute) Viscosity

Dynamic Viscosity is defined as a fluid's resistance to flow, or the fluid's resistance to deform when subjected to a force. An easy way to visualise this is to imagine stirring two liquids, water and honey, in two separate containers. The honey provides more resistance to shear forces through the stirring process compared to water, and is said to have higher dynamic viscosity compared to water. Dynamic viscosity is commonly reported in Centipoise (cP).

$$1 \text{ cP} = 1 \text{ mPa}\cdot\text{s}$$

The most common method to measure dynamic viscosity is the use of Brookfield viscometer (it employs a rotating spindle that experiences torque as it rotates against fluid friction) as per ASTM D2983.

Kinematic Viscosity

Kinematic viscosity is defined as a fluid's resistance to flow. AGMA (American Gear Manufacturers Association) as well as many national standards organizations have adopted the ISO 3104 / ASTM D445 method for measuring kinematic viscosity.

In this method, the time is measured for a fixed volume of liquid lubricant to flow under gravity through a calibrated capillary viscometer at a given temperature, usually 40°C and 100°C. The kinematic viscosity is then calculated by multiplying the measured flow time by the calibration

constant for that viscometer. The viscosity is then reported in centistokes (cSt) at a given temperature (Industrial oils at 40°C and Automotive oils at 100°C).

1 cSt = 1 mm²/s.

It may be noted that most laboratories report viscosity as Kinematic Viscosity, whilst most onsite equipment reports in Dynamic Viscosity units. However, both Dynamic and Kinematic Viscosity are interchangeable by using the following formula.

Dynamic Viscosity (cP) = Kinematic Viscosity (cSt) x Fluid Density (kg/m³)

Viscosity Grades

The International Organization for Standardization (ISO) classifies and specifies fluid lubricants using kinematic viscosity. Many national standards organizations have adopted the ISO 3104/ASTM D445 method of measuring and specifying fluid viscosity. For fluid classification purposes, the viscosity of a lubricant is stated at a standard temperature, e.g., ISO 3448 specifies viscosity at 40°C.

The ISO viscosity classification consists of a series of 20 viscosity brackets between 1.98 mm²/s and 3520 mm²/s, each of which is defined by a number. The numbers indicate to the nearest whole number, the mid-points of their corresponding brackets. For example, ISO viscosity grade 32 relates to the viscosity bracket 28.8 to 35.2 mm²/s, the mid-point of which is 32.0 mm²/s. This is illustrated in the table given below, which shows the ISO viscosity grade numbers, the mid-points of each bracket, and the viscosity limits. This system is now used to classify all industrial lubricating oils where viscosity is an important criterion in the selection of the oil. The higher the number, the thicker the oil.

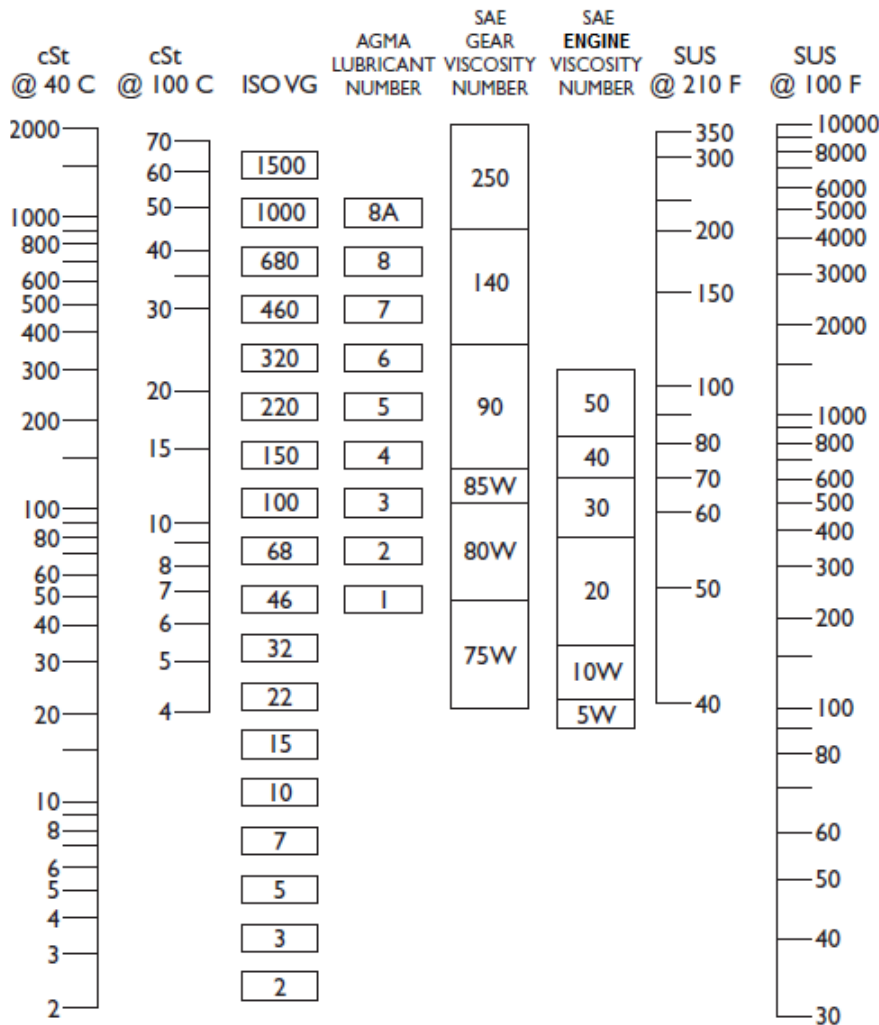
ISO Viscosity Grades for Industrial Oils			
ISO Viscosity Grade (ISO VG)	Kinematic Viscosity at 40°C (mm ² /s)		
	Minimum	Maximum	Mid-point
2	1.98	2.42	2.20
3	2.88	3.52	3.20
5	4.14	5.06	4.60
7	6.12	7.48	6.80
10	9.00	11.0	10.0
15	13.5	16.5	15.0
22	19.8	24.2	22.0
32	28.8	35.2	32.0
46	41.4	50.6	46.0
68	61.2	74.8	68.0
100	90.0	110	100
150	135	165	150
220	198	242	220
320	288	352	320
460	414	506	460
680	612	748	680
1000	900	1100	1000
1500	1350	1650	1500
2200	1980	2420	2200
3200	2880	3520	3200

The American Gear Manufacturers Association (AGMA) has also created a commonly used viscosity classification system. The AGMA numbers let the user know the ISO viscosity grade and some basic information about the gear lubricant's chemistry. If the product is a mineral oil

that contains only rust and oxidation (R&O) additives, it will be recognized with only the AGMA number. If it is a mineral oil with extreme pressure additives, it is recognized with the AGMA number followed by the "EP" designation. AGMA numbers followed by an "S" denote synthetic gear oils. Compounded gear oils contain 3% to 10% fatty or synthetic fatty oils and are noted by the AGMA number with "Comp" after it. Some gear oils contain residual compounds called diluent solvents that are used to temporarily reduce the viscosity of heavy oils for ease of application. The diluent evaporates after application leaving a thick film of lubricant on the gear teeth. In this case, the AGMA number is followed by an "R," which describes product prior to addition of diluent solvent (Viscosity of the base compound without diluent).

AGMA Lubricant Number	Viscosity Ranges, cSt at 100°C (212°F)
14R	428.5 to 857.0
15R	857.0 to 1714.0

The Society of Automotive Engineers (SAE) has created two viscosity standards for automotive lubricants. SAE J300 is a viscosity classification for engine (crankcase) lubricating oils, and SAE J306 is for gear (axle and manual transmission) lubricating oils.



Comparison of Viscosity Classifications

- ISO and AGMA viscosities are specified at 40°C. SAE 5W, 10W, 75W, 80W and 85W viscosities are specified at low temperature. Equivalent viscosities at 100°F and 210°F are shown.
- SAE 90-250 (Gear Oils) and SAE 20-50 (Engine oils) are specified at 210°F/99°C.

Above figure compares lubricating oils rated by different standards.

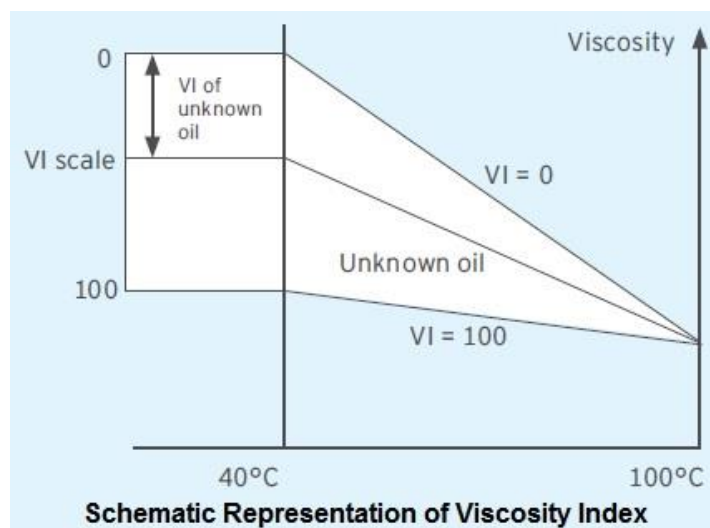
Viscosity Index (VI)

The viscosity of lubricating oil changes with temperature (Viscosity decreases as temperature increases, and vice versa) and the rate of change depends on the composition of the oil. Naphthenic base oils change more than paraffinic base oils. Certain synthetic lubricants change much less than paraffinic oils.

To assess this lubricating oil property, the American Society for Testing and Materials (ASTM) created a method to provide a number called the Viscosity Index (VI).

The viscosity index is an arbitrary number indicating the effect of change of temperature on the kinematic viscosity of an oil. A high viscosity index signifies a relatively small change of kinematic viscosity with temperature. Generally, the higher the VI, better the oil.

The viscosity index correlates the amount of viscosity change for a given oil, compared to two reference oils having the highest and lowest viscosity indices at the time the VI scale was first introduced (1929). A standard paraffinic oil was given a VI of 100 and a standard naphthenic oil a VI of 0.



Above figure (© 2007–2011 Chevron U.S.A.) shows the relationship between viscosities at 40°C and 100°C. The method has been updated and revised several times to include VI values higher than 100.

A lubricant's VI may be a natural physical property of the base fluid, or the result of chemical enhancement through the addition of a VI improver. VI improvers are typically polymers. When considering the use of a gear oil with a high VI, it is important to select a product that retains its VI over the expected drain and change interval.

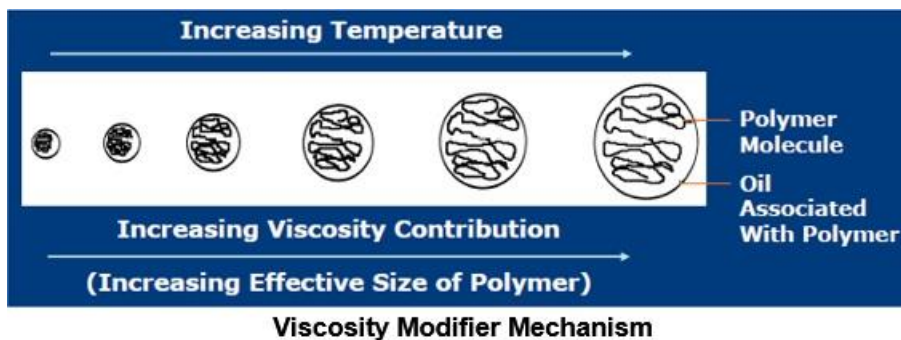
A gear oil with a low VI may be acceptable in applications where ambient start up temperatures remain constant, the drive operating temperature varies minimally, or the drive operating temperature is close to the ambient temperature. The use of a gear oil with a VI greater than 120 may be desirable in applications where ambient start up temperatures are much lower than normal operating environment or when ambient temperatures vary widely.

The viscosity index (VI) is internationally determined using ISO 2909. It describes procedures for calculating the viscosity index (VI) of petroleum products and related materials, such as lubricating oils, from their kinematic viscosities at 40°C and 100°C.

The procedure for the calculation of the viscosity index is also given in ASTM Method D 2270-74 for calculating viscosity index from kinematic viscosity at 40°C and 100°C.

The tables by ASTM – ASTM data series publication DS 39B permits direct reading of the viscosity index of a petroleum product or lubricant if its kinematic viscosities at 40°C and 100°C are known. Use of these tables eliminates the calculations which would be required to determine the viscosity index.

Viscosity Modifier Mechanism



As shown in above figure, viscosity modifier additives are long polymer molecules which uncoil at high temperatures to increase viscosity, while at low temperatures they form tight “balls” which no longer contribute much to viscosity.

Caution: Viscosity improvers do not last forever in a blend. They can be chopped up or “sheared down” by constant mechanical motion in the oil.

Viscosity Considerations

The lubricant should maintain a proper viscosity to form a stable oil film at the specified temperature and speed of operation.

Equipment loads must be considered when selecting the viscosity. Under a heavy load, the lubricant film is squeezed or compressed. Therefore, a higher viscosity lubricant is needed. The higher the viscosity, the more film strength the lubricant will generally possess.

Generally, the least viscous lubricant which still forces the two moving surfaces apart is desired. If the lubricant is too viscous, it will require a large amount of energy to move and a corresponding increase in temperature; if it is too thin, the surfaces will come in contact and wear will increase.

Pour Point

Small amounts of dissolved wax are usually present in petroleum lube oils. As temperature decreases, wax crystals start to appear and eventually solidifying the oil at a lower temperature (the Pour Point).

Pour point is an indicator of the lowest temperature at which an oil flows under the influence of gravity. In paraffinic oils, the pour point is the result of the crystallization of waxy particles.

In naphthenic oils, the pour point is the result of the decrease in viscosity caused by a decrease in temperature. However, pour point should not be used as the only indicator of the low temperature limit at which a lubricant may function satisfactorily because initial agitation by gears, bearings or a pump can break down the crystal wax structure of paraffinic oils and allow the gear oil to flow or to be pumped at temperatures well below its pour point.

ISO 3016 / ASTM D97 is used to determine pour point. It is recommended that the pour point of the oil should be at least 5°C lower than the minimum ambient temperature expected for an application.

Lubricity

Lubricity, also referred to as oiliness, with respect to lubricating oil, is defined as the ability of an oil to reduce friction between moving surfaces. Lubricity additives, usually vegetable or animal fats, enhance lubricity by tenaciously adhering to the metal's surface, forming an adsorbed film of high lubricating value.

Antiwear

Lubricant should prevent wear of the metal surfaces. Antiwear additives work by coating a metal's surface. If light metal-to-metal contact is made, the heat from the friction melts the additives, forming a liquid layer between the surfaces. This molten additive layer, being softer than the metal, acts as a lubricant, preventing wear of the metal surfaces.

Antiscuffing

Scuffing is adhesive wear. Other terms used for adhesive wear are scoring, galling and seizing. Adhesive wear involves transfer (loss) of material from one surface to another as a result of a welding or tearing process during interaction between the contacting surfaces. It occurs in lubricated applications due to loss of supporting film and/or inadequate protection of the surface(s). To provide antiscuffing property, EP (Extreme Pressure) additives are added to the base oil.

EP additives work by reacting with a metal to form a compound that acts as a protective layer on the metal's surface. Because this layer is softer than the metal itself, under extreme pressure conditions, the compound layer wears away first, protecting the metal. As this layer is removed, the EP additive acts to form another layer. In contrast to the action of antiwear additives, EP additives control wear instead of preventing it. Some EP additives, because of their reactive nature, can be corrosive to brass or copper-containing alloys. To prevent excessive corrosion, most EP additives are activated by the heat of friction created during extreme pressure conditions but do not react at room temperature.

Oxidation Resistance and Thermal Stability

Oxidation is a chemical process in which oxygen combines with the free radicals within a lubricant to produce acids that can corrode metals and polymers that produce sludge. Oxidation also leads to an increase in viscosity. Oxidation is enhanced by elevated temperature and the presence of catalyst like iron, copper, water or foreign matter.

Operating temperature is normally the most influential factor impacting the rate of oxidation. It has been observed that for every 10°C (18°F) increase above 80°C, an oil's oxidation rate doubles, which means the oil's life is essentially cut in half. Hence, sustained lubricant sump temperature approaching 95°C may require more frequent lubricant changes. In any gear drive, localized heating (for example, instantaneous frictional heat at the mesh point of the

gear teeth) must also be taken into account in addition to bulk lubricant operating temperature. The areas of localized heating can be sites where accelerated oxidative aging and thermal decomposition occurs.

Thermal stability of a lubricant indicates its relative chemical stability due to thermal stress. A thermally unstable compound can decompose in response to heat alone. Thermal decomposition, like oxidation, may be catalyzed by metals, water, or other chemical compounds. Thermal breakdown products may themselves be reactive and promote oxidation, corrosion, or sludge formation.

Antirust and Anticorrosion

In general, good gear lubricants are not corrosive to steel or copper containing alloys. Rust inhibitors protect ferrous (iron or steel) parts by forming a film on the part that resists attack by water. Corrosion inhibitors act in a similar way to protect nonferrous parts and also act to neutralize acids with a basic compound such as calcium carbonate.

The corrosion test methods are intended to measure the ability of a lubricant to prevent corrosion on a metal surface in contact with oil.

The ISO 2160 / ASTM D130 copper corrosion test method measures the corrosive nature of lubricating oil on a copper strip that is immersed under static conditions in the oil.

The ISO 7120 / ASTM D665 test method evaluates the ability of an oil to prevent the rusting of ferrous parts in the event water becomes mixed with the oil. The method consists of two parts: Procedure A uses distilled water, and Procedure B uses synthetic seawater. In this test, 10% water (distilled or synthetic seawater) is mixed in the oil and a polished ASTM 1018 grade carbon steel rod is immersed in the stirred mixture for 24 hours at 60°C. If there is no rust on the specimen, the oil passes the test. AGMA 9005-E02 requires gear oils to pass Procedure B.

Demulsibility

Demulsibility, is the ability of a lubricating fluid to separate from water. The demulsibility test is generally carried out as per ASTM D2711.

Antifoaming

Foaming in a gear oil can be detrimental to the performance and durability of the gear drive as it may lead to lubricant starvation. Foaming in a lubricant is controlled by using foam inhibitor. These additives cause the foam to dissipate more rapidly by promoting the agglomeration of small bubbles into large bubbles which burst more easily.

Cleanliness

Although lubricants are produced under relatively clean conditions, their handling provides many opportunities for contaminants to be introduced into the lubricant before it is used in machinery. Hence, an assessment of a lubricant's cleanliness prior to putting it into service is highly recommended.

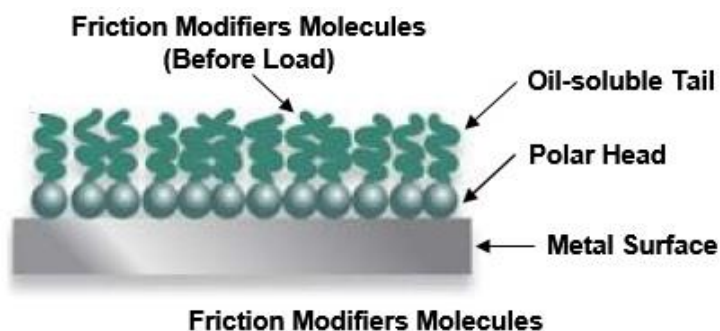
During a lubricant's service life, it may have opportunity to be contaminated with particulate matter from poor maintenance practices, the operating environment (airborne particulates), the machinery (wear), and oil degradation byproducts. Cleanliness assessments of lubricants while they are in service can enable the end user to take appropriate corrective action.

While many methods have been used to define lubricant cleanliness, the current practice is to use ISO 4406 cleanliness code levels.

Friction Modifiers

[Source: Machinery Lubrication (<http://www.machinerylubrication.com>), What are Friction Modifiers?]

Friction modifiers and mild anti-wear agents are polar molecules added to lubricants for the purpose of minimizing light surface contacts. They are also called boundary lubrication additives.



Esters, natural and synthetic fatty acids as well as some solid materials such as graphite and molybdenum disulfide are used for these purposes. These molecules have a polar end (head) and an oil-soluble end (tail). Once placed into service, the polar end of the molecule finds a metal surface and attaches itself. If you could see the orientation of the molecules on the surface, it would appear something like the fibers of a carpet, with each molecule stacked vertically beside the others as shown in above figure.

As long as the frictional contact is light, these molecules provide a cushioning effect when one of the coated surfaces connects with another coated surface. If the contact is heavy, then the molecules are brushed off, eliminating any potential benefit of the additive.

When the designer anticipates more than light surface contact (from shock loading, for instance), then the designer would select a stronger type of friction modifier characterized as an antiwear additive. Zinc Dialkyl Dithiophosphate (ZDDP) is a common antiwear agent. This type of additive literally reacts with the metal surface when the reaction energy (temperature) is high enough. The reaction layer provides sacrificial surface protection.

As the loading and metallic contact increase, the strength of the additive and reaction process increases. This leads to the use of sulfur-phosphorus based EP (extreme pressure) chemicals. The EP additives form organo-metallic salts on the loaded surfaces that serve as sacrificial films to protect against aggressive surface damage.

There are two types of EP additives, those that are temperature-dependent, and those that are not. The most common temperature-dependent types include boron, chlorine, phosphorus and sulfur. They are activated by reacting with the metal surface when the temperatures are elevated due to the extreme pressure. The chemical reaction between the additive and metal surface is driven by the heat produced from friction.

In reacting with the metal surface, these additive types form new compounds such as iron chlorides, iron phosphides and iron sulfides (dependent upon which compound is used). The

metal salts produce a chemical (soap-like) film that acts as a barrier to reduce friction, wear and metal scoring, and eliminate the possibility of welding.

Gear Oil Ingredients

A gear oil's performance / property is achieved by the additive technology used in its formulation. Following table lists common ingredients used in various combinations in gear oil formulas. A formula does not have to contain every one of these additives, and there are multiple additives available to provide specific performance.

