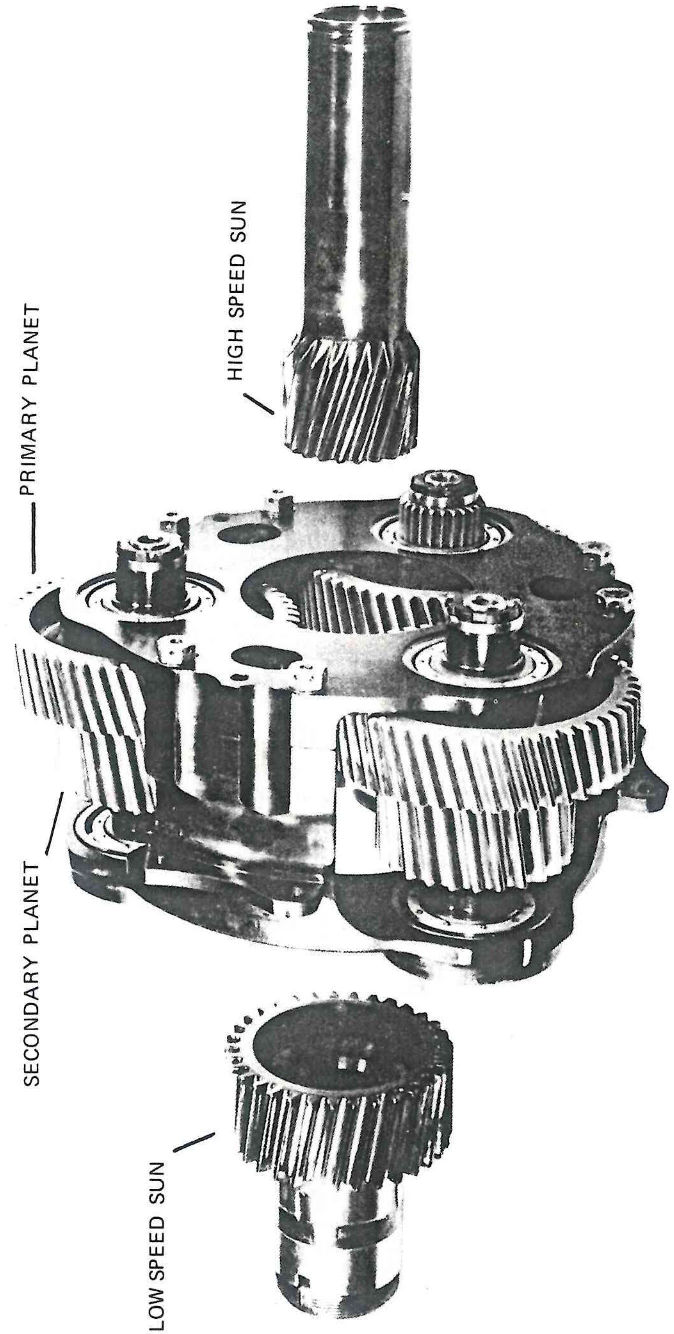

Figure 9.6 Reverted compound planetary. (Courtesy of AVCO, Lycoming Division, Stratford, Conn.)