Ingredient	Function
Base Fluid	Mineral oil or synthetic fluid (PAO, PAG, ester) makes up 50-98% of the formula.
Viscosity Modifiers	Viscosity index modifiers/improvers are used to increase the viscosity of the base fluid. They are long polymer molecules (polybutene polymers) which uncoil at high temperatures to increase viscosity, while at low temperatures they form tight "balls" which no longer contribute much to the viscosity.
Rust & Oxidation Inhibitors (R&O)	Rust inhibitors coat metal surfaces to protect against rusting. Oxidation inhibitors defend the oil against degradation (formation of sludge, gums, varnish and acids) due to reactions with oxygen in the air by decreasing the rate of chemical reaction when the lubricant is exposed to elevated temperatures.
Copper Deactivators	Gear systems can contain some copper-containing metal elements that can be tarnished by gear oil ingredients. These additives protect this metal.
Antiwear Additives	Some gear applications operate in the mixed film lubrication regime, meaning slight metal-to-metal contact. As the base fluid is not sufficient to protect the surfaces from wear, additives are included to form a sacrificial surface that decreases the friction between two metal surfaces.
EP Additives (Extreme Pressure)	Heavily loaded gear applications can operate in the boundary wear regime, meaning the oil film is squeezed out completely. EP additives put down an aggressive coating that carries the load and protects the metal surfaces in lieu of the lost oil film.
Demulsifiers	Surface-modifiers are used to promote the separation of water.
Defoamants	Surface tension reducing polymeric compounds are used to inhibit the formation of foam on the surface of the oil that could result in housekeeping, oxidation and wear issues.
Pour Point Depressants	Polymeric ingredients are added to modify wax crystals that form in oils at low temperatures. They promote formation of smaller, rounded particles which are less likely to form large interlocking crystals. They keep the oil from gelling up, thereby expanding the oil's operability range on the low end.
Compounding Additives	A vegetable or animal fatty acid is put into formulations as a friction modifier. It is added to provide lubricity/oiliness.
Tackifiers	Sticky polymers with high molecular weight are added to increase a gear oil's adhesiveness (ability to cling to gears).
Dispersants	These surface active additives grab onto and disperse contaminants in bulk oil so that they do not collect in the gearbox, but instead can be carried to the oil filter for removal. They are used more often in automotive gear oil than in industrial gear oil.
Detergents	They chemically react with oxidation products (gums) so that they remain soluble in the oil and do not stick to metal surfaces.
Solid Lubricants	These are not common in lower viscosity enclosed gear applications, but are often used for lubrication of open gears. EP additives lay down a layer of solids that keep the metal surfaces from rubbing. Common examples include molybdenum disulfide and graphite.

Additive Depletion

Some additives, such as antiwear and extreme pressure additives and rust, oxidation, and corrosion inhibitors, are consumed as they are used. When all of a particular additive has been consumed, the lubricant is no longer capable of performing as originally intended. Usually this condition requires replacement of the lubricant, but in some cases, replenishment of the additive is possible. The lubricant manufacturer should be consulted before this is attempted.

Oil Compatibility

When it becomes necessary to replenish an oil lubrication system, the best practice is to always use oil that is identical to the oil already in the system. Similar oils made by different manufacturers for the same service may be incompatible because of different additives in the oils. The additives may react with one another, causing a depletion of these additives and leaving the oil unable to perform as it was intended.

Lubrication System Maintenance

It has been said that a gear wears out until it wears in after which it never wears out. After a gear has worn in during initial start-up period, it will have an indefinite life, provided it is properly lubricated with a clean lubricant of the proper viscosity and film strength, and operating within its rated capacity. Hence maintenance of a lubrication system is very important for a long gear life. The best way to maintain lubricant performance is by regular oil changes. In view of this, information about initial lubricant change period and subsequent lubricant change interval is given in this chapter.

Initial Lubricant Change Period

The flanks of the gear tooth in a new gearbox will wear during the first few hours (called running-in period) in service. This creates microscopic particles and contaminates the lubricating oil. In view of this, it is generally recommended to change the lubricating oil after a period of 500 operating hours or four weeks, whichever occurs first.

If desired, the lubricating oil may be filtered through a filter of 30 microns or less and used again if it is free of water and the original additive strength is maintained. However, it is recommended not to use a filter finer than 25 micro inches (0.635 microns) or through fuller's earth to avoid filtering out the additives from the lubricating oil.

Cleaning and Flushing

Start the gear drive and shutdown it when it reaches the normal operating temperature. After shutdown, lubricating oil should be drained when it is warm. It is recommended that the gear housing be thoroughly flushed after the lubricating oil has been drained. Flushing and filling the housing with new lubricating oil may be carried out as under.

Fill the housing to the indicated level with SAE 10 straight mineral flushing oil which must not contain additives. Start gear drive and bring up to operating speed (preferably without load) and then stop. Drain flushing oil and fill the housing with recommended lubricating oil to proper level.

For flushing, one of the leading Indian gear drive manufacturer suggests to use the light low viscosity oil such as "Lubrex 22" of Indian Oil Corporation Ltd. (IOC) or equivalent (Kinematic viscosity = 20 - 24 cSt at 40°C).

As per Indian Oil Corporation Ltd., Lubrex 22 is recommended for flushing of automotive and industrial equipments. The characteristics of Lubrex 22 make it possible to easily clean all inaccessible internal surfaces of various equipments.

The use of a solvent should be avoided unless the gear drive contains deposits of oxidized or contaminated lubricant which cannot be removed with a flushing oil. When persistent deposits necessitate the use of a solvent, a flushing oil should then be used to remove all traces of solvent from the system.

When solvents are used, one should consult the unit manufacturer to assure their compatibility with paint, seals, sealant and other components.

Subsequent Lubricant Change Interval

After running-in period, under normal operating conditions, it is recommended that the lubricating oil should be changed every 2500 operating hours or six months, whichever occurs first.

Above interval may be exceeded if the oil is sampled on a regular basis to be sure the oil meets the original specifications.

In case of gear drives used for general applications, many gear drive manufacturers recommend subsequent lubricant change interval to be after every 3000 operating hours, but at the latest after 18 months.

However, unusual environmental or load conditions may necessitate replacement of oil as frequently as one (1) or two (2) month intervals as determined by field inspection. Special attention must be given to the inspection of lubricants when following conditions exist:

- High operation temperatures resulting from heavy intermittent loads, causing the temperature of the gear housing to rise rapidly and then cool.
- Ambient temperature conditions which may cause sweating on the inside walls of the gear housing, contaminating the oil and forming sludge.
- Operating oil temperatures that remain constantly above 95°C (200°F).
- When the gear drive is exposed to ambient conditions of extreme dust, dirt, moisture and/or chemical fumes.

Precautions must be taken to prevent any foreign matter from entering the gear housing. Dust, dirt, moisture and chemical fumes form a sludge which is detrimental to proper and adequate lubrication.

Filtration

Gear drives with forced/pressurized lubricating oil systems should have a filter on the pressure side of the system to remove contamination particles. As a guideline, the filter should be no coarser than 50 µm (microns) absolute for gear drives with ball or roller bearings, and 25 µm absolute for gear drives with journal bearings. In addition, a coarse mesh/screen may be used on the suction side to protect the pump.

Caution

Lubricants should not be filtered through fuller's earth or other types of filters which could remove the additives of the original formulation.

Important Note

If premature wear occurs, lubricant selection should be reviewed

Condition Monitoring of Gear Drives

To prolong the operating life of gear drives, regular inspection and maintenance is essential. As opening a gear drive for full inspection and overhaul can cause undesirable lengthy downtime and money, it is recommended to carry out predictive maintenance to ensure the maximum interval between repairs to minimize downtime and cost of maintenance. Predictive maintenance is a condition-driven preventive maintenance program. To be effective, gear drives must be monitored on a regular basis and records maintained which will allow the user to observe trends and spot adverse changes in their condition to schedule maintenance activities on an as-needed basis. The three nondestructive techniques normally used for predictive maintenance of a gear drive are: lubricant analysis, vibration monitoring and thermography. Information about these techniques is given in this chapter.

Lubricant Analysis

Effective lubrication is extremely critical for gear drives as it will prevent gear and bearing failures. By keeping track of the condition of the lubricant/oil, damage to equipment caused by lubricant deterioration can be prevented. In view of this, the primary purpose of lubricant analysis is to determine the most cost effective interval for lubricant change based on the lubricant's actual condition.

Lubricant analysis, as the name implies, is an analysis technique that determines the condition of lubricant.

Generally, when one refers to lubricant analysis, it means that one will take a sample from the equipment in service at the prescribed interval and send it to a laboratory for analysis. However, there are number of quick spot checks that can be performed on the site by the user. These on the site checks also help user to maintain the equipment.

Lubricant Sampling

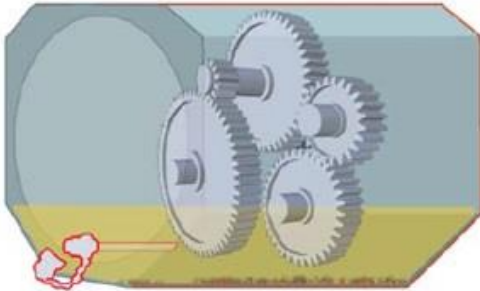
The effectiveness of a lubricant analysis program, whether it is on-site or laboratory, depends on proper sampling techniques.

The three sampling procedures commonly used include: sampling from a pressurized line before the system filter, using a drop tube in the dipstick tube or oil filling port and collecting a sample from a drain port.

For the lubricant analysis to be effective, one should ensure that the sample is representative of the bulk of the lubricant in operation. A sample skimmed off the top of an oil sump/reservoir after the oil has cooled and contaminants have settled out, may test cleaner than the oil actually is. Conversely, a sample taken from the bottom of an oil sump will likely contain sediment and contaminants that have built up over time, providing a much worse picture of the oil condition. Ideally, the oil sample should be drawn from the middle of the oil sump while the unit is operating. In most cases, this is not practical, and other procedures are required.

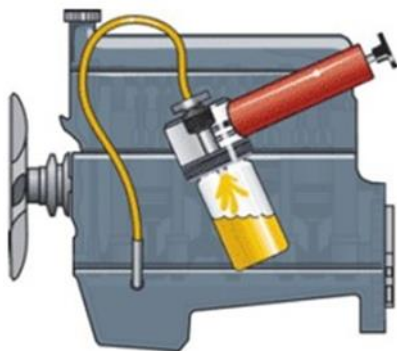
In a circulating system, it is best to obtain a sample from the pressurized line before the inline filter. The sampling port should be from an area where there is turbulence in the line, such as at or directly after an elbow or other fitting. This will ensure that the oil is well mixed.

The most common place to take a sample on a non-circulating system is from the drain. If this is the only place to obtain a sample, it is best to wait until the unit is shut down. However, in such cases the sample should be collected immediately after shutting down while the oil is still warm.



Sampling Tube Inserted into Gearbox

As shown in above figure, if the sampling is to be accomplished on a regular basis, it is recommended to permanently mount a sampling/pitot tube. The tubing can be installed on the bottom or the side of the sump and extend into the working oil area. A sample drawn from the working oil area of the sump will be more representative of the system oil. For more information on sampling/pitot tubes, please see www.lubrigard.com.



Using a Drop Tube

Above figure shows use of a drop tube to collect oil sample from dipstick tube or oil filling port. There are vacuum devices (for example, pumps) available that can assist in obtaining the sample.

Sampling from stagnant pool areas of the equipment may produce unreasonably high levels of contaminants. The initial sample of oil emerging from the drain or sampling valve would typically be a dead zone and likely to contain undue amount of particulate and wear debris. Hence in such cases, it is recommended to purge a large volume of oil prior to filling the sample bottle.

The intent of the monitoring program is to find out the condition of the oil in the equipment. Therefore, cleanliness is very important to prevent external contaminants to enter the oil during the sampling process. It is recommended that sampling equipment should be kept clean and, where possible, dedicated to a particular piece or series of equipment, and that clear, plastic/glass sample bottles be used to store and ship the oil to the laboratory. The inner surfaces of the drop tube and the sampling bottle should be flushed/rinsed with system fluid before collecting the sample. Another cleanliness issue to help the condition monitoring program is to ensure that the area around the sampling port is clean before opening the valve or port for sampling. This will prevent ingress of any unwanted debris or other contaminant to the system.

Consistency in sampling is another critical element to obtain an effective trend analysis from the data. Sampling should be done, whenever possible, from the same location in the

equipment and at the same point in the duty cycle. Varying either of these could lead to significant variation in the results obtained.

On Site Analysis

The on-site tests include: appearance test, odor test, sedimentation test and crackle test.

Appearance test is useful to identify potential problems with gross contamination or oxidation. In this test, a sample of the lubricant is filled in a clean, glass bottle (tall, narrow bottle is best) and compared to a new oil sample in the same type container. The oil should appear clear and bright. A hazy, cloudy, or milky appearance suggests the presence of water; if so, carry out the "crackle" test. A darkened color may indicate oxidation or contamination with very fine wear particles. Tilting the bottles (new and used oil samples) simultaneously will give an indication of changes in viscosity which could be related to oxidation or shear losses. Look for sediment in the bottom of the sample bottle; if present, carry out the sedimentation test.

For odor test, carefully sniff the oil sample and compare it to the sample of new oil. Oils that have oxidized noticeably will have a burnt odor or smell acid, sour or pungent.

For sediment test, place a sample of the oil in a clean, white plastic cup and allow it to stand covered in a clean and dust free area for two days. After two days, carefully pour off all but a few milliliters of the oil. If any particles are visible at the bottom of the cup, contaminants are present. If the particles respond to a magnet under the cup, then it indicated that it contains ferrous debris. If there is no response from the magnet and the solids feel gritty, they are likely to be sand, dirt or non-ferrous debris.

For crackle test, place a small drop of the oil in question onto a hot plate that has been warmed to 135°C. If the sample bubbles, possible water is present in excess of 0.05% (500 ppm). If the sample bubbles and crackles, the water level could be in excess of 0.1% (1000 ppm). The crackle test can also be conducted by immersing a hot soldering iron in a sample of the oil. This should be confirmed with the laboratory analysis.

Laboratory Analysis

There are many tests methods available to provide information about the condition of the lubricant. The basic analyses should include tests for the following.

- Viscosity
- Water content
- Acid number
- Particle count and size distribution
- Additive and wear elements

In all cases, comparison of the results should be made to a sample of the new oil that was actually used in the equipment to be sure that the starting material was within the specified limits stated by the supplier. The baseline values of the new oil also should be based on analysis of the actual sample (they should not be as per a brochure).

The results of analysis can be used to determine if the lubricating oil meets the lubrication requirements of the application. Based on the results of the analysis, lubricant can be changed or upgraded to meet the specific operating requirements.

Viscosity

Viscosity is a key property of the gear oil. The oil might lose its ability to lubricate properly if its viscosity changes significantly. Oxidation of oil leads to an increase in viscosity. There are many other possible causes for an increase or decrease in viscosity. For example, some oils have additives known as viscosity index (VI) improvers that might not be shear stable. With time-in-service, these oils lose viscosity because the VI improves shear down. Oxidation is accelerated by elevated temperature (overheating). In addition, contamination by water and wear debris also accelerates oxidation.

Kinematic viscosity can be checked by ISO 3104 / ASTM D445 test method. It is recommended to change the gear oil if its viscosity changes more than $\pm 15\%$.

Water Content

Water is a significant factor in lubricant degradation. When combined with iron or copper particles, water becomes even more powerful in attacking lubricant base stocks and additives. The adverse effects of water in oil include:

- Lubricant breakdown, through oxidation and additive precipitation.
- Changes in viscosity, affecting the ability of a lubricant to maintain the film thickness necessary to protect the lubricated surfaces.
- Corrosion.

Water content can be checked by ASTM D6304 test method. It is recommended to change the gear oil when its water content is greater than 0.05% (500 ppm).

Acid Number

Lubricant degradation occurs as a result of additive depletion, contamination and oxidation. The acid number (AN) also referred as total acid number (TAN) test is one of the methods to estimate the amount of additive depletion, acidic contamination and oxidation. It may be noted that AN test does not directly measure the rate of oxidation, it merely measures the by-product of oxidation. It is also beneficial to trend AN to determine the rate of depletion of certain additives.

AN is the measure of acid concentration in petroleum products and lubricants. It is determined by the amount of potassium hydroxide (KOH) required to neutralize the acid in one gram of the sample. The standard unit of measure is mg KOH/g. The AN measurement detects both weak organic acids and strong inorganic acids. However, AN does not represent the absolute acid concentration of the oil sample because the AN and base number (BN) tests are not affected by extremely weak acids and bases that have a dissociation constant in water of less than 10^{-9} . This is the reason why the term total acid number (TAN) is being replaced with acid number (AN).

Number of standard test methods for determining AN have been designed for specific purposes, with ASTM D664 and ASTM D974 being the two most commonly used one.

AN tests can be carried out by two titration methods: potentiometric and color-indicator. The potentiometric method (ASTM D664 / ISO 6619) uses a potentiometer to detect the acidic constituents and converts it to an electronic read out. The output is plotted and analyzed to determine the inflection of the test method. The color-indicator method (ASTM D974 / ISO 6618) uses p-naphtholbenzein solution (orange in acid and green-brown in base). Once the

acidic constituents have been neutralized by the KOH, the sample change from orange to green, indicating the end point.

In general, AN trend of the rust and oxidized (R&O) oils show parabolic curve. The AN remains almost constant during the additive depletion phase. Once the R&O additives have depleted, the base oil begins to oxidize. However, EP oils may show a switching trend if some of the additives (like sulphur) are acidic which can elevate the oil's initial AN level. In such oils, as additives deplete, initially the AN reduces. However later it shows increasing trend due to oxidation.

Acid numbers should not be allowed to increase more than +0.5 AN higher than that of new oil, and if +1 AN is spotted immediate action is required (i.e. if new oil has 0.5 AN, then 1.0 AN is alert and 1.5 AN is alarm value).

Acid can be neutralized or removed from oil in different ways. The obvious is to use the alkalinity of the oil to neutralize incoming acid. This is done in gas and diesel engine lube oils. These oils utilize high base numbers (BN or TBN), i.e. new oil are having high BN.

The units of BN and AN can be somewhat confusing. Although they are different tests, the results are both expressed in the same units: milligrams of potassium hydroxide per gram of the sample, represented as mg KOH/g. This is because the BN of an oil is the number of milligrams of KOH needed to neutralize the acid needed to neutralize the basic constituents in one gram of the sample.

Particle Count and Size Distribution

A good filtering system for the lubricant is very important. The design filtration level may vary, but filtration to a 25 micron or finer nominal particle size is a generally accepted value. Filtration finer than 25 microns is recommended when light turbine lubricants are used, particularly for higher operating temperatures.

Since particle contamination of oil is one of the main reasons for a machine to break down, monitoring the level of hard contaminants is vital. ISO 4406 establishes the relationship between particle counts and cleanliness in hydraulic fluids (common practice has extended the application of this standard to lubricants as well). The ISO 4406 method for coding the level of contamination of solid particles is a classification system that converts a given particle count into an ISO code.

The test methods used most frequently for counting particles are: automatic particle count according to ISO 11500 and manual particle count according to ISO 4407.

In automatic particle count method according to ISO 11500, the contamination level of a liquid sample is determined by automatic particle counting, using the light extinction principle. In this method, the particle concentration is reported at three sizes: ≥ 4 , ≥ 6 and ≥ 14 μm

In manual particle counting method according to ISO 4407, particles deposited on the surface of a membrane filter are counted using an optical microscope. It includes particle counting by two manual methods and image analysis, using either transmitted or incident lighting systems. In this method, the particle concentration is required to be reported at three sizes: ≥ 2 , ≥ 5 and ≥ 15 μm .

According to ISO 4407, counts at 5 and 15 μm of manual particle counting method are equivalent to 6 and 14 μm when using an automatic particle counter calibrated in accordance with ISO 11171.

The older ISO 4406 standard published in 1987 required particle concentration to be reported at two sizes: 5 & 15 µm. This has been replaced in the new standard published in 1999 with three sizes: 4, 6 & 14 µm.

ISO 4406: 1999 provides a 3-part code to represent the number of particles per milliliter (mL) of fluid greater than or equal to 4 µm, 6 µm, and 14 µm, respectively.

For example, referring to the following table, an ISO code of 17/15/12 would indicate that 640 to 1300 particles/mL greater than or equal to 4 µm, 160 to 320 particles/mL greater than or equal to 6 µm, and 20 to 40 particles/mL greater than or equal to 14 µm are present in the lubricant. In the following table, it can be seen that each step in the ISO code represents either double or half the particle count relative to an adjacent code.

It is important to note that the "/" character in the written form of the code is merely a separator, and does not signify a ratio of the scale numbers.

ISO 4406 - Allocation of Code Numbers to Particle Counts		
Code* Numbers	Number of Particles/mL	
	More than	Up to and including
> 28	2,500,000	-
28	1,300,000	2,500,000
27	640,000	1,300,000
26	320,000	640,000
25	160,000	320,000
24	80,000	160,000
23	40,000	80,000
22	20,000	40,000
21	10,000	20,000
20	5,000	10,000
19	2,500	5,000
18	1,300	2,500
17	640	1,300
16	320	640
15	160	320
14	80	160
13	40	80
12	20	40
11	10	20
10	5	10
9	2.5	5
8	1.3	2.5
7	0.64	1.3
6	0.32	0.64
5	0.16	0.32
4	0.08	0.16
3	0.04	0.08
2	0.02	0.04
1	0.01	0.02
0	0.00	0.01

* The ISO standard calls the codes "scale numbers." You may also find them referred to as "range numbers"

It is recommended to keep base cleanliness target of 17/15/12 for industrial gear drives.

After implementing cleanliness programs, many users discover that the dirtiest oil in their plant is the incoming “new” oil. Therefore, proper filtering of new oil during or before filling is a prudent and highly desirable practice in order to extend machine life.