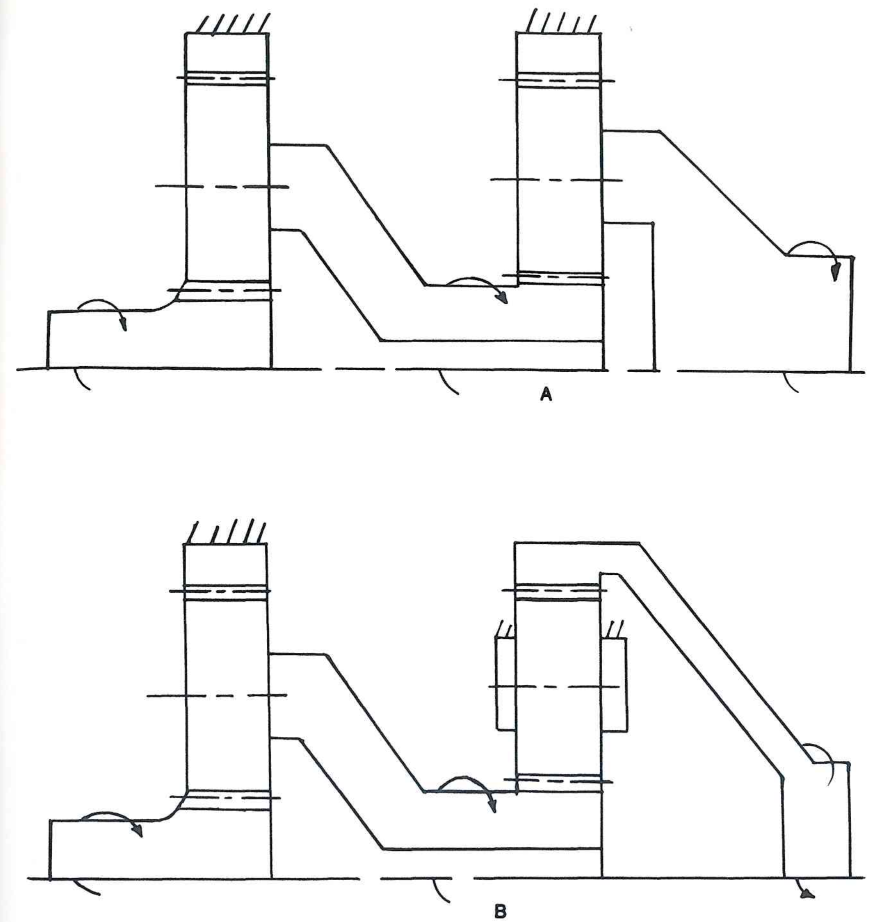

Figure 9.7 Two-stage planetary gearing.

The input and output rotation are in the same direction. Figure 9.7B shows a ring gear output from the second stage and in this case the second-stage reduction ratio is

$$R_2 = \frac{W_{s2}}{W_{r2}} = \frac{R_{r2}}{R_{s2}}$$

and the total reduction ratio is

$$R = R_1 R_2 = \left(1 + \frac{R_{r1}}{R_{s1}}\right) \frac{R_{r2}}{R_{s2}}$$

The input and output directions of rotation are opposite in sense.

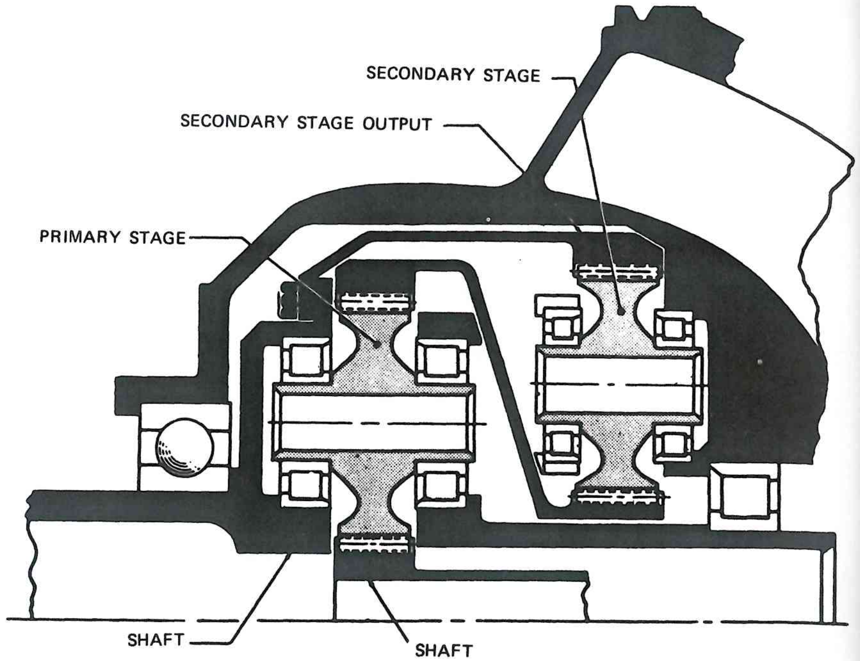

Figure 9.8 Split power transmission.

Split Power Transmissions

An interesting planetary arrangement which results in a very compact design is the split power transmission. Figure 9.8 illustrates one version which is most useful in a ratio range of approximately 15:1. As shown in the figure the primary planetary stage is driven by the sun gear and has both a rotating carrier and ring gear. The primary ring gear drives the sun gear of the second stage which through the stationary second-stage planets drives the second-stage ring gear. Both the second-stage ring gear and the primary stage carrier are connected to the output shaft, thus enabling the power split. Part of the input power is transmitted directly to the output shaft through the primary stage carrier, thereby bypassing the second stage of gearing. Following are the speed and power equations describing the split power transmission:

$$W_{s1}R_{s1} = W_{p1}R_{p1} + W_{c1}R_{s1}$$

$$W_{r1}R_{r1} = W_{p1}R_{r1} - W_{c1}R_{r1}$$

$$W_{r1} = W_{s2}$$

$$W_{s2}R_{s2} = W_{p2}R_{p2} = W_{r2}R_{r2}$$

$$W_{r2} = W_{c1}$$

where the subscripts $_{s1}$, $_{c1}$, $_{p1}$, and $_{r1}$ refer to the primary-stage sun, carrier, planet, and ring and the subscripts $_{s2}$, $_{p2}$, and $_{r2}$ refer to the second-stage sun, planet, and ring gears.

Combining the equations above we arrive at the expression for the speed ratio:

$$R = \frac{W_{s2}}{W_{c1}} = \frac{W_{s1}}{W_{r2}} = 1 + \left(\frac{R_{r1}}{R_{s1}}\right) + \left(\frac{R_{r2}}{R_{s2}} \quad \frac{R_{r1}}{R_{s1}}\right)$$

To determine the horsepower split between the primary and secondary stage, the following expressions are derived:

$$HP_{in} = \frac{W_T R_{s1} n_{s1}}{63{,}025}$$

where n_{s1} is the primary sun gear rpm.

$$\text{HP primary carrier} = 2W_T \frac{(R_{s1} + R_{r1})n_c}{2(63{,}025)}$$

where n_c is the carrier or output rpm. The percentage of power going out the carrier is

$$\frac{\text{HP primary carrier}}{HP_{in}} = \frac{(R_{s1} + R_{r1})n_c}{R_{s1}N_{s1}} = \frac{1}{R}\left(1 + \frac{R_{r1}}{R_{s1}}\right)$$

The exact percentage of total power transmitted directly by the primary carrier is dependent on the primary- and secondary-stage ratio split but is approximately one-third of the total. The remaining two-thirds of the power is transmitted through the secondary stage. The power split-up serves to reduce secondary-stage loading and increases gear efficiency. The disadvantage of the split power system is that it is mechanically complex.

Figure 9.9 shows a different split power variation which is used to achieve ratios in the range 24:1 to 55:1. The application here is a wheelmotor which combines a hydraulic motor and the planetary gearing as a single unit in the wheel hub. Wheelmotors are used in mobile construction and off-the-road equipment where a centrally located pump drives the hydraulic motors at each wheel.