Note:

Use of (c) after particle size, for example 4 µm(c) / 6 µm(c) / 14 µm(c), designates using of an automatic particle counter calibrated in accordance with ISO 11171.

Although the most common standard, ISO 4406:99, is not the only method by which the cleanliness of an oil sample can be reported. National Aerospace Standard (NAS) 1638 is other commonly used standard.

Following correlation table may be used to compare NAS 1638 with ISO 4406.

NAS 1638	4	5	6	7	8	9	10	11
ISO 4406: 1999	15/13/10	16/14/11	17/15/12	18/16/13	19/17/14	20/18/15	21/19/16	22/20/17

However, if you obtain the latest issue of NAS 1638 you will find the following statements: “Inactive for new designs after May 30, 2001, see AS4059” and “This standard should not be used with Automatic Particle Counting.” This is a result of recent changes to the ISO contamination standards for automatic particle counter (APC) calibration, which necessitated the review of NAS 1638 and resulted in its withdrawal for newly designed systems.

Changing of the American standard NAS 1638 to SAE AS4059 (issue E) is considered to be a significant advance because it presents data in terms of cumulative counts (>X µm) rather than interval mode (X-Y µm), introduces a cleaner (Class 000) and extends the size range to smaller sizes (>4 µm) for increased sensitivity. It may be noted that SAE AS4059 is technically identical to ISO 11218.

Elemental Analysis for Additives and Wear Metals

Some additives, such as antiwear and extreme pressure additives and rust, oxidation, and corrosion inhibitors, are consumed as they are used. When all of a particular additive has been consumed, the lubricant will be no longer capable of performing as originally intended. Monitoring the concentrations of wear metals in a lubricant can indicate abnormal wear of the machine components if baseline concentration data are available for comparison. Elemental analysis determines concentrations of wear metals, contaminants and oil additives in a sample for condition monitoring.

The two different instruments utilized for elemental analysis are the Inductively Coupled Plasma Spectrometer (ICP) and the Rotating Disk Emission (RDE) Spectrometer, or arc spark. Both instruments use a high energy source to excite atoms within a sample. The atoms do not want to be in this excited state and as energy is reduced, they give off the energy in the form of light energy. The light energy emitted is specific to each atom, therefore, the amount of light energy can be converted to a concentration of each element tested.

The RDE instrument does not need sample dilution/preparation, while the ICP instrument requires sample dilution. The RDE instrument can vaporize particles up to 10 µm, while the ICP instrument can vaporize particles up to 5 µm. While the ICP has the sample preparation and particle size limitations, the benefit of using this instrument is the accuracy. Some ICP units can get accuracy down to 0.1 ppm, while for the RDE it would be 1 - 2 ppm. It may be noted that additional testing would be needed for analysis of wear particle greater than 10 µm because the plasma cannot fully atomize the larger particles.

While ASTM D6595 method is used for the RDE instrument, ASTM D5185 method is used for the ICP instrument. Generally, ASTM D5185 method is used for elemental analysis.

A total of 22 elements (Al, Ba, B, Ca, Cr, Cu, Fe, Pb, Mg, Mn, Mo, Ni, P, K, Na, Si, Ag, S, Sn, Ti, V, Zn) can be determined by the ASTM D5185 test method - "Standard Test Method for Multielement Determination of Used and Unused Lubricating Oils and Base Oils by Inductively Coupled Plasma Atomic Emission Spectrometry (ICP-AES)".

Typical wear metal elements, contaminants and additive elements are as under.

- Wear metals are: Aluminium (Al), Boron (B), Copper (Cu), Chromium (Cr), Iron (Fe), Lead (Pb), Silicone (Si) and Tin (Sn)
- Contaminants are: Silicone (Si) from dirt; Sodium (Na), Potassium (K) and Boron (B) from coolant
- Additive elements are: Boron (B), Barium (Ba), Calcium (Ca), Magnesium (Mg), Manganese (Mn), Phosphorus (P), Sulphur (S), Silicone (Si) and Zinc (Zn)

It may be noted that some elements are used as both additives and wear metals. Monitoring their content is then crucial because a decreasing or an increasing value will induce different actions.

When additives are consumed as they are used, usually lubricant requires replacement. However, in some cases replenishment of the additives is possible. The lubricant manufacturer should be consulted before this is attempted.

Note:

It is recommended to consistently use the same lab and test method for a specific lubricant analysis to eliminate reproducibility error. As per ASTM, reproducibility is "the difference between two single independent results obtained by different operators working in different laboratories on identical test material."

Vibration Monitoring

Vibration can be defined as the cyclic or oscillating motion of a machine or machine component from its position of rest.

All machines with moving parts generates mechanical forces during normal operation. These forces cause them to vibrate when they are running. As the mechanical condition of the machine changes because of wear, damage, load variations, and so on, so do these forces. Since most machinery problems show themselves as excessive vibration, we use vibration signals as an indication of a machine's mechanical condition/health.

Vibration monitoring can be carried out for identifying and quantifying a problem, so that corrective action can be taken before the problem becomes too serious and cause unscheduled downtime. For vibration monitoring, vibration readings can be taken either on continuous basis or at scheduled intervals. As each mechanical problem or defect generates vibration in its own unique way, vibration analysis can be carried out to find out the "type" of vibration to identify its cause (for example unbalance, misalignment, bearing defects, gear defects, mechanical looseness, general looseness or wear, etc.) and take appropriate repair action.

In vibration, we look at two components of the vibration signal: its amplitude and its frequency.

Amplitude is the size of the vibration signal. The amplitude of the vibration signal determines the severity of the fault. The higher the amplitude, the higher the vibration, the bigger the problem. Amplitude depends on the type/class of the machine and is always relative to the vibration level of a “good” or “new” machine.

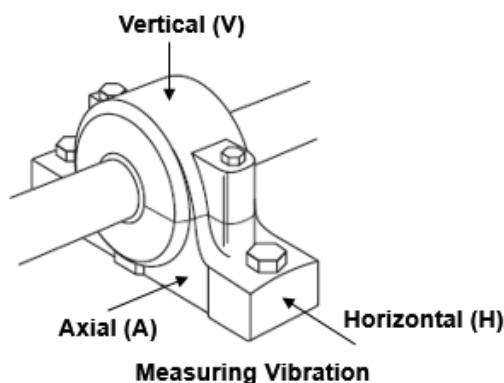
Frequency is the number of times an event occurs in a given time period (the event being one vibration cycle). The frequency at which the vibration occurs indicates the type of fault. That is, certain types of faults “typically” occur at certain frequencies. By establishing the frequency at which the vibration occurs, we can find out what could be causing it.

In this article, for condition monitoring, information is given to quantifying a problem only.

Vibration Measurement

Vibrations caused by defects in a rotating machinery travels to the ground through its supports. The bearings are the location where the vibrations ‘jump’ from the rotor to the stator to ultimately get grounded. Thus, it is at the bearings where the best signals for condition monitoring can be measured and hence these are generally the best locations for vibration measurements.

To measure vibration, a transducer or a vibration sensor/pickup is used. A transducer is a device that converts one type of energy, such as vibration, into a different type of energy, usually an electric current or voltage.



All vibration sensors measure motion along their major axis. Due to the structural asymmetry of the machine, the vibration amplitudes in the vertical, horizontal and axial directions (with respect to the shaft) may differ. Hence, as shown in above figure where possible, a vibration sensor should be mounted/held in the vertical (V), horizontal (H) and axial (A) directions to measure vibration in all the three directions. The intention is to detect/measure the strongest amplitude.

Typically, amplitude in horizontal direction is more than vertical direction due to the machine being more flexible in the horizontal plane.

Under ideal conditions, amplitude in axial direction should show very little amount as most forces are generated perpendicular to the shaft. However, misalignment and bent shaft problems do create vibration in the axial direction.

When measuring vibration with a hand-held sensor, it is important that you take consistent readings, paying close attention to the sensor’s position on the machinery, the sensor’s angle to the machinery, and the contact pressure with which the sensor is held on the machinery.

Position on Machinery

Measurements should be taken at exactly the same location to enable direct comparisons of data sets. Moving the probe only a small distance on a machine can produce drastically different vibration levels. To ensure measurements are taken at the same spot, it should be marked with permanent ink, or a shallow conical hole should be drilled for identification.

Sensor Angle

The sensor or the probe should always be oriented perpendicular to the machine surface ($90^\circ \pm 10^\circ$). Tilting the probe at an angle may show incorrect value.

Contact Pressure

Even and consistent pressure of the hand is required to get comparable readings with handheld meters (firm, but not so firm as to dampen the vibration signal).

What to Measure

Displacement is the change in distance or position of an object relative to a reference. The magnitude of the displacement is measured as amplitude.

There are two measurable derivatives of displacement: velocity and acceleration. Velocity is the change in displacement as a function of time, it is speed at which the distance is traveled, for example 0.5 mm/sec. Acceleration is the rate of change of velocity. For example, if it takes 1 second for the velocity to increase from 0 to 1 mm/sec, then the acceleration is 1 mm/sec^2 .

Thus, vibration has three measurable characteristics: displacement, velocity, and acceleration. Although these three characteristics are related mathematically, they are three different characteristics, not three names for the same quantity.

The displacement, velocity and acceleration characteristics of vibration are measured to determine the severity of the vibration and these are often referred to as the 'amplitude' of the vibration. Amplitude refers to the maximum value of a motion or vibration.

In terms of the operation of the machine, the vibration amplitude is the first indicator to indicate how good or bad the condition of the machine may be. Generally, greater vibration amplitudes correspond to higher levels of machinery defects.

It is generally accepted that between 10 Hz (600 cpm) and 1000 Hz (60 kcpm) velocity gives a good indication of the severity of vibration, and above 1000 Hz (60 kcpm), acceleration is the only good indicator. Displacement is used to check the severity of vibration below 10 Hz (600 cpm). Note: cpm and kcpm are abbreviations for cycles per minute and kilo cycles per minute respectively.

Since the majority of general rotating machinery (and their defects) operate in the 10-1000 Hz range, velocity is commonly used for vibration measurement.

It may be noted that displacement is typically measured with a sensor commonly known as a displacement probe or eddy probe. Displacement probes measure only the motion of the shaft or rotor relative to the casing of the machine. If the machine and rotor are moving together, displacement is measured as zero, while in fact the machine could be vibrating heavily.

Vibration Severity

One method to assess/check the vibration severity is to directly compare the acquired data to industrial standards.

The most widely used standard as an indicator of vibration severity is ISO 2372 (ISO 2372:1974 has been revised by ISO 10816-1:1995). BS 4675-1 is equivalent to ISO 2372:1974. ISO 2372 is based on the German standard VDI 2056 (withdrawn). The standard can be used to determine acceptable vibration levels for various classes of machinery.

ISO 2372 is applicable to a machine with running speed between 10 to 200 rev/s (600 to 12,000 rpm).

ISO 2372 defines concepts such as vibration evaluation zones and machine classes to classify the severity of the vibrations measured on the machine case.

The following typical evaluation zones are defined in the standard to permit a qualitative assessment of the vibration on a given machine and to provide guidelines on possible actions.

Zone A: The vibration of newly commissioned machines would normally fall within this zone.

Zone B: Machines with vibration within this zone are normally considered acceptable for unrestricted long-term operation.

Zone C: Machines with vibration within this zone are normally considered unsatisfactory for long-term continuous operation. Generally, the machine may be operated for a limited period in this condition until a suitable opportunity arises for remedial action.

Zone D: Vibration values within this zone are normally considered to be of sufficient severity to cause damage to the machine.

Another way to view above would be to consider the zones as having the following meaning:

- Zone A: Good
- Zone B: Acceptable (Allowable)
- Zone C: Unsatisfactory (Just Tolerable)
- Zone D: Unacceptable (Not Permissible)

Machines are classified in the standard according to size and mounting conditions as under.

Class I: Individual parts of engines and machines, integrally connected to the complete machine in its normal operating condition. (Production electrical motors of up to 15 kW are typical examples of machines in this category.) [Group K as per VDI 2056]

Class II: Medium-sized machines (typically electrical motors with 15 kW to 75 kW output) without special foundations, rigidly mounted engines or machines (up to 300 kW) on special foundations. [Group M, VDI 2056]

Class III: Large prime-movers and other large machines with rotating masses mounted on rigid and heavy foundations which are relatively stiff in the direction of vibration measurements. [Group G, VDI 2056]

Class IV: Large prime-movers and other large machines with rotating masses mounted on foundations which are relatively soft in the direction of vibration measurements (for example, turbogenerator sets and gas turbines with outputs greater than 10 MW). [Group T, VDI 2056]

By combining above definitions of evaluation zones and machine classes, ISO 2372 provides evaluation criteria as shown in the following table.

ISO 2372 - Evaluation Criteria for Vibration Severity					
Vibration Velocity		Machine Class			
in/s (peak)	mm/s (r.m.s.)	Class I	Class II	Class III	Class IV
0.015	0.28	A	A	A	A
0.025	0.45				
0.039	0.71				
0.062	1.12	B	B	A	A
0.099	1.8				
0.154	2.8	C	C	B	B
0.248	4.5				
0.392	7.1	D	D	C	C
0.617	11.2				
0.993	18				
1.54	28	D	D	D	D
2.48	45				

Evaluation Zones: A = Good, B = Acceptable, C = Unsatisfactory and D = Unacceptable

The values given in above table are in terms of overall vibration levels (in the frequency range 10 Hz to 1000 Hz).

For the overall vibration, ISO 2372 uses millimeters per second (mm/s) in terms of the root mean square (r.m.s.) values. The values in terms of inches per second (in/s), peak, are also listed in above table because inches per second, peak, is commonly used in the United States for vibration monitoring programs. [Note: r.m.s. value is equal to 0.707 of the peak or zero-to-peak value.]

It may be noted that the vibration limits for the smaller machines are lower than the larger machines (for example: Class II compared to Class III) because the rotors of the smaller machines are lighter and therefore produce lower forces of unbalance.

It may be observed that the evaluation zones boundaries defined in the ISO standard increase by a factor of 2.5 for each zone. That is, the transition from the upper level of zone A to the upper level of zone B is related by a factor of 2.5 and so on. For example, a Class III machine is considered “good” if the overall vibration remains below 1.8 mm/s, “acceptable” if the overall vibration remains below 4.5 mm/s (2.5 times 1.8) and “unsatisfactory”, if the overall vibration remains below 11.2 mm/s (2.5 times 4.5).

As per API 613: 1995, Special Purpose Gear Units for Petroleum, Chemical, and Gas Industry Services, during the shop test of the assembled gear drive, the overall casing vibration levels, unfiltered (peak), in the frequency range 10 Hz - 2.5 kHz, shall not exceed 4.07 mm/sec (0.15 in./sec).

ISO 2372 is good for a start (until you develop a machine history). However, as ISO 2372 defines “good” or “not good” conditions for various wide-ranged machinery classifications, it is unrealistic to judge a machine’s condition by comparing its current measurement value against a wide classification ISO 2372 or any other standards or general rules.

Another method, probably the most efficient and reliable method of evaluating vibration severity is to compare the most recent overall reading against previous readings for the same machine (when it was in good condition as reference), allowing you to see how the machine's vibration values are changing, "trending" over time.

Past values should include a base-line (reference) reading. The base-line value may be acquired after the machine is installed or an overhaul or when other indicators show that the machine is running well. Subsequent measurements are compared to the base-line reading to determine machinery changes.

Comparing a machine to itself over time (comparing apple to apple) is the much preferred method for detection of machinery problems as each machine is unique in its operation. For example, some installation/component has a certain amount of vibration. The current reading, as per ISO 2372 by itself might lead an analyst to believe that a problem exists, whereas the trend plot and base-line reading would clearly show that a certain amount of vibration is normal for this machine. This is sometimes the case with cooling tower fan units since the gearbox might not be rigidly supported. The lack of rigid support will allow a higher case vibration level than "normal" for that size of machine. This will happen when the support structure is adequately sized to support the weight of the gearbox, but not adequate to restrain the gearbox from "swaying" due to blade imbalance. In other words, the blade imbalance causes the gearbox to "rock" back and forth, albeit on a small enough scale that intervention is not required.

In such cases, where the ISO 2372 guidelines are not adequate, the user is left to try to determine what levels to set for severity limits. This is where the knowledge of the zone boundary relationships observed in ISO 2372 can help. Using the zone "width" of a 2.5 multiplier as a guide, the user can establish limits (with respect to the base-line reading) that will help determine when vibration has changed enough to merit attention or action.

Thermography

A rise in temperature or localized hot spots can indicate that the gearbox is not operating as efficiently as it once was due to a problem with either the gears or bearings. Thermography, RTDs or thermocouples can be used for the temperature measurement. For monitoring temperature of a gearbox, a baseline reading/value should be established and subsequent readings should be compared against the previous values.

Thermography uses instrumentation designed to monitor the emission of infrared energy (i.e. surface temperature) to determine the operating condition.

Infrared imaging provides the means to scan the infrared emissions of complete machines, process, or equipment in a very short time. Most of the imaging systems function much like a video camera. The user can view the thermal emission profile of a wide area by simply looking through the instrument's optics.

However, the user of thermographic techniques must consider the atmosphere between the object and the measurement instrument. Water vapor and other gases absorb infrared radiation. Airborne dust, some lighting, and other variables in the surrounding atmosphere can distort measured infrared radiation.

Most infrared monitoring systems or instruments provide filters that can be used to avoid the negative effects of atmospheric attenuation of infrared data; however, the instrument user must recognize the specific factors that affect the accuracy of the infrared data and apply the correct filters or other signal conditioning required to negate that specific attenuating factor/s.

Inspection and Maintenance of Gearboxes

To prolong the operating life of a gearbox (gear drive), regular inspections and maintenance are essential. In view of this, general information about visual inspection and overhauling (disassembly and assembly) of a gearbox is given in this chapter. In addition to these instructions, it is very important to follow the manufacturer's instructions, which are specific to the actual gearbox being maintained.

Tips for Visual Inspection

Often gearboxes operate in a dusty environment. Whilst this is usually unavoidable, it is important to minimize the effect of the workplace environment. Dust could result in an increased operating temperature of the gearbox. Therefore, gearboxes should be regularly cleaned.

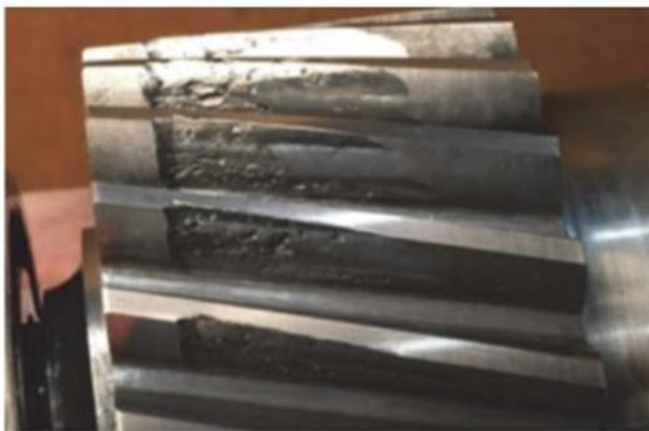
Look for signs of overheating - discolored or burnt exterior paint or dark oil in the sight glass. If found, take corrective action.

Check for oil leaks at the input and output shaft of the gearbox. Leaks indicate that the seals have failed and may allow ingress of dust, debris and water from the environment together with a loss of lubrication. Seals should be replaced without delay to prevent contamination of the lubricant.

Visual inspection of gears for wear/damage and tooth contact pattern through an inspection port should be carried out periodically. Detecting a problem in its earliest stages can save time and money in the long run. If the gears or bearings are damaged but still functional, you may decide to continue operation and monitor damage progression. In this case, the gearbox should be continuously monitored. You should also make certain that there is no risk to human life.

Many times gearboxes operate in dirty environments. Therefore, areas around inspection ports should be cleaned before they are opened for inspection. Inspectors should take care not to drop anything into the gearbox. Inspection ports should never be left open during breaks and should be closed after the inspection is complete.

Purpose of Checking Tooth Contact



Macropitting on Pinion due to Severe Misalignment

Gear teeth must have an even load across the entire face width to minimize stress on the teeth. The contact between gear teeth is a line contact. Therefore, the alignment between the

rotating elements (pinion and gear) is critical. Tooth contact checking is carried out because they can disclose gear mesh misalignment.

The pinion is often wider than the gear, and if there is misalignment, a wear step is likely to be at either end of the pinion. Severe misalignment usually causes macropitting as shown in above figure. Such damage could be prevented by checking tooth contact pattern.

Tooth contact should be checked on all new installations, after any disassembly of the gearbox, and after any major housing-to-foundation change. It can also be checked as a part of routine annual maintenance or when a problem related to alignment is suspected. For detail information on checking and correction of tooth contact pattern, please see the chapter on tooth contact checking.

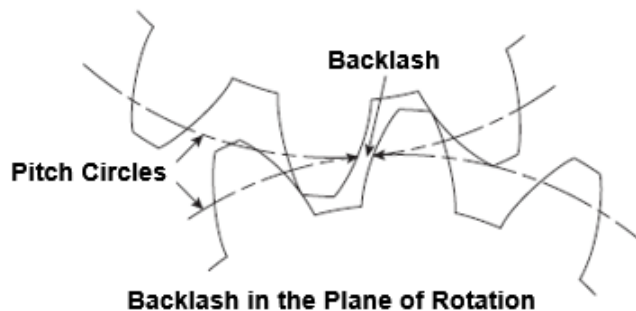
Backlash and Shaft Endplay

For long life of a gearbox, it is very important to set its gears and bearings properly during its assembly. Proper setting of gears usually involves checking backlash and tooth contact pattern. While proper setting of bearings calls for proper internal clearances and usually involves checking shaft endplay. In view of this, detail information about backlash and shaft endplay is given in this section.

It is recommended to check for any increase in backlash and for any increase in the end play or lift at the input and output shafts during disassembly of a gearbox. An increase in backlash could be an indication of wear in gear teeth which is not always visible to the naked eye. An increase in shaft end play or lift would indicate wear within the rolling elements of the bearings or even wear in the bearing housings.

Backlash

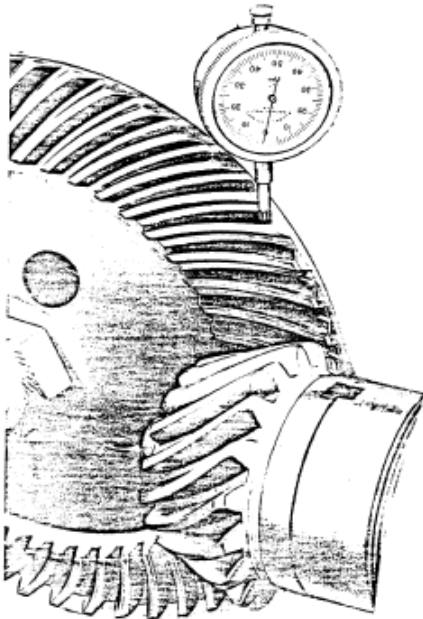
An individual gear does not have backlash. Backlash is only present when one gear mates with another. Backlash is necessary to achieve correct operation of the gears and varies with the size of the tooth and the operating conditions. Excessive or insufficient backlash can result in noise, excessive wear, and damage.



Backlash in gears is the clearance or play between mating tooth surfaces. Backlash in the plane of rotation (in terms of tooth dimensions) is the amount by which the width of tooth space exceeds the thickness of the engaging tooth on the pitch circles. It is measured at the tightest point of mesh.

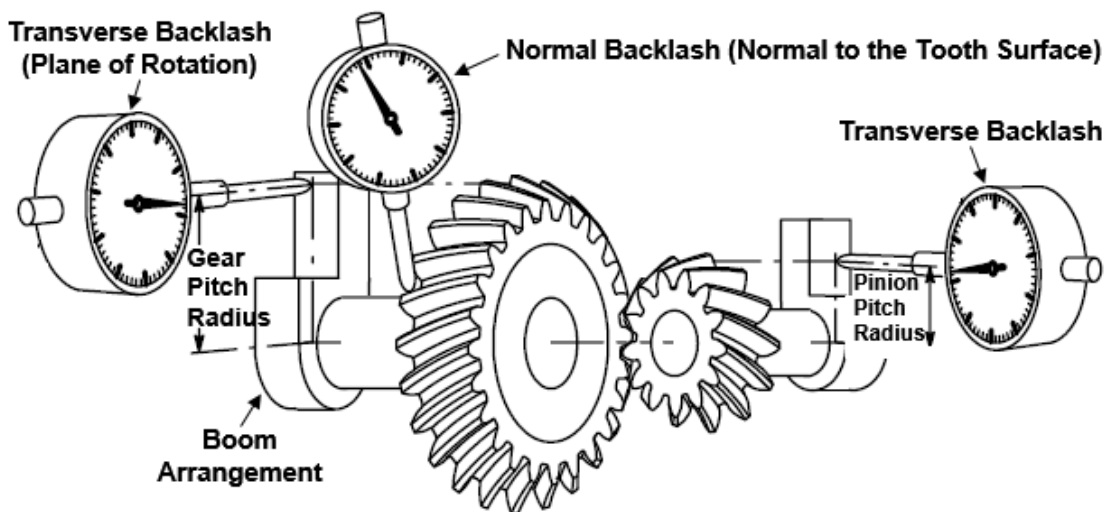
Normal backlash is the backlash at the tightest point of mesh on the pitch circle in a direction normal to the tooth surfaces when the gears are mounted in their assembled positions.

Normal backlash is typically measured with feeler gauge or dial indicator normal to the gear tooth for a given mesh.



Measurement of Normal Backlash

Above figure shows measurement of a normal backlash in a bevel gear. As shown in the figure, normal backlash of a pair of bevel gears is measured in a direction normal to the surface of the tooth. It can be checked by locking the pinion against rotation, placing a dial indicator against the gear tooth perpendicular to the tooth surface at the extreme heel of the tooth, and rotating the gear back and forth (in both directions).



Bevel Gear Backlash, Normal and Transverse

As shown in above figure, transverse backlash, backlash in the plane of rotation can be measured by attaching a boom arrangement or can be established from the normal backlash by dividing it by the cosine of the spiral angle of the bevel gear and the cosine of the pressure angle of the gear teeth. Transverse backlash of a spiral bevel gear is approximately 30% higher than the normal backlash.

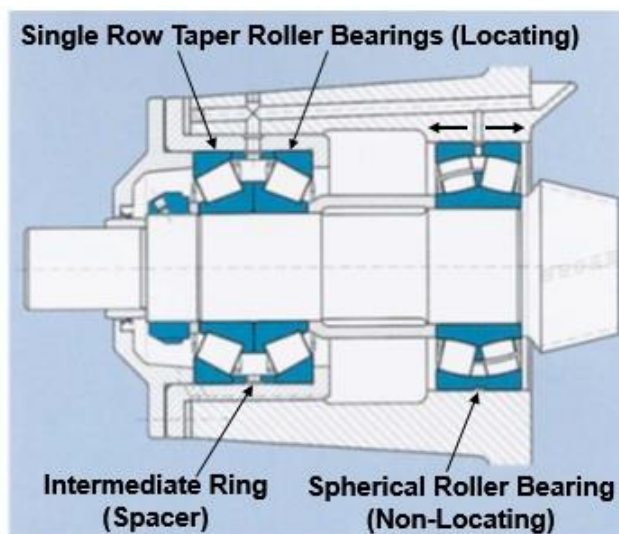
In bevel gears, unless specified otherwise, the normal backlash is etched or stamped on one or both of the members.

Shaft Endplay

A major consideration for any bearing application is to provide the means to compensate for any linear expansion of the shaft due to temperature rise. This is required to prevent additional stresses from being imposed on the bearings.

The bearing arrangement of a rotating machine component (e.g. a shaft) generally requires two bearings to support and locate the component radially and axially relative to the stationary part of the machine (such as the housing). This is typically accomplished using a locating bearing and a non-locating bearing.

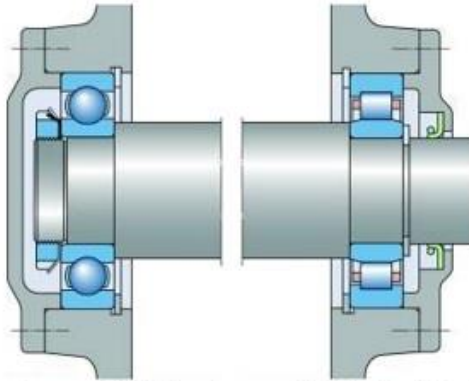
The locating bearing at one end of the shaft provides radial support, and at the same time locates the shaft axially in both directions. It must therefore be laterally fixed, both on the shaft and in the housing. Suitable bearings as a locating bearing are radial bearings, which can accommodate combined loads (e.g. deep groove ball bearings, spherical roller bearings or double row or paired single row angular contact ball bearings or taper roller bearings). Combinations of a radial bearing to take purely radial load (e.g. a cylindrical roller bearing without flanges on one ring) with a bearing to take axial load (e.g. a deep groove ball bearing), can also be used as a locating bearing. The second bearing, non-located (axially free) bearing then provides axial location in both directions. However, sufficient radial clearance must be provided in the non-locating (free) bearing housing for the outer ring to be able to move freely. As displacement of the non-locating bearing is always assured, there is no risk of excessive load on the bearings when the shaft expands.



Use of Single Row Taper Roller Bearings and Spherical Roller Bearing for Locating and Non-Locating Bearing Arrangement

Above figure shows bearing arrangement for a bevel pinion shaft with two matched single row taper roller bearings arranged face-to-face (as a locating bearing) and one spherical roller bearing (as non-locating bearing).

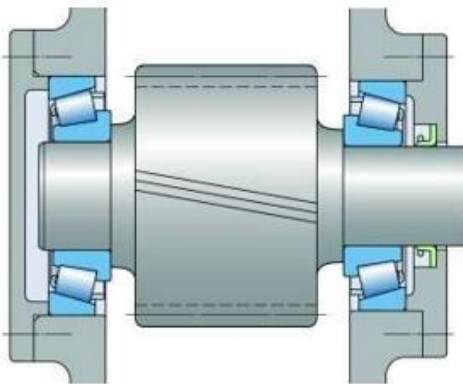
As shown in above figure, when paired single row taper roller bearings in a face-to-face arrangement (designation suffix DF for SKF make bearings) is used, the bearing pair is supplied with an appropriate intermediate ring so that the bearing assembly will have desired internal clearance. As these bearings and intermediate ring are supplied as an assembly, parts must not be interchanged, even between bearings of the same size. Also make sure that the bearings are installed with the proper cone and cup if they are marked (for example, cone marked "A" with cup marked "A").



Use of Deep Groove Ball Bearing and Cylindrical Roller Bearing for Locating and Non-Locating Bearing Arrangement

Above figure shows use of a deep groove ball bearing as a locating bearing and use of a cylindrical roller bearing without flanges on the inner ring/race as a non-locating bearing. As flanges are not there on the inner race of the cylindrical roller bearing, the shaft is free to expand axially in both the directions.

“Cross Location” is used to describe bearing arrangements where each of the two bearings locates the shaft axially in one direction only, the directions being opposed. This arrangement is mainly used for short shafts. All types of radial bearings that can accommodate axial loads in at least one direction are suitable for this type of arrangement. Where single row angular contact ball bearings or taper roller bearings are used for cross-located arrangements, preload may be necessary in certain cases.



Use of Taper Roller Bearings for Cross Location of Bearings

Above figure shows use of single row taper roller bearings for cross location arrangement. The internal clearance of single row angular contact ball bearings and single row tapered roller bearings is only established when the bearing is adjusted against a second bearing. Since there is a definite relationship between the radial and axial internal clearance of angular contact ball bearings and tapered roller bearings, it is sufficient to specify one value, generally the axial internal clearance.

Hence, above bearing arrangement must be adjusted for axial clearance or preload during assembly. Information for axial clearance or preload value must be obtained from the machine manual or bearing manufacturer.

As the bearings have to be adjusted on mounting, i.e. in an unloaded condition at ambient temperature, the changes produced when the bearings are in operation must be considered

when determining the adjustment values. The main influencing factors are temperature and deformations (e.g. gear drive housing).

If single row angular contact bearings (angular contact ball bearings, taper roller bearings) are used in a gearbox, light preload is recommended. Light preload provides accurate shaft guidance, increased stiffness, extended service life and quiet running as compared with clearance.

When rolling element bearings are used in a gearbox, the manufacturer, based on his experience and the recommendations of his bearing supplier determines the type of bearings and their settings (e.g. fit between inner race and shaft, fit between outer race and housing, end play). These settings are very important for long operating life of a gearbox. The end play allows for thermal expansion of the gear element assemblies in the axial direction with respect to the housing due to the temperature changes from ambient to operating temperature to prevent excessive preload on the bearings. Hence, end play of the shafts should be checked to ensure compliance with the specification. The end play is typically measured with the shaft moved all the way in one direction and then moved fully in the other direction. Total movement is the end play.

Overhauling (Disassembly and Assembly) of Parallel Shaft Gearboxes

The purpose of this section is to describe what should be carried out before disassembly, during disassembly and during assembly for overhauling gearboxes. Though the tips are given for overhauling a single stage parallel shaft horizontally split housing gearbox as an example, they are general in nature and are applicable to all type of gearboxes.

Before Disassembly

Note down important operating parameters if any. For example, in case of a pressure-fed systems with an oil cooler, temperatures of oil inlet and outlet as well as the cooler water inlet and outlet.

Survey operating temperature of the gearbox by touching it with the palm of your hand, using temperature-sensitive crayons, a digital/infrared thermometer or infrared imaging camera.

Look for evidence of movement, including cracked paint or fretting corrosion at structural interfaces.

Disassembly and Inspection

Do not work on the gearbox or couplings until absolutely certain that the prime mover cannot be remotely or automatically started.

Check alignment of the gearbox with the driver and driven machines.

Check the tightness of structural fasteners that carry significant loads.

To check for possible twist in the gear housing, install a dial indicator at each corner of the gearbox and then measure movement of the mounting feet as bolts are loosened. If there's no twist, each indicator will record the same vertical movement.

Padded slings/chains or nylon slings should be used when handling machined parts. Clean up the area around the gearbox before disassembly to keep clean parts. Sufficient space should be available to lay parts in proper order for assembly. Remember that parts are usually

assembled in reverse order of disassembly. Provide wooden planks or skids for storing machined parts to prevent damage to the machined surfaces.

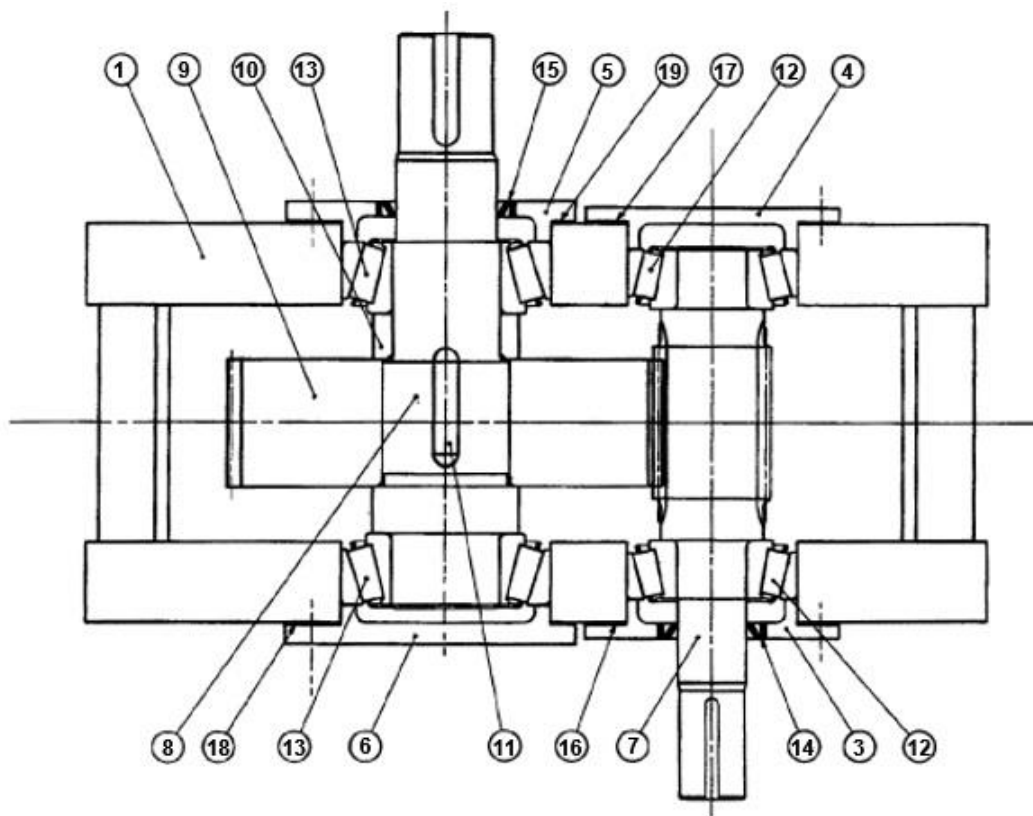
Drain oil from the housing.

Disconnect any piping or wiring that joins the housing sections. Remove any bearing temperature sensors, probes or other auxiliary instruments that could be damaged by removal of the housing cover.

Move gearbox to a clean prepared area for disassembly when possible.

Throughout the disassembly sequence, carefully observe what might have occurred inside the unit. When parts are removed from the gearbox, they should be inspected for wear or damage.

Note any parts or holes that are numbered or match marked. They must be reassembled as matched for correct assembly.



Gearbox with Horizontally Split Housing

Item No.	Description	Item No.	Description
1	Lower Housing	11	L.S. Gear Key
3	H.S. Thru Cap	12	H.S. Bearing
4	H.S. Blind Cap	13	L.S. Bearing
5	L.S. Thru Cap	14	H.S. Oil Seal
6	L.S. Blind Cap	15	L.S. Oil Seal
7	H.S. Pinion and Shaft	16	H.S. Cap Gasket
8	L.S. Shaft	17	H.S. Cap Shim Set
9	L.S. Gear	18	L.S. Cap Gasket
10	L.S. Bearing Spacer	19	L.S. Cap Shim Set

Legend: H.S. = High Speed, L.S. = Low Speed

Above figure shows parts of a typical single stage horizontally split housing gearbox (after removing the upper housing).

To disassemble gearboxes with horizontally split housings, proceed as follows.

Measure backlash and shaft endplay.

Mark cap position as a guide for reassembly. Remove cap bolts and caps. If old oil seals are to be salvaged, cover keyways with cellophane tape.

Be sure to clean, save and record placement of all shims and gaskets during disassembly to help in the reassemble process. These items may be wired together and tagged.

Remove stud nuts and dowel pins at the housing split. Leave the studs in place to serve as guides for housing removal.

It may be necessary to break the seal at the housing split in order to lift the upper housing. Some housings have jackscrew taps on the upper housing flange for this purpose. Striking the upper housing corners with a soft-faced mallet will help break the seal. Prying with a large screwdriver can be used but should be done with extreme care to avoid marring the sealing surfaces.

The upper housing should be lifted off vertically. When attaching the hoist, extreme care should be used to distribute the load to achieve a vertical lift.

Place the upper housing on wooden planks so that the machined split surface will not be damaged.

After the upper housing has been removed, the shaft assemblies starting with pinion shaft can be lifted from the housing. The shaft may be rotated slowly for the gear teeth to come out of engagement.

Further disassembly of the shaft assemblies will require the use of a bearing puller and/or hydraulic ram to remove the parts. Before removing parts, carefully note their location and check for match marks.

Care must be taken when dismounting the bearings if they are going to be re-used.

Using a cutting torch to remove bearings is not recommended. Grease may ignite or explode and may result in personnel injury and/or fire.

Clean all interior surfaces of the housing, the housing cover, and all components that will be reused. Split (parting line) surfaces must be clean and smooth; use a paint and gasket remover fluid and/or carefully scrape the surfaces if necessary. Corroded spots can be cleaned by using a fine emery cloth, rubbing shafts in a rotary or circumferential direction. Do not rub shafts in a lengthwise (axial) or spiral direction as that might cause seal leaks.

For critical applications, examine the gears with a nondestructive testing method such as magnetic particle inspection to ensure that there is no crack that prevent safe, continued operation.

Parts such as gaskets and oil seals should be replaced. It is also recommended to replace the bearings.

Check all parts for dimensional and form accuracy prior to assembly. Non-observance of the tolerances for shaft and housing seat diameters, out-of-roundness of these parts, out-of-square of abutment shoulders etc. impair bearing performance and may lead to premature failure.

If after measurement, the shaft does not meet the recommended fit tolerances, the shaft must be repaired or replaced.

Knurling and/or peening the shaft is generally ineffective. The burrs raised in the process have very little contact area. Under load, the asperities flatten and creep recurs.

The shaft surfaces where the oil seals were rubbing should be carefully inspected for damage. If a shaft is even lightly grooved where the oil seal lip rubs, if possible, during assembly, the oil seal may be installed in a new axial location to allow contact in a non-grooved area. If it is not possible to install the oil seal at new axial location, the shaft must be repaired or a "Speedi Sleeve" may be installed on the shaft.

Assembly

First make shaft assemblies.

If gear has a chamfer (as a guide for assembly) on one end of bore only, this end must go against shaft shoulder.

Cleanliness is critical to getting the proper service life from bearings. The shaft should be completely cleaned to receive the bearing before the protective wrapping is removed from the bearing.



No Hammer Blows

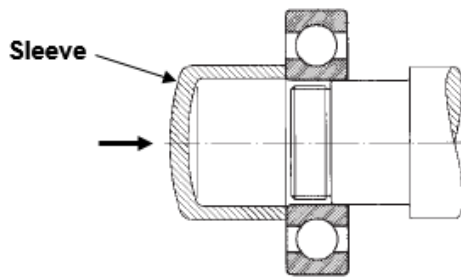
As the hardened bearing rings are sensitive to blows, hammer blows must never be applied directly to the rings. Hammering directly on a bearing will often cause denting damage (brinelling) to the raceways and its service life will be reduced.

For mounting of a non-separable bearing, the mounting force must always be applied to the ring which will have the tight fit. If the force is applied to the ring via the rolling elements, it will damage the raceways and the rolling elements.

Bearings with a maximum bore of approximately 80 mm can be mounted cold. For cold mounting, the use of a mechanical or hydraulic press is recommended.

However, if tight fits are specified for the inner rings on cylindrical shaft seats, even the small bearings should be heated for easy of mounting. Sufficient expansion is obtained when heated

between 80 and 100°C. Accurate temperature control is essential in heating the bearings. If the temperature exceeds 120°C there is the risk of alteration of bearing grain structure resulting in a drop of hardness and dimensional instability.



Mounting Bearing with a Sleeve

If press is not available, the bearing can be driven on the shaft by gentle taps with a hammer or mallet. However, a mounting sleeve with a flat face must be used to distribute the mounting force evenly over the entire ring circumference to avoid damage to the bearing.

Loosely assemble all caps (bearing retaining caps) to the lower half of the housing.

Starting with the output shaft, lower the shaft assemblies into the lower half of the housing. The bearing cups (outer races of single row taper roller bearings) should be on the cones (taper rollers) at this time. Carefully place the shaft assemblies in their bores, making sure that the bearings are not cocked and that they seat squarely in the bores. Be sure to allow the shafts to rotate and the teeth to mesh as each shaft is placed.