Figure 9.10 illustrates a wheelmotor casing. In the arrangement of Figure 9.9 the hydraulic motor drives through a sun gear. The output members are the primary- and secondary-stage ring gears which are connected to the wheel rim. The power split up is as follows: Part of the input goes directly from the primary

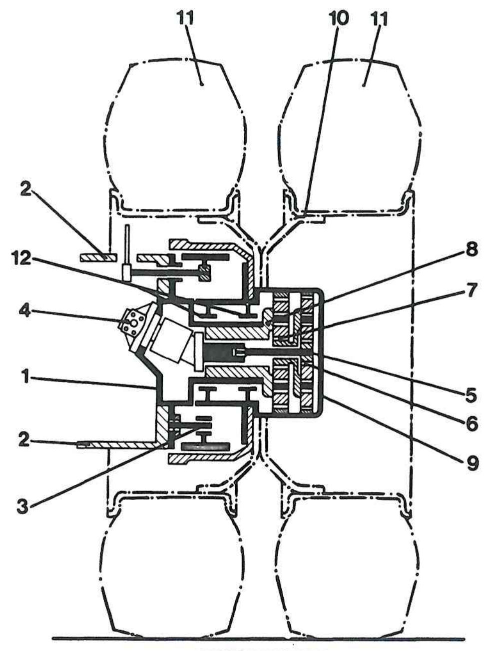

Figure 9.9 Wheelmotor transmission. Key: 1, casing; 2, vehicle chassis; 3, road brake; 4, hydraulic motor; 5, drive shaft; 6, first planetary stage; 7, second planetary stage; 8, axle journal; 9, hub; 10, rim; 11, twin tires; 12, roller bearing. (Courtesy of American Lohmann Corporation, Hillside, N.J.)

sun to the primary ring gear. The remainder goes directly from the primary carrier to the secondary sun gear and then to the secondary ring gear.

Power Feedback Systems

When working with complicated gear systems it is possible to arrange the gears in such a manner that the power transmitted by some components is greater than the input power. Although these arrangements generally achieve unusually large ratios, the component size must be sufficient to handle the recirculating power and this disadvantage may offset the large ratio obtained.

If it is not recognized that a system has recirculating power, failures may occur. Figure 9.11 illustrates a feedback system. The input is to the primary-stage sun gear, which is connected to the secondary-stage sun gear. The output is the secondary-stage carrier. Following are the speed equations for this system:

Figure 9.10 Wheelmotor casing. (Courtesy of American Lohmann Corporation, Hillside, N.J.)

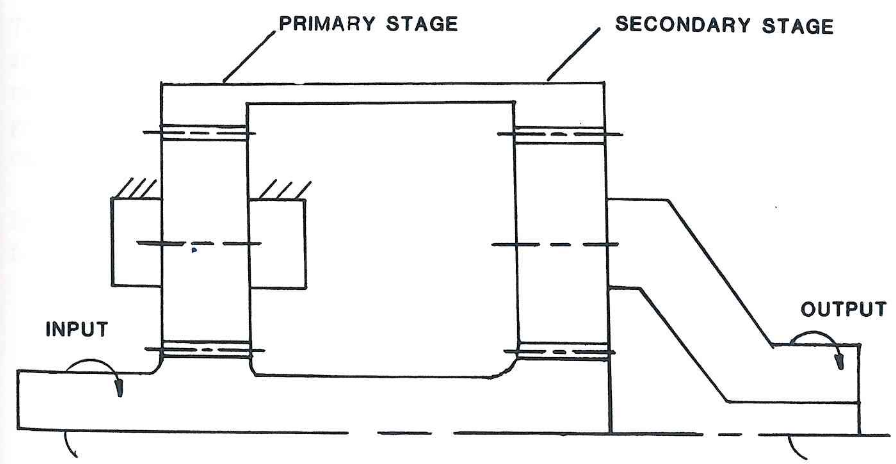

Figure 9.11 Feedback system.

$$W_{s1}R_{s1} = W_{r1}R_{r1} = W_{p1}R_{p1}$$
$$W_{r2}R_{r2} = W_{p2}R_{p2} - W_{c2}R_{r2}$$
$$W_{s2}R_{s2} = W_{p2}R_{p2} + W_{c2}R_{s2}$$
$$W_{s2} = W_{s1}$$
$$W_{r2} = W_{r1}$$