Set the axial clearance required on single row tapered roller bearings by installing gasket and properly tightening bolts on caps on one side, then tighten the bolts on the caps on the other side without gasket and shim pack until they are snug (zero axial end play). Rotate the shaft assembly slightly for the bearing rollers to seat properly. Measure the gap between the housing face and the cap with a feeler gauge. Check the gap at several points to make sure that the cap is square with the housing. Add the required amount of axial clearance to the dimension obtained and insert a gasket and shim pack of that thickness between the cap and the housing face and tighten cap bolts fully. Measure backlash and confirm required axial end play. End play is checked by positioning a dial indicator at the end of the shaft having its pointer parallel to the shaft axis. A slight axial force with some oscillations is then applied to the shaft to seat the bearings in both axial positions to determine the assembled end play. Loosen the cap bolts on both sides.

If factory made gaskets are not available, a good quality manila paper may be used to make new gaskets. Gaskets/shims are never used between the bearing cup (race) and caps.

It may be noted that the thickness of gasket and shim pack required will be equal to original thickness if new bearings are used (assuming no damage to the shaft and the bearing housings). If worn (old) bearings are used, the thickness of gasket and shim pack required may be less than the original thickness (to compensate for the bearings wear which had resulted in increase in the axial clearance).

One leading gearbox manufacturer recommends total axial end play to be between 0.050 and 0.125 mm (based on size of the housing).

Check lower housing split surface for cleanliness. Carefully apply a thin coat of non-hardening type liquid sealing compound to lower housing split surface (or place a new gasket of equal thickness if the housing split is designed for a gasket to maintain bore sizes).

Check upper housing split surface for cleanliness. Carefully lower upper housing into position, making sure that all internal passages are correctly lined up and that dowel pins and bolt holes are lined up.

Install dowel pins at housing split. Insert bolts in the caps above the split line and tighten them evenly. Be sure the caps rest flat against the housing. Now torque tighten all cap bolts and housing studs as per manufacturer's recommendations.



The next step is to install the oil seals. Coat the lips of the oil seals with a small amount of grease and then assemble them in the thru caps. For oil seal installation, it is recommended to use oil seal sleeve and oil seal driver as shown in above figure. If oil seal sleeve is not available, apply cellophane tape over the full length of the keyway to avoid damage to the oil seal.

Now follow recommended procedures for handling, installation and start-up in the operation and maintenance manual except running in if gears are not changed.

The remaining major activities are:

- Installing the gearbox back on the base plate if it was moved to a clean area.
- Coupling the gearbox to the driver and driven machines and align it with them.
- Confirm for proper assembly of the gearbox and shaft alignment by checking tooth contact.
- Reconnect any instrumentation and lubrication lines as necessary and fill lubricant.
- Take trial run of the gearbox and record various operating parameters.

It is very important to set gears and bearings properly during its assembly. However, in the tips given above, information is given only about how to set bearings properly (to recommended shaft endplay). Except recommending to check backlash and tooth contact pattern, no information is given about correcting them if the desired results are not achieved. The reason is: parallel axis gearing (spur and helical) usually offer very little opportunity for change.

Most of the time, the two supporting shafts are fixed in their relationship to each other by the location of the gearbox support bearings. All you can do is move the shafts axially in relation to each other. If the gears have crowning, the axial movement can be used to center the contact pattern. The backlash is controlled by tooth thickness and center distance. Since neither can be changed, you cannot adjust backlash.

However, in case of non-parallel axis gear drives, it is possible to carry out changes. While the shafts are still fixed in their relationship to each other, axial movement can greatly affect the contact pattern and the backlash. This is true for worm and bevel gears. To get the proper contact and backlash, one should mount the gears at the position the manufacturer made them at. In view of this, information about mounting the gears at the position the manufacturer made them at is given in the following sections.

Assembly of Worm Gearboxes

Note: During disassembly, a torch may be used to heat the worm gear starting at the outside diameter and ending at the gear hub, to expand the gear slightly while pressing the shaft in the hydraulic press.

The performance capability that is designed and manufactured into a worm gear set can only be achieved by correct assembly. For this, during assembly, the worm and worm gear must be located to provide tooth contact pattern and backlash to the design specification.

Generally, in the assembly of a worm and worm gear, the only possible adjustments are the axial positioning of the worm and worm gear. For adjusting their axial position, means of shimming is provided during its design.

Assemble worm and worm gear in the housing. Worm gearboxes are generally assembled with single row taper roller bearings. Hence their axial clearance (shaft end play) is very important. Set the end play of both worm and worm gear shafts to the design/recommended value by either adding or removing shims from the caps (shaft/bearing end covers/retaining plates). When making an adjustment, rotate the shaft to insure that the bearings are seated properly. Now to check correctness of the assembly, measure backlash and tooth contact pattern.



To check tooth contact pattern, apply a thin coat of Prussian Blue to the worm threads. Rotate the worm several revolutions in both directions to transfer blue to the worm gear. Check the contact pattern. It should be as per manufacturer's recommendation (generally it is recommended to have a contact on the leaving side). To change the contact pattern, as shown in above figure, adjust the position of the worm gear axially by removing shims of desired thickness from one cap and installing them (or one having exactly the same thickness) on the opposite cap. This will move the worm gear in the direction the shims were removed and by exactly the amount of the shims thickness.

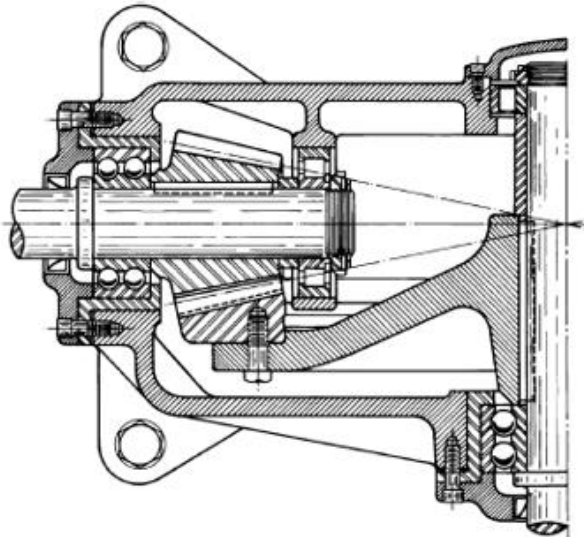
Backlash is generally measured by holding the worm stationary and rotating the worm gear in both directions. The movement is registered by a dial indicator having its pointer in the plane of rotation at or near the worm gear pitch diameter and parallel to a tangent to the pitch circle of the worm gear. If the housing does not allow for direct measurement on the worm gear, a

measurement fixture with an indicator location equal to the worm gear pitch diameter can be installed on the worm gear shaft as an alternative measuring device. The backlash will slightly increase from the initial value during assembly to the final backlash value after the worm gear profile has developed to match the worm thread during run-in.

Assembly of Bevel Gearboxes

In this section, information related with bevel gearboxes and tips for their assembly is given.

Types of Mountings



Typical Straddle Mounting for Pinion and Gear

Generally, as shown in above figure, straddle mounting for both gear and pinion are used for industrial and other heavily loaded applications. However, when it is not feasible to use straddle mounting arrangement for both the member, for example due to gearbox space limitation, the element having the higher radial load should be straddle mounted and the other overhung mounted.

In an assembly where the gear can move axially due to the internal clearance of the bearing (shaft end play), the gear should be located in its normal running position (to its foremost position toward the crossing point) when tooth contact pattern and backlash are checked.

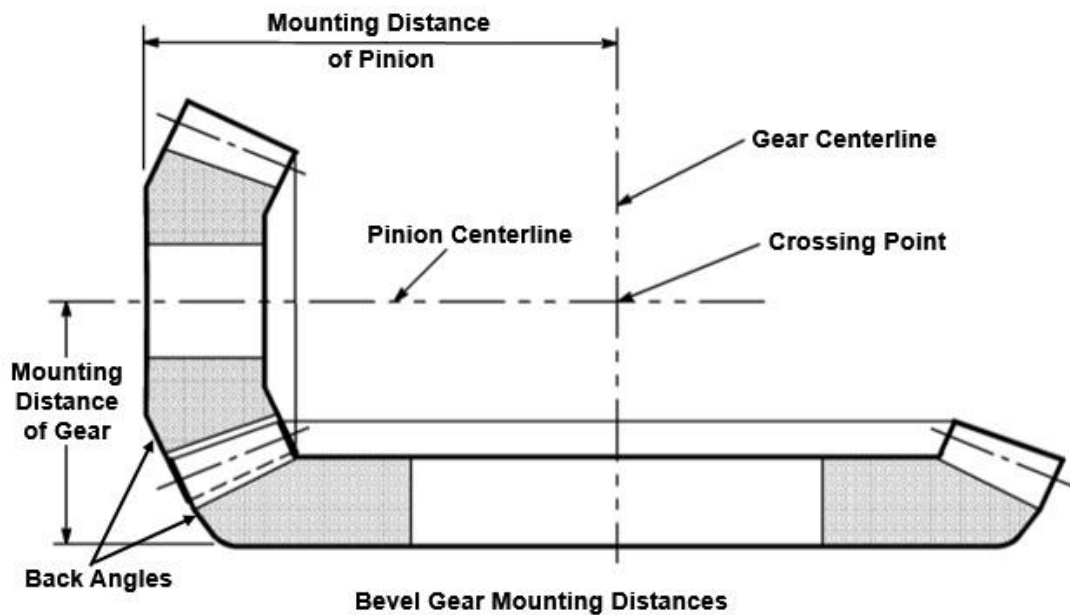
Assembly Parameters and Procedure

All types of bevel gears have an optimum position for best performance. The most important parameters for assembly are mounting distance, matched teeth and backlash.

It is possible to manufacture all gears to the nominal mounting distance specified on the drawing. But, the additional cost to do so is high. Hence, bevel gears of AGMA quality 8 or better are normally manufactured and tested in sets. The manufacture marks the preferred values for each of the three parameters on individual gears after testing them as a set in a bevel gear testing machine.

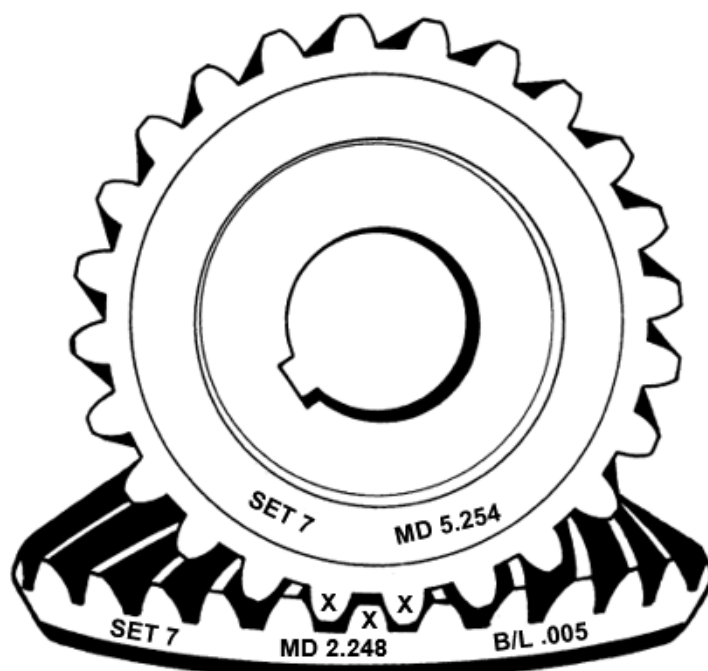
When mounting distances are marked on the gears, and when provisions are made for shimming, the assembler should shim to achieve these mounting distances. These adjustments eliminate the effects of axial tolerance accumulations in both the gears and mountings. It may be noted that shimming cannot correct the shaft angle errors or shaft offset

errors. In cases where mounting distances have not been marked on the gears, assembly must be carried out by trial and error method of marking teeth with gear marking compound and adjusting the gears to obtain a suitable tooth contact pattern.



As shown in above figure, the distance from a locating surface on the back of one gear to the centerline of a mating gear is called the mounting distance.

The manufacturer establishes the optimum value for mounting distances by running the gear set in bevel gear testing machine and adjusting its position to obtain a tooth contact pattern that is consistent with smooth running and optimum load distribution between mating gear teeth. Because of dimensional variations between parts, each gear in a set has a unique value for the mounting distance and in most cases the manufacturer permanently marks this value on each gear as shown in the following figure.



Typical Markings on a Bevel Gear Set

As shown in above figure, the mounting distance is usually shown as “MD” followed by the actual dimension (All dimensions shown in this example are in inch).

The minimum amount of total backlash of a bevel gears set is measured at the tightest point of mesh. This value is generally marked on one or both members of a set of gears. When a value of backlash is marked on only one member of the gear set, this is the total backlash requirement for the gear set. When total backlash is marked on both members to insure visibility of the number after assembly in the gear housing, the value should then be designated on each member as total backlash, so they will not be added during assembly. The amount of backlash is denoted by the markings, for example B/L .005. Unless otherwise specified, backlash (B/L) is assumed to be normal backlash and cannot be measured in the plane of rotation.

The following table shows recommended values of backlash. However, in many applications these may have been altered to meet specific operating conditions.

Diametral Pitch	Module	Outer Normal Backlash	
		inches	mm
1.00 to 1.25	(25.00 to 20.00)	0.024 - 0.030	(0.61 - 0.76)
2.00 to 2.50	(12.00 to 10.00)	0.012 - 0.016	(0.30 - 0.41)
3.00 to 4.00	(8.00 to 6.00)	0.008 - 0.011	(0.20 - 0.28)
4.00 to 5.00	(6.00 to 5.00)	0.006 - 0.008	(0.15 - 0.20)
5.00 to 6.00	(5.00 to 4.00)	0.005 - 0.007	(0.13 - 0.18)
6.00 to 8.00	(4.00 to 3.00)	0.004 - 0.006	(0.10 - 0.15)
8.00 to 10.00	(3.00 to 2.50)	0.003 - 0.005	(0.08 - 0.13)
10.00 to 16.00	(2.50 to 1.50)	0.002 - 0.004	(0.05 - 0.10)
16.00 to 25.00	(1.50 to 1.00)	0.001 - 0.002	(0.03 - 0.05)

Some bevel gears are lapped in sets to improve their operation. In such cases, after optimum tooth contact is obtained in the running tests, the manufacturer marks mating teeth in engagement for identification. These marks usually consist of x's (as shown in above figure) or dots, on two adjacent teeth of one gear and the mating tooth of the other gear. When assembling the gear set, one should position the single marked tooth in the space between the two adjacent marked teeth. When checking backlash, the set of gears should be rotated to the position where the marked teeth are engaged.

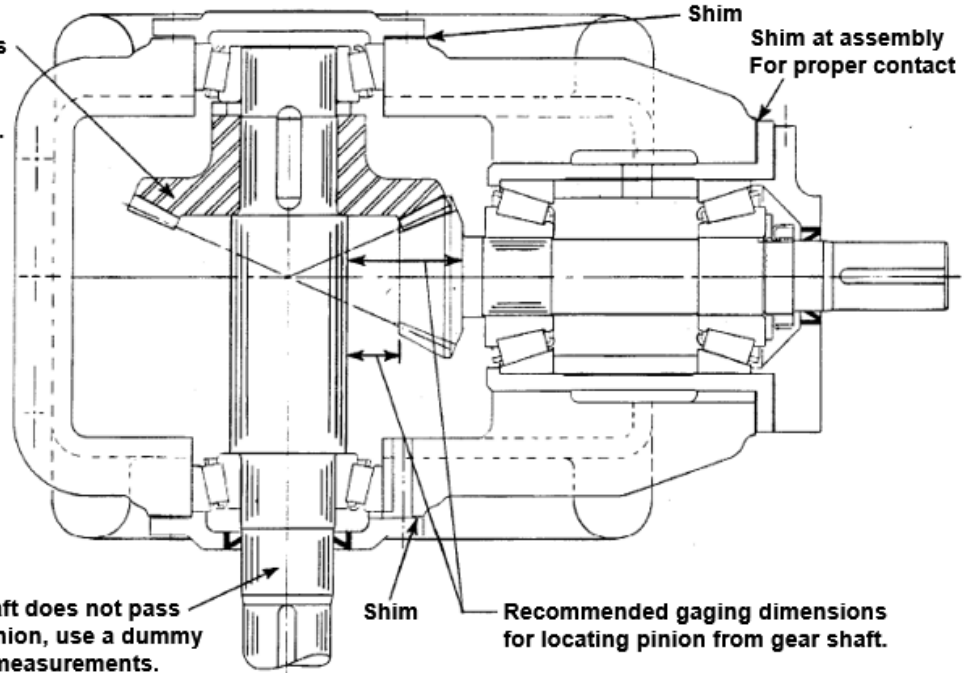
While the teeth of bevel gears are manufactured to close tolerances, they are marked with a serial number on each member of a set of gears to assure matched identification, for example, SET 7 (in above figure). Gears manufactured in sets should only be assembled with their correct mates. It is also recommended that if one gear in a set fails, the entire set should be replaced.

Assembly

Pinion and gear shaft bearings must be set to specified clearances before measuring mounting distances or backlash.

It is advisable to first adjust the pinion to its correct mounting distance. The pinion shaft assembly, including bearings, is usually contained in a cylindrical cartridge. Pinion mounting distance is set by adjusting thickness of shims between cartridge flange and gear drive housing. It is recommended that a gauge be used to set the pinion member to its proper mounting distance. When a gauge is not available, the pinion may be located axially by measuring from the locating surface to the axis of the mating gear. The reference points for the measurement are shown in the following figure.

After the pinion is mounted, the gear is located by setting it to have the proper amount of backlash.



Reference Points for Pinion Assembly

In most bevel gear sets, particularly those with ratios above 2:1, the pinion position (mounting distance) effects tooth contact to a larger extent than the gear position. While the gear position has a larger effect on backlash.

Hence, after the pinion is in position, the proper position of the gear may be determined by measuring the backlash. If backlash does not conform to specifications, gear member must be repositioned axially.

To repositioning the gear member, remove caps on housing at shaft ends. Remove shims from one end of shaft as needed, and add same thickness to opposite end. Replace caps. This procedure allows repositioning of gear/shaft assembly while maintaining bearing clearances.

However, both members control backlash and tooth contact pattern equally when gear set is having 1:1 ratio. Hence when gear ratio is 1:1, each member must be adjusted.

After the backlash is corrected, carry out the tooth contact pattern check. The settings are acceptable only when the tooth contact pattern and backlash are both satisfactory.

It may be noted that when mating gears are adjusted to their optimum position, their back angles, will probably not be flush with each other. Do not attempt to position bevel gears by making the back angles flush.

However, in the case of large bevel gears where direct measurement of mounting distance is difficult, flush surface may be ground on the back cone faces (back angles) of the gear and pinion when in proper position on the testing machine. In such cases, when the gears are assembled, they must be positioned so that the ground areas on the back cone surfaces are flush. These surfaces are usually marked "O" (like lapped gears having "X" marked on them).

Inspection and Maintenance Intervals

Recommended intervals for inspection and maintenance activities are as under.

Interval	Inspection and Maintenance Activities
Every month	Visual inspection for cleanliness of the gearbox surfaces.
Every year	Check coupling alignment. Check end play of input and output shafts. Check backlash. Visually inspect gear elements (from inspection cover).
Every X operating hours or every 18 months	Change lubricating oil. This interval should be shortened if the gearbox is operating under extreme conditions (high humidity, extreme temperature changes, aggressive environment).
Every 10 years	Complete overhaul. This interval should be shortened if the gearbox is operating under extreme conditions (high humidity, extreme temperature changes, aggressive environment).

Troubleshooting

This section provides troubleshooting tips for gearboxes. Following table provides an overview. For detailed information, refer to the text following the table.

Problem	Possible Cause	Remedy
Abnormally high temperature	Housing coated with foreign material.	Clean outside of housing.
	Oil cooler problems.	Check coolant flow (pressure). Vent air from system. Check cooler for internal deposits from coolant water.
	High ambient temperature.	Provide adequate ventilation.
	Lack of oil to bearings and/or mesh.	Check lubrication system.
	Excessive lubrication.	Check lubricant level and adjust down to recommended levels.
	Oil oxidized or dirty.	Drain, flush and fill with clean oil.
	Wrong lubricant.	Drain, flush out and refill with correct lubricant as recommended.
	Bearings improperly adjusted.	Check that bearings are not pinched (radial clearance should be correct). Check bearings for proper axial (lateral) clearance and adjust if necessary.
	Overloading.	Check rated capacity of the gearbox, replace with unit of sufficient capacity or reduce load.
Insufficient thermal HP capacity.	Check and replace with gearbox of sufficient capacity or provide more cooling.	
Low oil pressure (for pressurized lubrication)	Use of lubricant with lower viscosity than required.	Use correct viscosity lubricant.
	Low lubricant viscosity from high lubricant temperatures	See "Abnormally High Temperature".
	Clogged oil filter.	Replace filter element.
	Pump cavitation.	Maintain proper oil level in reservoir.

	Air leak in suction line.	Check and tighten all pipe fittings.
	Incorrect relief valve setting.	Set relief valve correctly.
Bearing failures	Gearbox overloaded.	Reduce loading or replace with gearbox of sufficient capacity.
	Overhung load excessive.	Reduce overhung load.
	Coupling misalignment.	Realign coupling.
	Bearings improperly adjusted.	Check that bearings are not pinched (radial clearance should be correct). Check bearings for proper axial (lateral) clearance and adjust if necessary.
	Inadequate lubrication.	Investigate and correct problem in the lubrication system.
Oil leakage	Worn seals.	Caused by dirt or grit entering seal. Replace seals.
	Overfilled.	Check lubricant level and adjust to recommended level.
	Air breather clogged.	Clean or replace the air breather.
	Improper mounting position.	Correct mounting position.
Unusual / Excessive noise	Worn parts (e.g. bearings, gears).	Pinpoint noise with mechanic's stethoscope, replace parts.
	Coupling misalignment.	Realign coupling.
	Transmission from other equipment.	Add sound blanket or enclosure.
Excessive vibrations	Insufficient foundation rigidity.	Reinforce foundation.
	Loose foundation bolting.	Tighten the bolts.
	Dynamic instability (critical speed).	Design to attenuate critical speeds in operating range.
	Unbalanced parts.	Determine which parts require balancing and balance them.
No Sensor Readings	No power.	Check power supply & repair or restore.
	Failed sensor.	Replace sensor.
	Lead wire braid rubbed through; wire contacting metal.	Replace wire / sensor.