Combining the equations above the speed ratio is

$$R = \frac{W_{s1}}{W_{c2}} = \frac{1 + R_{r2}/R_{s2}}{1 - \dfrac{R_{s1}R_{r2}}{R_{r1}R_{s2}}}$$

Let us assume the following dimensions:

$$R_{s1} = 1 \text{ in.}$$
$$R_{s2} = 1.05 \text{ in.}$$
$$R_{r1} = 8.3 \text{ in.}$$
$$R_{r2} = 8.25 \text{ in.}$$

The reduction ratio $R = 166{:}1$. Let us determine the horsepower transmitted:

$$\text{hp out} = 2W_{TS_2}\,\frac{R_{s2} + R_{r2}}{2}\,\frac{n_{c2}}{63{,}025}$$

where

W_{TS_2} = secondary-stage sun tangential load, lb
n_{c2} = carrier rpm

$$\text{hp secondary sun} = \frac{W_{TS_2}R_{s2}N_{s2}}{63{,}025}$$

where N_{s2} is the secondary sun in rpm.

$$\frac{\text{hp secondary sun}}{\text{hp out}} = \left(\frac{R_{s2}}{R_{s2} + R_{r2}}\right) R = 18.74$$

The power circulating in the gear train is 18.74 times the output power.

Planetary Gear Design Considerations

Although the basic gear tooth design of planetary gear configurations is no different from parallel shaft gearing, there are several points that must be considered when rating planetary gears and defining the detail geometry. This section discusses load sharing, assembly, and choice of numbers of teeth and numbers of planets.

Load Sharing

The main advantage of planetary gearing, of course, is transmittal of load through two or more parallel paths. Ideally, the load should be shared equally by each planet gear, but due to manufacturing errors this is never the case. Some of the factors affecting load sharing are:

Inaccurate carrier bore location
Inaccurate gear tooth spacing
Variation in planet tooth thickness
Eccentricities between sun, ring, and carrier

Figure 9.12A shows the load distribution of a three-planet system. The situation is such that in order for the sun gear to be in static equilibrium the planets must share load equally. If the sun gear is free to move and one planet is overloaded, the sun gear will shift to equalize the planet loads. Figure 9.12B shows static equilibrium; however, the load is not equally shared between planets.

In general, there are three methods of attacking the problem of load sharing between planets:

1. A completely rigid system which relies on precise component tolerances
2. A system with flexibility built in by floating one or more of the members
3. Systems that rely on mechanical means to adjust the planets to provide load equalization

The most straightforward approach is a combination of 1 and 2, where tolerances are closely held and the sun or ring or both are flexible to adjust to load maldistributions. This technique is used in high-speed planetaries such as those in gas turbine engines. Strain gage studies have shown that load sharing within 10% can be practically achieved.

AGMA Standard 420.04, Practice for Enclosed Speed Reducers or Increasers Using Spur, Helical, Herringbone and Spiral Bevel Gears, states the following:

> To compensate for unequal loading of multiple planet pinions, the total power capacity of all pinions should be the calculated capacity of one pinion plus a maximum of 0.9 times the calculated capacity for each additional pinion. For two planet pinions multiply the calculated capacity of one pinion by 1.9; for three pinions multiply by 2.8. If a load balancing device is used which insures equal loading of all planets, the capacity modifying factors need not be used.

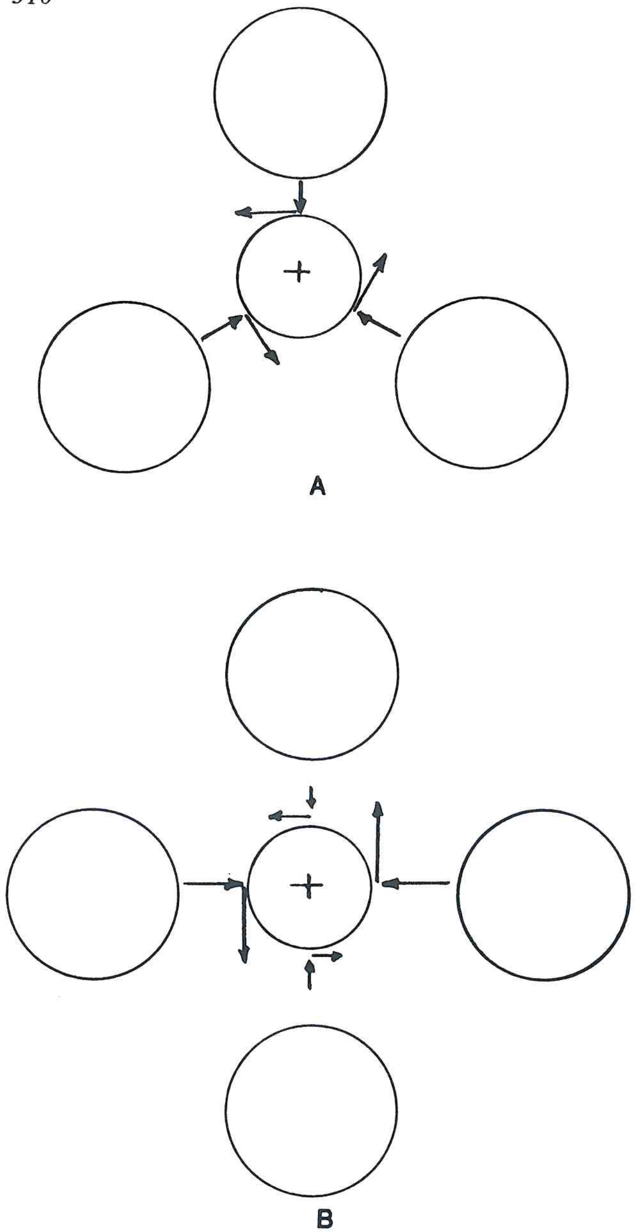

Figure 9.12 Planetary load distribution. A. Three planet system—load must be equally shared if sun is free to float. B. Four planet system in static equilibrium with unequal load distribution.

For any specific application the gear designer must determine what degree of load sharing is anticipated and rate the gearset accordingly. This estimate may be based on analysis, test, or experience, but it is important to understand that perfectly equal load sharing will not be achieved.

STOECKICHT DESIGN

A planetary configuration, extensively used in Europe for many years, is commonly called the Stoeckicht design, named after the inventor. Figure 9.13 illustrates this double helical configuration, which features flexibility not only in the sun gear but also in the ring gears, which are connected to the housing by a series of splines. The purpose of the splines is to achieve load equalization not only between the planets but also between each of the two helical gears. This system has been very successful in the past since the many degrees of freedom tend to compensate for tooth errors. There are disadvantages to the Stoeckicht design:

1. Use of through-hardened double helical gearing does not achieve as small an envelope as hardened and ground single helical gearing.
2. The design is mechanically complex.
3. The design does not lend itself to compounding to achieve higher ratios.