Variance from Design Parameters

The first step when troubleshooting of a gearbox is to determine if there is any variance from design parameters. Specifically, to determine if the gearbox was used in an application suitable for its design and whether it was exposed to different design parameters (for example, torque, rotational velocity, and so on) or ambient conditions than those specified by the manufacturer.

Abnormally High Temperature

Oil level too high. - If the oil level in a gearbox is so high that the gear runs in the oil, then the resulting churning action will heat the oil.

Coated housing. - If the gear housing gets coated with a foreign material (e.g. thick dust layer) that will not permit natural heat removal by convection, high temperature may result. To prevent this, the drive should be cleaned periodically.

Hot weather. - Obviously, a high ambient temperature will cause abnormally high oil temperature. To prevent this, provide adequate ventilation around the drive.

Low oil pressure (in case of a forced lubrication system). - If the oil flow to the bearings and gear mesh is below normal (indicated by below normal oil pressure), the heat created by friction at the mesh and bearings will cause abnormally high temperatures. To correct this situation, check the lubrication system for proper operation.

Low Oil Pressure

Use of a lubricant which has a viscosity less than that for which the lube system was designed - Many times several orifices are installed in the lube system. They are sized for lubricants with a particular viscosity. A lubricant with less than this normal viscosity will pass through the orifices without building up pressure. This situation can be prevented by using the lubricant designated on the name plate of the gear unit. Abnormally low viscosity may also result from high lubricant temperatures.

Clogged oil filter - Replacing the filter will allow more oil to flow through it, thus bringing the oil pressure back to normal.

Pump cavitation - Should the oil level in the reservoir get so low that the pump suction line sucks both air and oil, then the oil pressure will drop. This problem may be cured by maintaining proper oil level in the reservoir.

Air leak in the suction line to the pump - This situation is similar to pump cavitation in that air gets in the oil and results in low oil pressure. To remedy this problem, check and tighten all pipe fittings in the suction line.

Incorrect relief valve setting - Adjusting the relief valve setting properly will avoid venting the pump discharge line back to the sump.

Unusual or Excessive Noise

Worn parts - One common cause of unusual noise is worn parts. If a part wears enough to cause slack in the system, the slack may be heard as a rattle or noise of some sort. A mechanic's stethoscope may be used to pinpoint the worn part which should be replaced.

Misalignment - A coupling that is out of alignment may also cause noisy operation. The misaligned coupling causes misalignment in the gear train which then produces noise or vibrations. The coupling should be immediately realigned before damaging wear occurs.

Transmitted sound - Occasionally other machinery or equipment may be transmitting excessive noise. Enclose one or the other or use a sound blanket.

Excessive Vibration

Soft foundation - A foundation that is not sufficiently rigid may cause vibration problems. To correct this, reinforce the foundation.

Critical Speeds - At certain speeds, a rotating shaft will become dynamically unstable and the resulting vibrations and deflections that occur might cause damage to the gear unit. The speeds at which the shaft becomes unstable are called critical speeds. They are a function of the shaft geometry and the type and spacing of the supporting bearings. If such dynamic instability is suspected, the manufacturer should be consulted.

No Sensor Readings

Various sensing devices for temperature and vibration are installed to provide warnings that can prevent catastrophic failure. If no readings are being received, check for the following.

No power - Check that the power supply to the devices is on.

Failed equipment - Check that the monitoring or recording equipment is functioning.

Worn wires - Visually inspect lead wire to ensure that moving parts are not rubbing through the lead wires and causing shorting out.

Note

One of the primary causes of gear failure is the fact that, with few exceptions, gear sets are designed for operation in one direction only. Failure is often caused by inappropriate bi-directional operation of the gearbox or backward installation of the gear set. Unless specifically manufactured for bi-directional operation, the “non-power” side of the gear’s teeth is not finished. Therefore, this side is rougher and does not provide the same tolerance as the finished “power” side.

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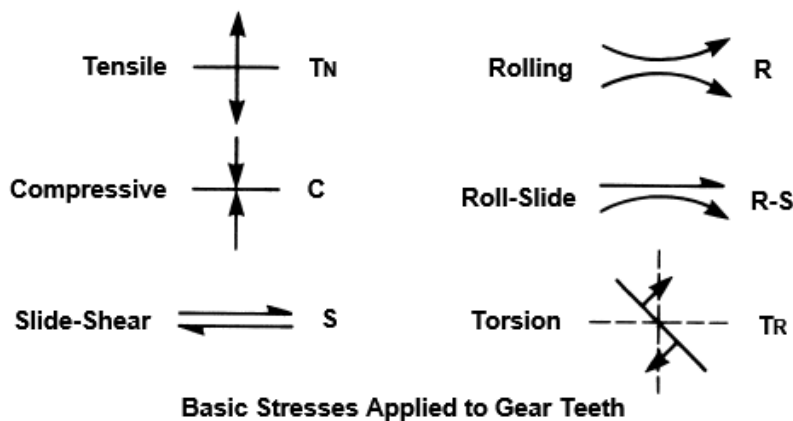
Gear Wear and Failure Modes

One may say that a gear has failed when it can no longer efficiently do the job for which it was designed. Cause of failure may range from excessive wear to catastrophic breakage. Failure in a gear drive can in many cases be prevented. However, when a failure occurs, it is important to correctly identify the failure mode so that the appropriate actions can be taken to prevent a reoccurrence of the same type of failure.

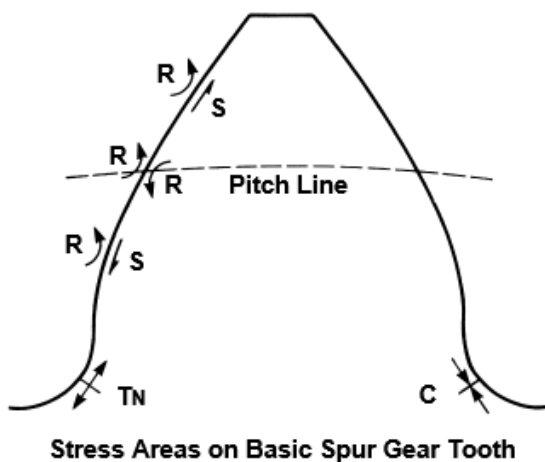
Gear drive failures can be caused by fundamental design issues, manufacturing defects, deficiencies in the lubricant or lubrication system, excessive time at standstill, high loading, and many other reasons. In view of this, information on gear failure modes is given in this chapter.

Basic Applied Stresses

The loads applied to one tooth by the action of its mating tooth are at any moment of time a line contact at the most; or, at the least, a point contact. As the loads are increased, the line may lengthen or even broaden, or the point may expand to a rounded area.



The basic stresses applied to a gear tooth include the six types listed in above figure. They are tensile, compressive, shear (slide), rolling, rolling-slide, and torsion. Often, a combination of two or three types is applied at a time.



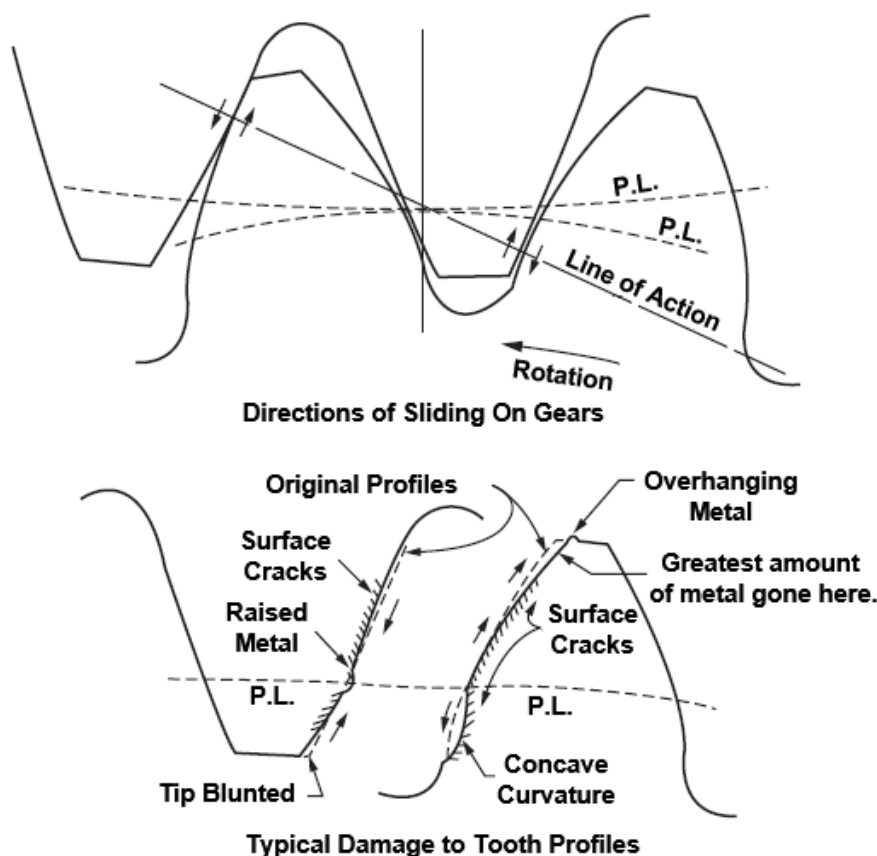
In a spur gear, as the contacting tooth moves up the profile of the loaded tooth, a sliding-rolling action takes place at the profile interface. At the pitch line, the stresses are pure rolling. Above the pitch line, the rolling-sliding action again takes over, but the sliding will be in the opposite

direction as shown in above figure. However, the action on the profile of the contacting tooth will be exactly the same as the loaded tooth except in reverse order. The sliding action of two surfaces, when lubricated properly, will have no problem. However, surface disparities, insufficient lubrication, improper surface hardness, higher temperatures, and abrasive or adhesive foreign particles will contribute to a breakdown during a sliding contact. At the same time, there is a tensile stress at the root radius of the loaded side of the tooth and a compressive stress at the root radius of the opposite side.

The helical gear tooth receives the same contact action as the spur gear. However, in a helical gear tooth, an additional stress is being applied to the helical tooth. In helical gear tooth, a lateral sliding action is applied at all contact levels, including the pitch line. The force component at 90° to the direction of rotation increases as the helical angle increases.

One additional stress common to all gearing is because of rolling surfaces. It is a shear stress running parallel to the surface at a distance from 0.18 to 0.3 mm (0.007 to 0.012 in.) below the surface. The distance below the surface given above is the average depth for a normal loading condition. The subsurface shear stress is most often the originator of initial line pitting along the pitch line of gear teeth.

Pitting, like tooth breakage, is a fatigue failure. Helical pinions of medium hardness with 20 or more teeth frequently pit along the pitch line. The mating gear may also pit, but if it is about the same hardness as the pinion and has been heat treated in the same way as the pinion, it is likely that most of the pitting will be on the pinion.



Pinions are more apt to pit than gears for two reasons: First, the pinion is ordinarily the driver. The directions of sliding are such that sliding is away from the pitch line on the driver and toward the pitch line on the driven member. Above figure shows how the sliding motion on the driver tends to pull metal away from the pitch line. This leaves the pitch line high and also

tends to stretch the metal at the pitch line. On the gear, the sliding tends to compress the metal at the pitch line. The cracks that form when a surface is severely loaded have a tendency to intersect at the pitch line of the driver, while on the driven member, they do not. Second, the pinion, being smaller, has more cycles of operation than the gear. The slope of the fatigue curve makes the part with the most cycles the most apt to fail.

When the gear drives the pinion, the cycles favor the pinion failing first, but the sliding action is worst on the gear. Tests on a 4:1 ratio helical gears of medium hardness showed that the gear would usually pit first when it was driving the pinion and made of the same hardness material as the pinion.

Classification of Gear Failure Modes

Several failure analysts have classified gear failure modes in different ways.

Gear Failure Modes (As per ANSI/AGMA 1010-E95)		
Class of Failure	General Mode	Specific Mode or Degree
Wear	Adhesion	Mild and Moderate, Severe (see Scuffing)
	Abrasion	Mild, Moderate and Severe
	Polishing	Mild, Moderate and Severe
	Corrosion	
	Fretting Corrosion	
	Scaling	
	Cavitation	
	Erosion	
	Electrical Discharge	
	Ripplings	
Scuffing	Scuffing	Mild, Moderate and Severe
Plastic Deformation	Plastic Deformation	Indentation
		Cold flow
		Hot flow
		Rolling
		Tooth hammer
		Rippling
		Ridging
		Burr
		Root fillet yielding
		Tip-to-root interference
Contact Fatigue	Pitting (Macropitting)	Initial, Progressive, Flake and Spall
	Micropitting	
	Subcase Fatigue	
Cracking	Hardening Cracks	
	Grinding Cracks	
	Rim and Web Cracks	
	Case/Core Separation	
	Fatigue Cracks	
Fracture	Brittle Fracture	
	Ductile Fracture	
	Mixed Mode Fracture	
	Tooth Shear	
	Fracture after plastic deformation	
Bending Fatigue	Low-cycle Fatigue	
	High-cycle Fatigue	

As shown in above table, the American Gear Manufacturers Association (AGMA) has classified gear failure modes under the broad categories of wear, scuffing, plastic deformation, contact fatigue, cracking, fracture and bending fatigue.

Detail information on the most common gear failure modes is given in the following sections.

Wear

Wear is a term describing change to a gear tooth surface involving the removal or displacement of material due to mechanical, chemical, or electrical action.

The amount of wear considered tolerable depends on the expected lifetime for the gears and requirements for control of noise and vibration. Wear is considered excessive when the tooth profiles wear to the extent that high dynamic loads occur or the tooth thickness is reduced to the extent that bending fatigue becomes possible.

Adhesion

Adhesion is caused by the transfer of material from one tooth surface to another due to micro welding and tearing. It is confined to surface films and oxide layers on the tooth surface. Adhesion can be categorized as mild or moderate. Severe adhesion is termed scuffing.

Mild Adhesion

Mild adhesion typically occurs during running-in and usually subsides after it has smoothed the tooth surfaces by removing minor imperfections through local wear. To the unaided eye, the tooth surface appears undamaged and the original machining marks are visible. Microscopically, smooth micro plateaus can be seen between the machining furrows.

When new gear drives are first operated, the contact between the gear teeth is not optimum because of unavoidable manufacturing inaccuracies. If the tribological conditions are favorable, mild adhesive wear occurs during running-in and usually subsides with time, resulting in a satisfactory lifetime for the gears. The wear that occurs during running-in is beneficial if it smooths the tooth surfaces (thereby increasing the specific film thickness) and if it increases the area of contact by removing minor imperfections through local wear. In view of this, to ensure that the wear rate remains under control, new gear drives should be run-in by being operated for at least the first ten hours at one-half load.

Moderate Adhesion

Adhesion is classified as moderate if it removes some or all of the original machining marks from the active surface of the tooth. Under certain conditions, adhesion can cause continuous removal of surface films and oxide layers, resulting in severe wear.

Abrasion

Abrasion is the removal or displacement of material due to the presence of hard particles (for example, metallic debris, scale, rust, sand, or abrasive powder) suspended in the lubricant or embedded in the flanks of the mating teeth. Based on the severity of the damage, abrasion can be categorized as mild, moderate, or severe.

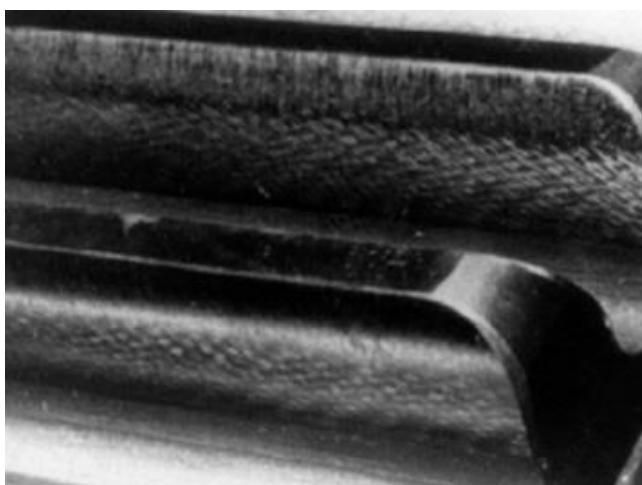
Abrasion causes scratches or gouges on the tooth surface that are oriented in the direction of sliding. Abrasion normally appears at the addendum and dedendum where sliding is present. Under magnification, the scratches appear as parallel furrows that are smooth and clean.

Abrasion due to loose contaminants is called three-body abrasion. Two-body abrasion occurs when embedded particles or asperities on one gear tooth abrade the opposing tooth surface.

Abrasive wear on gear teeth is usually caused by contamination of the lubricant by hard, sharp-edged particles. Contamination enters gearboxes by being built-in, internally generated (wear debris from gears or bearings), ingested through breathers and seals, or inadvertently added during maintenance.

It is not uncommon to find sand, machining chips, grinding dust, weld splatter, or other debris in new gearboxes. To remove built-in contamination, the gearbox lubricant should be drained and flushed before start-up.

Mild Abrasion



Mild Abrasion near the Tip of a Ground Gear

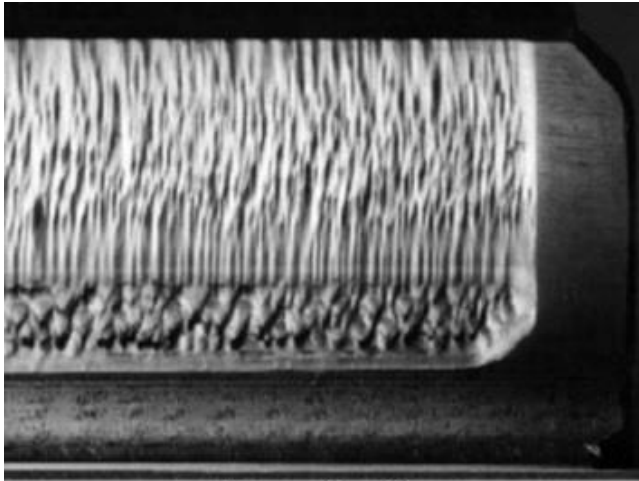
Abrasion is classified as mild if it consists of fine scratches that are not numerous or deep enough to remove significant amounts of material from the tooth surface. Above figure illustrates an example of mild abrasion.

Moderate Abrasion

Abrasion is classified as moderate if remnants of the original machining marks are visible on the tooth surface.

Severe Abrasion

Severe abrasion removes all of the original machining marks from the active surface of the tooth. There might be wear steps at the ends of the active face and in the dedendum. The tooth thickness might be reduced significantly, and, in some instances, the tooth tip might be reduced to a sharp edge. The following figure illustrates an example of severe abrasion.



Severe Abrasion

Polishing

Polishing is a fine scale abrasion that causes gear teeth to have a bright mirror like finish. The gear tooth surface might be smooth or wavy with local bumps. Under magnification, the surface appears to be covered by fine scratches that are oriented in the direction of sliding. When a hard surface mates with a soft surface, polishing is more likely to occur on the hard surface. Polishing is promoted by chemically active lubricants that are contaminated with fine abrasives. Based on the severity, polishing can be categorized as mild, moderate, or severe.

Although the polished gear teeth may look good, polishing wear is undesirable because it generally reduces gear accuracy by wearing the tooth profiles away from their ideal form.

Antiscuff additives used in lubricants to prevent scuffing, such as sulfur and phosphorus, function by forming iron-sulfide and iron-phosphate films on areas of the gear teeth where high temperatures occur. Ideally, the additives should react only at temperatures where there is a danger of welding. If the rate of reaction is too high, and there is a continuous removal of the surface films caused by very fine abrasives in the lubricant, the polishing wear may be excessive. To prevent polishing wear, it is recommended to use less chemically active antiscuff additives (for example, borate).

Mild Polishing

Polishing is classified as mild if it is confined to the peaks of the surface asperities. Mild polishing typically occurs during running-in and ceases before the original machining marks are removed from the tooth surface.

Moderate Polishing

Polishing is classified as moderate if remnants of the original machining marks are visible on the tooth surface.

Severe Polishing

Severe polishing removes all of the original machining marks from the active surface of the tooth. The polished surface might be wavy and there might be wear steps at the ends of the active face and in the dedendum.

Corrosion



Extensive Corrosion

Corrosion is the chemical or electrochemical reaction between the surface of a gear and its environment. The tooth surfaces might appear stained or rusty and there might be reddish-brown deposits of rust. If the loose corrosion products are removed, etch pits might be revealed. Corrosion commonly attacks the entire tooth surface and it might proceed intergranularly by preferentially attacking the grain boundaries of the tooth surfaces. Above figure illustrates an example of corrosion.

Fretting Corrosion



Fretting Corrosion

Fretting corrosion occurs between contacting surfaces that are pressed together and subjected to cyclic, relative motion of extremely small amplitude. Under these conditions, the lubricant is squeezed from between the surfaces, permitting metal-to-metal contact and causing adhesion of the surface asperities. The relative motion breaks the welded asperities and generates iron oxide powder, which has the fineness and reddish-brown color of cocoa. The wear debris is hard and abrasive. Fretting corrosion tends to be self-aggravating because the wear debris builds a dam that prevents fresh lubricant from reaching the contact area. Fretting corrosion can occur in nonrotating gears if they are subjected to structure-borne vibrations such as those encountered during transport. Above figure illustrates an example of fretting corrosion.

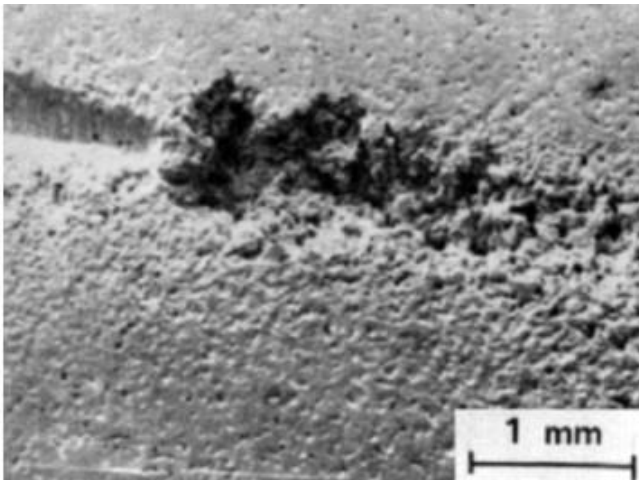
Scaling



Scaling

The patchy raised areas on the tooth flanks in above figure are due to an oxidation process during heat treatment. When running under load, the tooth force is initially transmitted by way of these projections, which rapidly acquire a metallic sheen. This phenomenon is known as scaling.

Cavitation

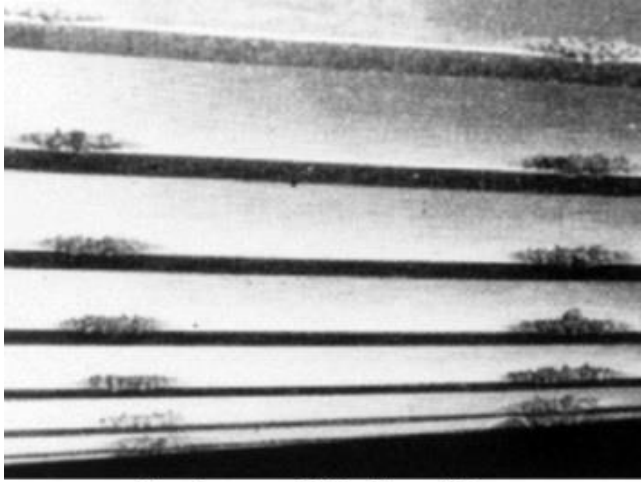


Cavitation Damage Showing Sandblasted Appearance

Cavitation is the nucleation and implosion of bubbles in the lubricating fluid. This can cause damage in the gear tooth surface, which appears to the unaided eye to be rough and clean as if it were sandblasted. Microscopically, the craters caused by cavitation are deep, rough, clean, and have a honeycomb appearance. Above figure illustrates a gear that has undergone cavitation.

Erosion

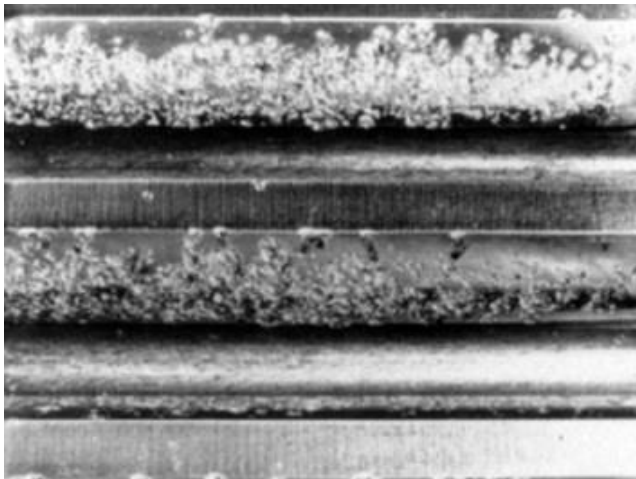
Erosion is the loss of material from a gear tooth surface due to the relative motion of a high velocity fluid. The following figure shows erosion on a high-speed gear.



Erosion on a High-Speed Gear

Electrical Discharge (Stray Shaft Currents)

An electric arc discharge across the oil film between mating gear teeth produces temperatures that can be high enough to locally melt the gear tooth surface. The sources of the electric arc can be stray shaft currents from the generator/gearbox driver or from welding apparatus improperly grounded to the gearbox frame.



Severe Electrical Discharge Damage

Microscopically, the damage appears as small hemispherical craters. The edges of the craters are smooth, and they might be surrounded by burned or fused metal in the form of rounded particles that were once molten. Sometimes micro-cracks are found near the craters. Above figure shows severe electrical discharge damage due to an electric current of high intensity.

Rippling

Rippling is generally associated with plastic deformation, but it can also be a form of wear.

Scuffing

Scuffing is severe adhesion that causes the transfer of metal from one tooth surface to another by means of welding (solid-phase welding caused by intense frictional heat generated by the combination of high sliding velocity and intense surface pressure) and tearing. The scuffed

areas appear to have a rough or matte texture. The damage typically occurs in the addendum, dedendum, or both, away from the operating pitch line and in narrow or broad bands that are oriented in the direction of sliding. Scuffing can occur in localized patches. Under magnification, the scuffed surface appears rough, torn, and plastically deformed.

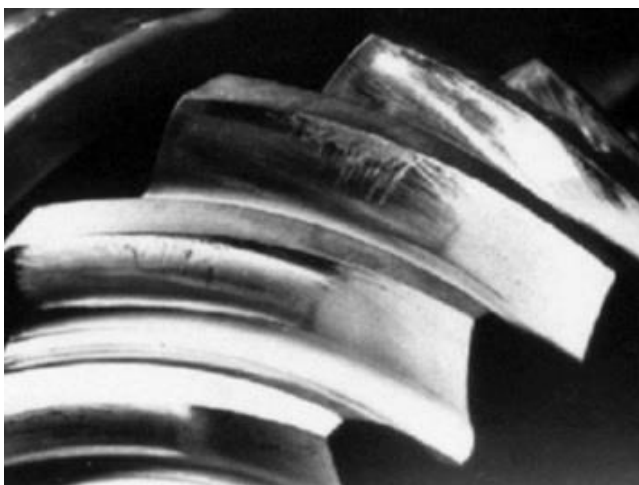
Scuffing can occur in gear teeth when they operate in the boundary lubrication regime. If the lubricant film is insufficient to prevent significant metal-to-metal contact, the oxide layers that normally protect the gear tooth surfaces may be broken through, and the bare metal surfaces get welded together. The sliding that occurs between gear teeth results in tearing of the welded junctions, metal transfer, and damage.

In contrast to pitting and bending fatigue, which only occur after a period of running time, scuffing may occur immediately upon start-up. In fact, gears are most vulnerable to scuffing when they are new and their tooth surfaces have not yet been smoothed by running-in. For this reason, it is wise to run-in a new gearbox under one-half load for at least 10 hours to reduce the surface roughness of the teeth before applying full load. The gear teeth can be coated with iron manganese phosphate or plated with copper or silver to protect them from scuffing during the critical running-in period.

Most antiscuff additives are sulfur-phosphorus compounds, which form boundary lubricating films by chemically reacting with the metal surfaces of the gear teeth at local points of high temperature. Antiscuff films help prevent scuffing by forming solid films on the gear tooth surfaces and inhibiting true metal-to-metal contact. The films of iron sulfide and iron phosphate have high melting points, allowing them to remain as solids on the gear tooth surfaces even at high contact temperatures. The rate of reaction of the antiscuff additives is greatest where the gear tooth contact temperatures are highest. Because of the rubbing action of the gear teeth, the surface films are repeatedly scraped off and reformed. As sulfur-phosphorus compounds are corrosive, in effect, scuffing is prevented by substituting mild corrosion in its place.

The term scoring, which was incorrectly used in earlier gear nomenclature for scuffing, is, in reality, scratching; it is now classified as a form of abrasive wear. Based on the severity of the damage, scuffing can be categorized as mild, moderate, or severe.

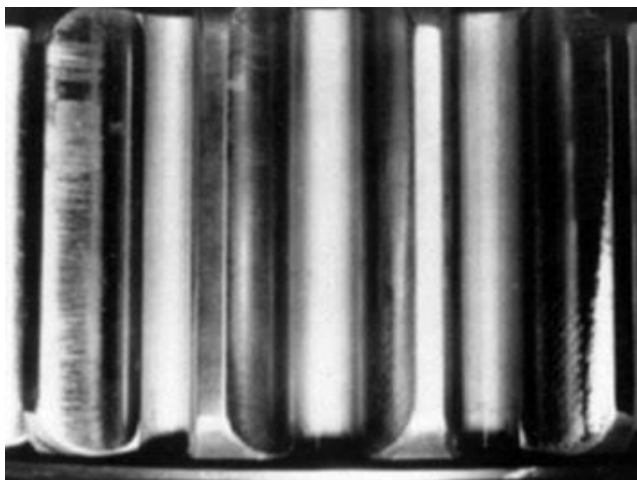
Mild Scuffing



Mild Scuffing

As illustrated in above figure, scuffing is classified as mild if it occurs only on small areas of the teeth and is confined to the peaks of the surface asperities.

Moderate Scuffing



Moderate Scuffing

Moderate scuffing, as shown in above figure, occurs in patches that cover significant portions of the teeth. If the operating conditions do not change, moderate scuffing might be progressive.

Severe Scuffing



Severe Scuffing

Severe scuffing occurs on significant portions of the gear tooth (for example, the entire addendum, the entire dedendum, or both). In some cases, the surface material might be plastically deformed and displaced over the tip of the tooth or into the root of the tooth. Unless corrective measures are taken, severe scuffing is usually progressive. Above figure illustrates an example of severe scuffing.

Plastic Deformation

Plastic deformation is permanent deformation that occurs when the stress exceeds the yield strength of the material. It can occur at the surface or sub-surface of the active flanks of the gear teeth due to high contact stress or at the root fillets of the gear teeth due to high bending stress.

Indentation



Severe Indentations

As shown in above figure, the active flanks of gear teeth can be damaged by indentations caused by foreign material that becomes trapped between mating teeth.

Cold Flow

Cold flow is plastic deformation that occurs at a temperature lower than the recrystallization temperature.

Hot Flow



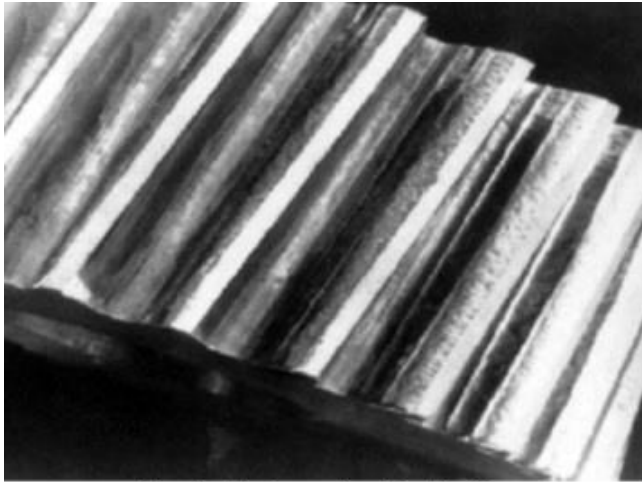
Hot Flow

Hot flow is plastic deformation that occurs at a temperature higher than the recrystallization temperature. Above figure illustrates an example of hot flow.

Rolling

Plastic deformation might occur on the active flanks of gear teeth caused by high contact stresses, in combination with both the rolling and sliding action of the gear mesh. Displacement of surface material might form a groove along the pitch line and burrs on the

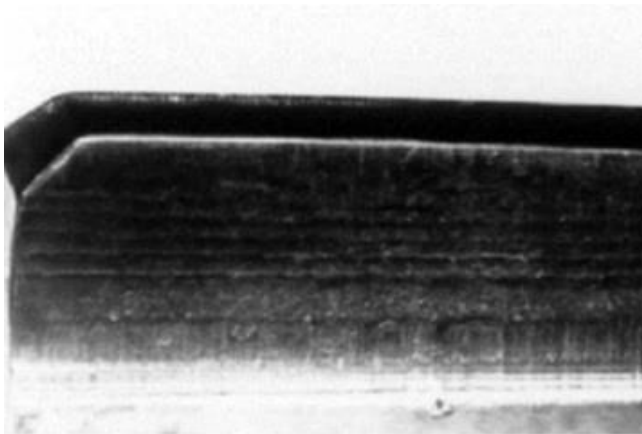
tips and in the roots of the driving gear teeth. The surface material of the driven gear might be displaced toward the pitch line, forming a ridge.



Plastic Deformation by Rolling

Above figure illustrates an example of plastic deformation by rolling.

Tooth Hammer

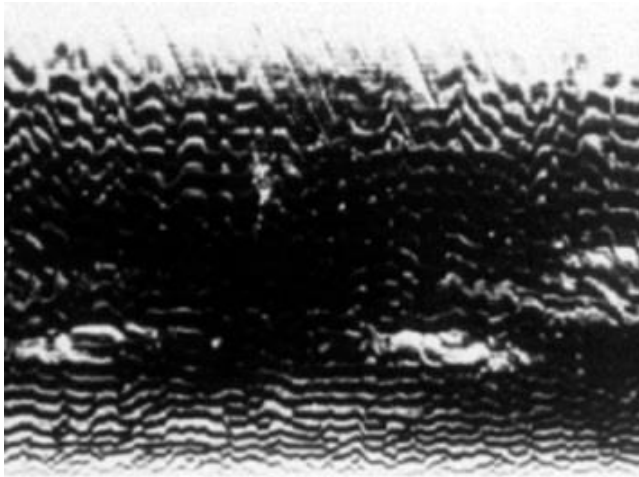


Plastic Deformation by Tooth Hammer

Local, sub-surface yielding might occur on gear teeth that are subjected to high contact stresses, such as those caused by “tooth hammer” (vibratory impact with intermittent tooth contact separation). The sub surface plastic deformation causes shallow grooves (Brinelling) on the surfaces of the active flanks of the gear teeth along lines of contact between mating teeth. This failure mechanism is illustrated in above figure.

Rippling

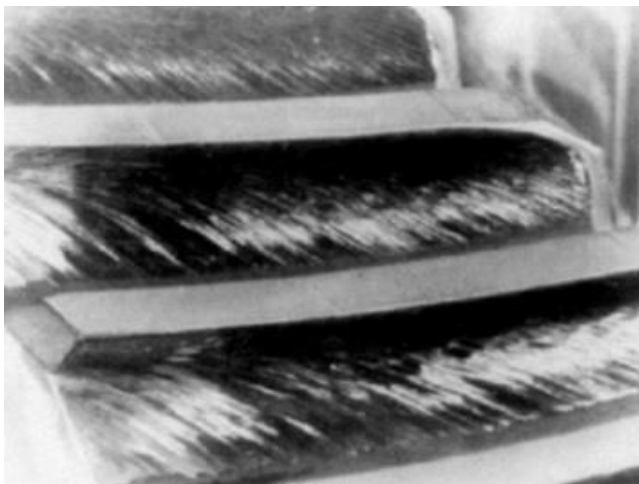
Rippling is the periodic, wavelike deformation of the active gear tooth surface, as shown in the following figure. The peaks of the waves run perpendicular to the direction of sliding. The peaks are wavy along the length of the tooth, creating a fish-scale appearance. Rippling usually occurs under high contact stress and boundary lubricated conditions. Although rippling can be considered a wear phenomenon, it is most often associated with plastic flow.



Rippling

Above figure illustrates an example of rippling.

Ridging



Ridging

Ridging, as illustrated in above figure, is the development of pronounced ridges and grooves on the active flanks of gear teeth. It frequently occurs on slow speed, heavily loaded worm or hypoid gear teeth.

Burr

A pronounced burr can be seen at the tip of the worm thread's working flank in the following figure. This burr was generated by plastic deformation due to the pressure and sliding action along the active surface of the flanks. Burrs are rough, often sharp, extensions formed on the edges of components caused by heavy loading, high friction, rolling, or scuffing. The manufacturing process also sometimes causes burrs.



Pronounced Burr

Root Fillet Yielding

Gear teeth can be permanently bent if the bending stress in the root fillets exceeds the tensile yield strength of the material. If the yielding causes significant spacing errors between teeth that are permanently bent and unloaded teeth that are not, subsequent engagement can result in destructive interference between mating teeth.

Tip-to-Root Interference



Tip-to-Root Interference

Plastic deformation, adhesion, and abrasion can occur on the tooth tips of one gear and in the roots of the mating gear teeth due to tip-to-root interference. The interference can be caused by geometric errors in the profiles of the gear teeth, engagement below the form diameter, inadequate tip or root relief, spacing errors, or insufficient center distance. An example of tip-to-root interference is illustrated in above figure.

Contact Fatigue

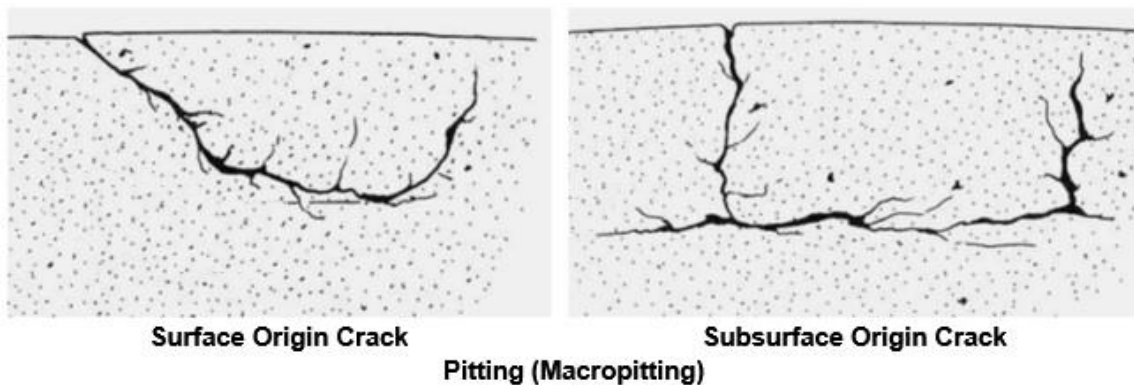
Contact fatigue is the cracking of a surface subjected to alternating Hertzian stresses produced under controlled rolling and sliding loading conditions.

Repeated contact stresses can cause surface or subsurface fatigue cracks and the detachment of material fragments from the gear tooth surface.

Hertzian Stress

Contact stresses appear when two bodies with curved surfaces, e.g. spheres or cylinders, are in contact or when one body with a curved surface is in contact with a plate (because a plate can be thought of as a sphere with an infinite radius). If two spheres were in contact the theoretical contact area would be a point and the theoretical contact area between two cylinders would result in a line. The theoretical contact stresses in both cases would thus be infinite due to the minimal contact area and thereby initiate immediate yielding. This is not the case in reality since the contact areas of the bodies are elastically deformed which implies that the stresses are limited. These stresses are known as Hertz stresses.

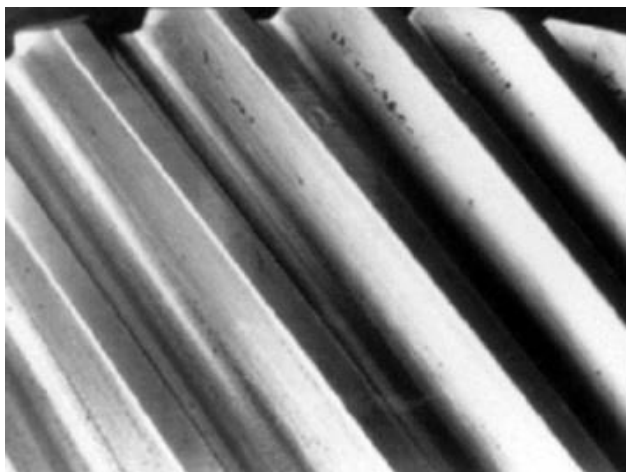
Pitting (Macropitting)



As shown in above figure, macropitting can occur when fatigue cracks initiate either at the surface of the gear tooth or at a shallow depth below the surface. The crack usually propagates for a short distance in a direction roughly parallel to the tooth surface before turning or branching to the surface. When the cracks have grown long enough to separate a piece of the surface material, a pit is formed. The edges of a pit are usually sharp and angular. Cracks can be found near the boundary of the pit and fatigue “beach marks” might be evident on the crater bottom.

Based on the nature and severity of the damage, macropitting can be categorized as initial (non-progressive), progressive, flake, or spall.

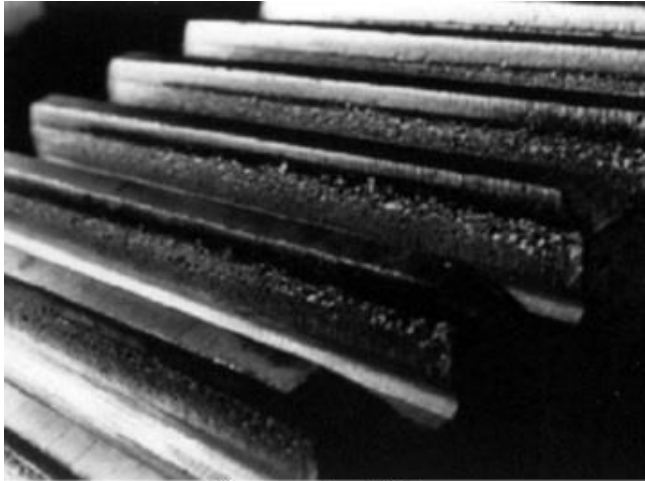
Initial Pitting



Initial Pitting

Non-progressive macropitting normally consists of small pits less than 1 mm in diameter. They occur in localized areas and tend to redistribute the load by removing high asperities. When the load is more evenly distributed, the macropitting stops. Initial signs of pitting are illustrated in above figure.

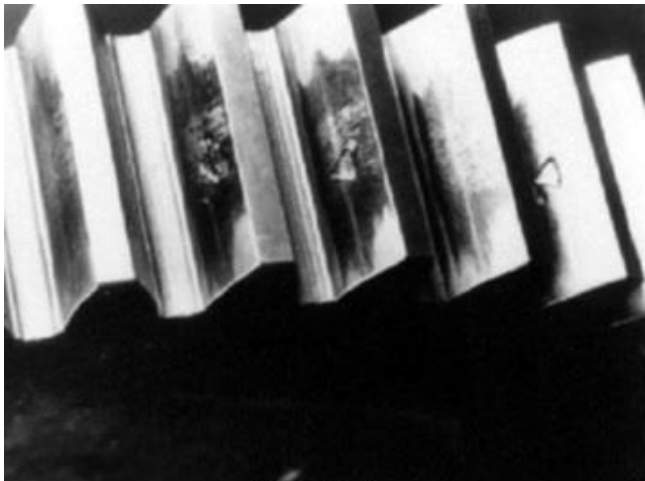
Progressive Pitting



Progressive Pitting

Progressive macropitting normally consists of pits significantly larger than 1 mm in diameter. Pitting of this type might continue at an increasing rate until a significant portion of the tooth surface has pits of various shapes and sizes. More progressed pitting is illustrated in above figure.

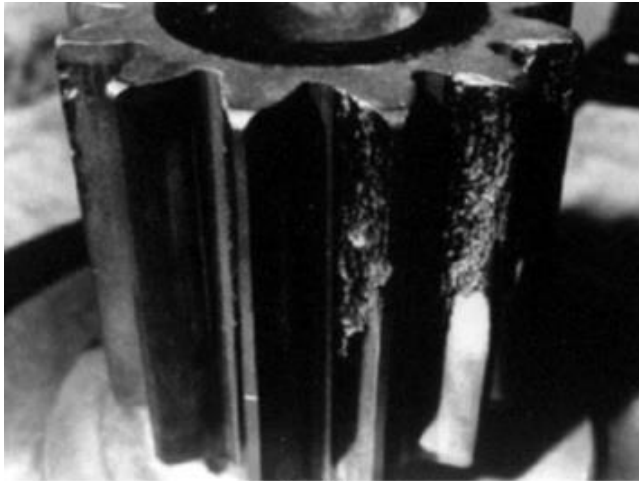
Flake Pitting



Flake Pitting

Flake macropitting consists of pits that are relatively shallow but large in area. The fatigue crack extends from an origin at the surface of the tooth in a fan-shaped manner until thin flakes of material break out and form a triangular crater. Flake pitting is illustrated in above figure.

Spalling



Spalling

Spall macropitting is progressive macropitting that occurs when pits coalesce and form irregular craters that cover a significant area of the tooth surface. An example of spalling is shown in above figure.

Micropitting



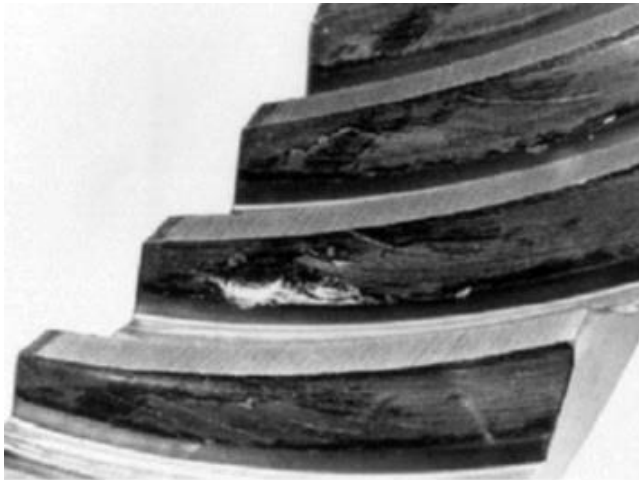
Micropitting

Micropitting gives the gear tooth a frosted, matte, or gray stained appearance. Under magnification, the surface appears to be covered by very fine pits (normally less than 20 μm deep). Metallurgical sections through the micropits show fatigue cracks that are inclined to the surface at an angle of less than 45° . The cracks might extend deeper than the visible micropits. Micropitting occurs most frequently on surface hardened gear teeth, although it might also occur on through hardened gear teeth. It can occur anywhere on the active profile of the gear tooth, as shown in above figure.

Subcase Fatigue

Subcase fatigue can occur in surface hardened gears (for example, carburized, nitrided, induction hardened, and flame hardened). The origin of the fatigue crack is below the surface of the gear teeth in the transition zone between the case and core. Typically, the crack runs parallel to the surface of the gear tooth before branching to the surface. The branched cracks

might appear at the surface as fine longitudinal cracks on only a few teeth. If the surface cracks join together, long shards of the tooth surface might break away. Fatigue beach marks might be evident on the crater bottom formed by propagation of the main crack.



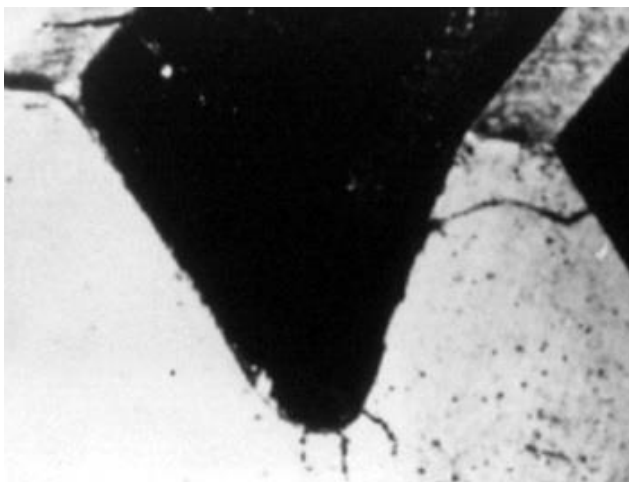
Subcase Fatigue

Above figure illustrates an example of subcase fatigue.

Cracking

Aside from cracks in the gear tooth root fillets caused by bending fatigue, cracks might occur elsewhere on the gear due to mechanical stress, thermal stress, material flaw, or improper processing.

Hardening Cracks

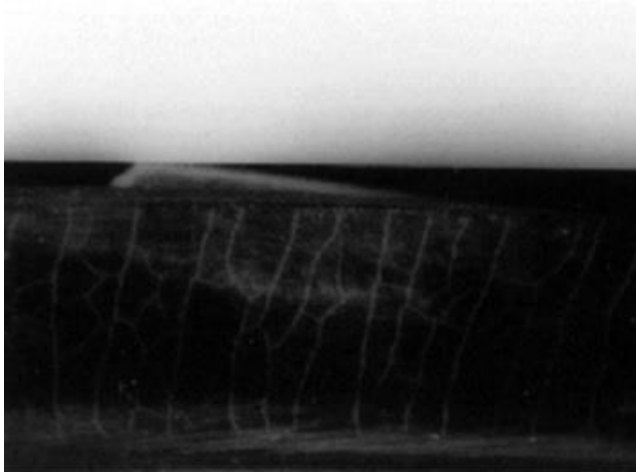


Hardening Cracks

Cracking in heat treatment usually occurs during or after quenching. Hardening cracks are generally intergranular with the crack running from the surface toward the center of mass in a relatively straight line. If the cracking occurs prior to tempering, the fracture surfaces will be discolored by oxidation when the gear is exposed to the furnace atmosphere during tempering. Above figure illustrates an example of gear teeth with hardening cracks.

Grinding Cracks

Cracks can develop on the tooth surfaces of gears that are finished by grinding. The cracks are usually shallow and appear either as a series of parallel cracks or in a crazed, mesh pattern. The cracks can appear immediately after grinding, during subsequent handling or storage, or after time in service. Grinding cracks can result from localized overheating. Areas of the tooth surface where overheating has occurred can be detected by acid etching the surface. Tempered areas appear brown or black on a light brown or gray background. Areas where untempered martensite has formed appear as white areas surrounded by black, tempered areas.

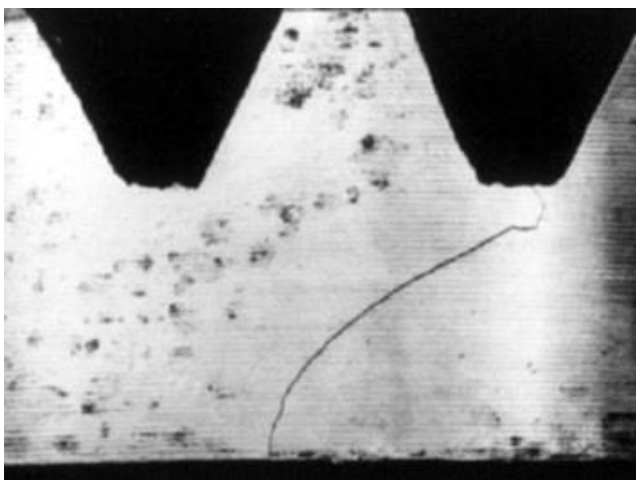


Grinding Cracks with a Crazed Pattern

Magnetic particle or dye penetrant inspection can be used to detect grinding cracks, such as those illustrated in above figure.

Rim and Web Cracks

Rim and web cracks typically occur on welded gears. The rim of the gear is the outermost portion of the item that contains the gear teeth. The web is the portion between the rim and the hub that provide structural support to the rim.

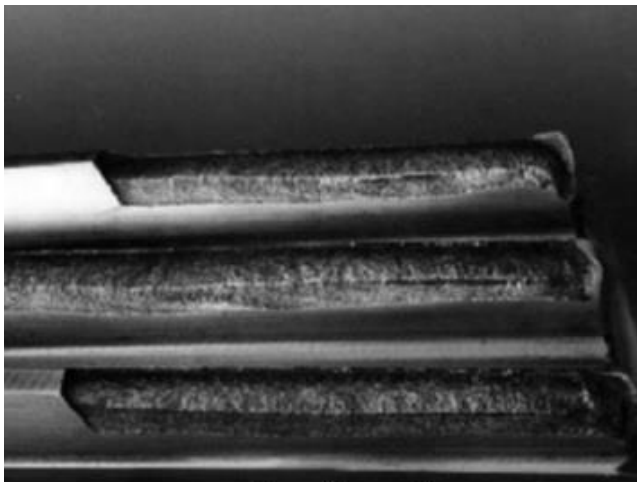


Rim Crack

If the gear rim is thin (for example, less than about twice the gear tooth whole depth), it can be subjected to significant alternating rim bending stresses, which are additive to the gear

tooth bending stress and might result in fatigue cracks in the rim. Rim cracks are similar to tooth bending fatigue cracks, except that rim cracks usually propagate radially through the gear rim, whereas, bending fatigue cracks propagate across the base of the teeth. Rim cracks can grow into the web of the gear. Web cracks can be caused by cyclic stresses due to vibrating loads near a natural frequency of the gear blank. A fatigue crack might originate in the web of the gear and grow into the rim of the gear. Web and rim cracks generally originate at stress concentrations. These concentrations can arise from one or more of the following: sharp corners or notches in the root fillets, keyways, splines, holes, shrink fits, web-to-rim or hub-to-web fillets, or metallurgical defects such as inclusions. Rim or web cracks can cause catastrophic failure in high-speed gears if centrifugal forces cause the fatigue cracks to propagate in a fast fracture mode, tearing open the rim. Above figure illustrates a gear with a severe rim crack.

Case/Core Separation



Case/Core Separation

Case/core separation can occur in surface hardened gear teeth when internal cracks occur near the case/core boundary, as shown in above figure. The internal cracks might propagate causing corners, edges, or entire tips of the teeth to separate. The cracks can appear immediately after heat treatment, during subsequent handling or storage, or after time in service.

Fatigue Cracks



Bending Fatigue Crack

Fatigue cracks are cracks that propagate under the influence of repeated alternating or cyclic stresses that are below the tensile strength of the material. These cracks can appear in tooth flanks and in tooth root fillets. Above figure illustrates cracks caused by the bending of a gear tooth.

Fracture

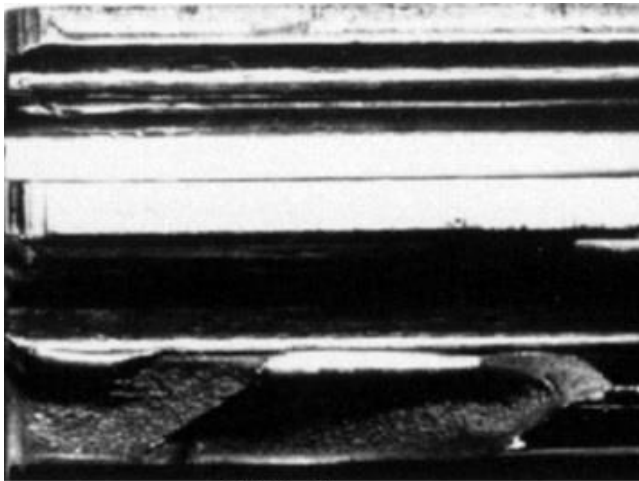
When a gear tooth is overloaded, it might fail by plastically deforming or fracturing. If it fractures, the failure might be a ductile fracture preceded by appreciable plastic deformation, a brittle fracture with little prior plastic deformation, or a mixed mode fracture exhibiting both ductile and brittle characteristics. Fatigue failures usually culminate in a fracture when the fatigue cracks grow to a point where the remaining tooth section can no longer support the load. In this sense, the remaining material is overloaded; however, the fracture is a secondary failure mode that is caused by the primary mode of fatigue cracking. Fractures are classified as brittle or ductile, depending on their macroscopic and microscopic characteristics.

Fracture Modes

Overload breakage usually occurs as a result of a single application, or very few applications, of very high load. Sometimes a crack initiated by an overload will progress as a fatigue crack with slow propagation, usually with evidence of fretting corrosion in the region of the initial crack. There are three types of overload fracture:

- Brittle fracture
- Ductile fracture
- Mixed mode fracture

Brittle Fracture



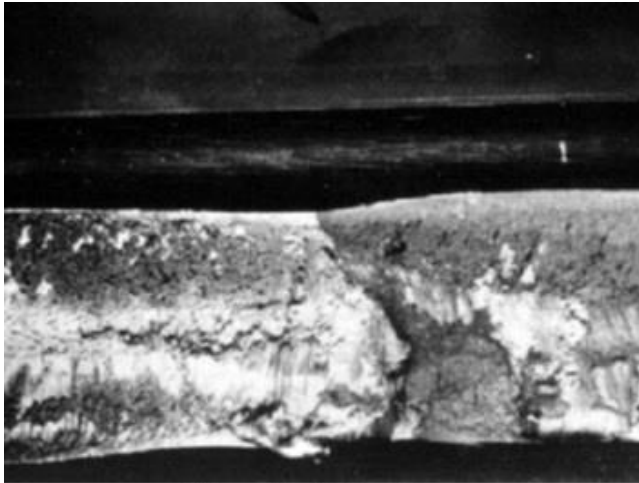
Brittle Fracture

Brittle fractures are characterized by rapid crack propagation without appreciable gross plastic deformation. Brittle fractures have a bright, granular appearance. The fracture surface is generally flat and perpendicular to the direction of the maximum tensile stress. Radial ridges or a chevron pattern may be present on the fracture surface pointing toward the origin of the crack. On a microscopic level, brittle fracture typically consists of transgranular cleavage facets or intergranular facets, as shown in above figure.

Ductile Fracture

Ductile fractures are characterized by tearing of metal accompanied by gross plastic deformation. Ductile fractures have a gray, fibrous appearance. The fracture surface might have a flat or slant orientation to the direction of the maximum tensile stress. The fracture surface might terminate with a shear lip that extends along the non-working side of the gear tooth. Microscopically, ductile fractures are characterized by numerous dimples that are formed by the nucleation and growth of microvoids.

Mixed Mode Fracture



Mixed Mode Fracture

A local area of a fracture surface might exhibit both ductile and brittle characteristics. Under these conditions, the fracture is termed mixed mode. An example of a mixed mode fracture is illustrated in above figure.

Tooth Shear



Tooth Shear

When teeth are sheared from gears, the appearance of the sheared surfaces is similar to that of machined surfaces. As illustrated in above figure, tooth shear is almost always caused by a single severe overload.

Fracture after Plastic Deformation



Fracture after Plastic Deformation

All such fractures begin with gross plastic deformations of the teeth before final breakage. Above figure illustrates an example of fracture occurring after plastic deformation.

Bending Fatigue

Fatigue is a progressive failure consisting of three distinct stages:

- Stage 1: Crack initiation
- Stage 2: Crack propagation
- Stage 3: Fracture

Most of the fatigue life is occupied by Stages 1 and 2 until the cracks grow to critical size where sudden fracture occurs in Stage 3. The fracture can be ductile, brittle, or mixed mode, depending upon the toughness of the material and the magnitude of the applied stress. During Stage 1, the peak bending stress is less than the yield strength of the material, and no gross yielding of the gear teeth occurs. However, local plastic deformation might occur in regions of stress concentrations or areas of structural discontinuities such as surface notches, grain boundaries, or inclusions. The cyclic, plastic deformation usually occurs on slip planes that coincide with the direction of maximum shear stress. The cyclic slip continues within the slip planes of a few grains, usually near the surface where the stress is highest, until micro cracks are initiated. The cracks grow in the planes of maximum shear stress until they form a major crack. The Stage 2 propagation phase begins when the crack turns and grows across grain boundaries (transgranular) in a direction approximately perpendicular to the maximum tensile stress. During the propagation phase, the plastic deformation is confined to a small zone at the leading edge of the crack, and the surfaces of the fatigue crack usually appear smooth without signs of gross plastic deformation.

If the crack propagates intermittently, it might leave a pattern of macroscopically visible beach marks. These marks correspond to positions of the crack front where the crack stopped. The origin of the fatigue crack is usually on the concave side of curved beach marks and the crack is often surrounded by several concentric beach marks. Beach marks might not be present, especially if the fatigue crack grows without interruption under cyclic loads that do not vary in

magnitude. The presence of beach marks is a strong indication that the crack was due to fatigue, but it is not absolute proof because other failure modes might also leave beach marks (for example, stress corrosion under changing environment).

If there are multiple crack origins, each producing separate crack propagation zones, ratchet marks might be formed. They are caused when adjacent cracks, propagating on different crystallographic planes, join together to form a step. Ratchet marks are often present on fatigue cracked surfaces of gear teeth because the stress concentration in the root fillet frequently initiates multiple fatigue cracks.

Low-Cycle Fatigue

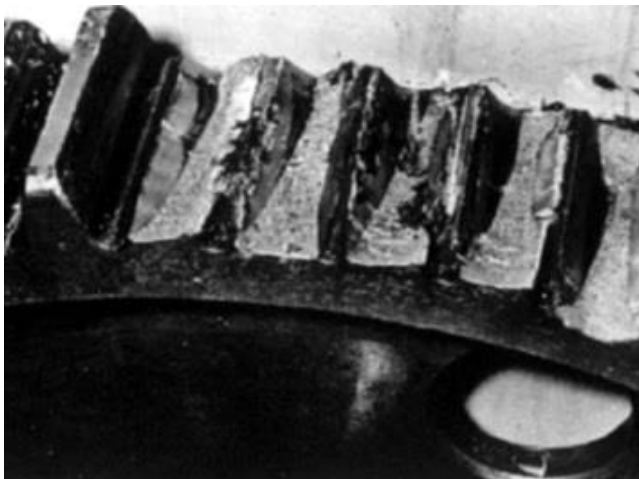
Low-cycle fatigue is defined as fatigue where macroscopic plastic strain occurs in every cycle and the number of cycles to failure is low (typically about 1000 to 10,000). Cracks might initiate within the gear teeth, as well as on the surface, and a smaller fraction of the life is spent initiating rather than propagating cracks.

High-Cycle Fatigue

High-cycle fatigue is defined as fatigue where the cyclic stress is below the yield strength of the material and the number of cycles to failure is high. Most gear tooth bending failures are due to high-cycle fatigue rather than low-cycle fatigue. A large fraction of the life is spent initiating rather than propagating cracks.

Depending on the place of initiation, fatigue cracks are called root fillet cracks, profile cracks, or tooth end cracks.

Root Fillet Cracks



Root Fillet Cracks on Several Teeth

Although bending fatigue cracks might occur elsewhere, they usually initiate in the root fillet on the tensile side of the gear tooth. Above figure shows an example of fatigue of several teeth that were loaded on both flanks.

Profile Cracks

Fatigue cracks can initiate on the active surface of the gear tooth if there are stress concentrations caused by macropits or material flaws.

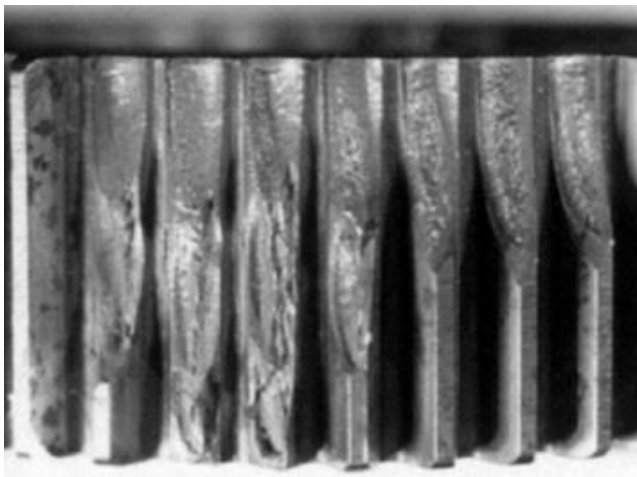


Profile Cracks Originating from Severe Pitting

Above figure illustrates an example of profile cracks originating from severe pitting.

Tooth End Cracks

Fatigue cracks can initiate at an end of the gear tooth if the load is concentrated at the tooth end. Stress concentrations or material flaws at the ends of the teeth can also be responsible for tooth end cracks.



Broken Tooth Ends

An example of broken tooth ends is shown in above figure.

Note

A broken tooth tends to be catastrophic to a gear drive, so the designer usually makes the teeth large enough so that they are definitely less apt to fail in breakage mode than in a pitting mode. This makes the design life of a gear drive primarily dependent on its surface fatigue capacity (pitting resistance) rather than on its cantilever beam capacity (capacity to resist tooth breakage).

For more information, please see ANSI/AGMA 1010-F14, Appearance of Gear Teeth - Terminology of Wear and Failure published by American Gear Manufacturers Association, 1001 N. Fairfax Street, Suite 500 Alexandria, VA 22314 | website: www.agma.org

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