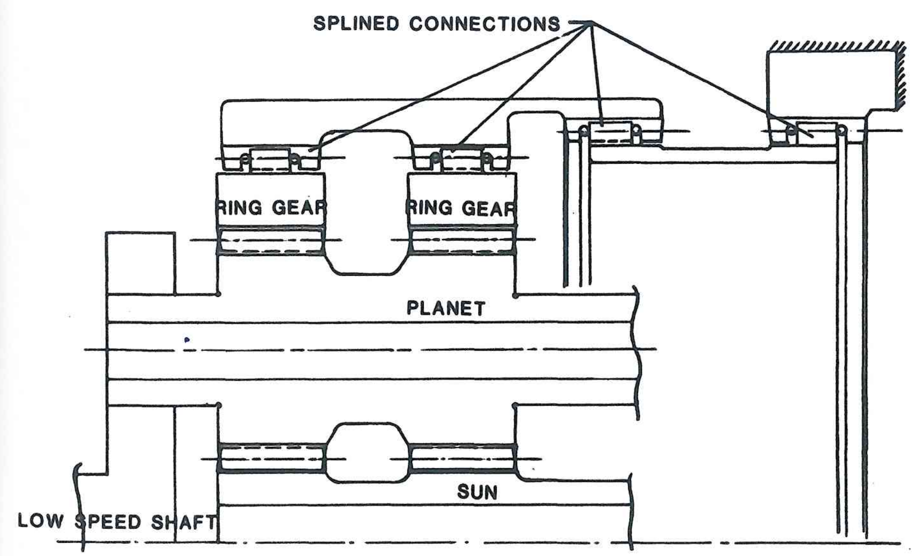

Figure 9.13 Stoeckicht planetary system design.

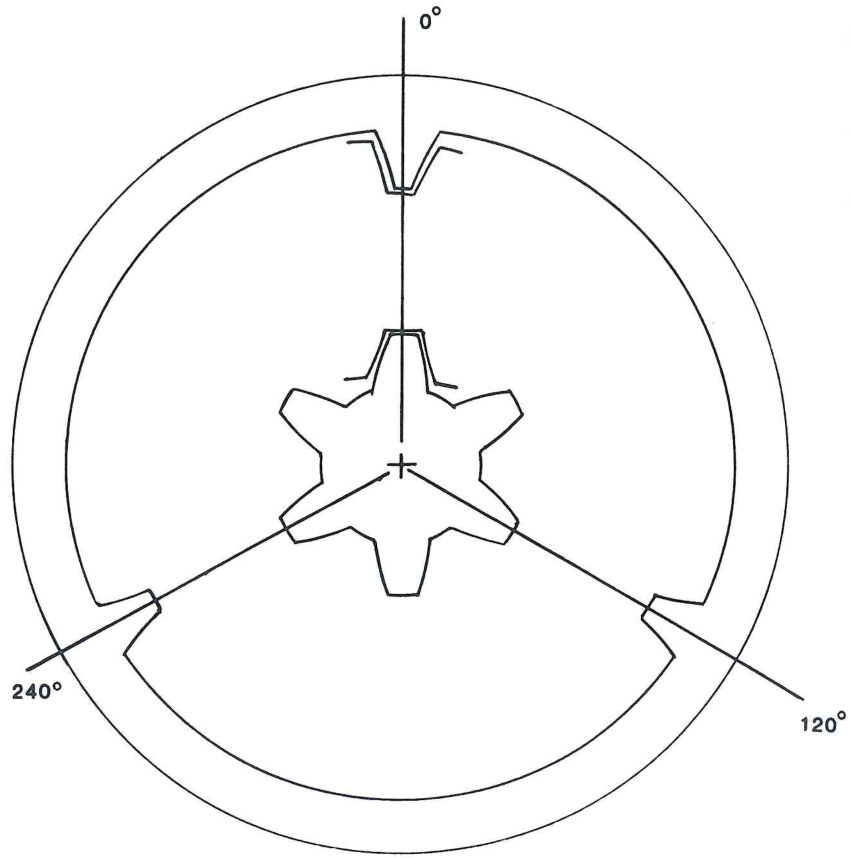

Figure 9.14 Simple planetary gear assembly.

Planetary Gear Assembly

When choosing the numbers of teeth in planetary gears, it must be understood that not every combination of gear teeth can be assembled. For instance, Figure 9.14 shows a three-planet set with a six-tooth sun gear and an 18-tooth ring gear. In this instance it is obvious that planet gears can be placed into mesh at the 0°, 120°, and 240° positions and there is no assembly problem. However, if the ring gear had 19 teeth the gearset would not be assembleable. For a simple planet set (sun, planet, ring) to be capable of being assembled, the following equation must be satisfied:

$$H = \frac{N_s + N_r}{K}$$

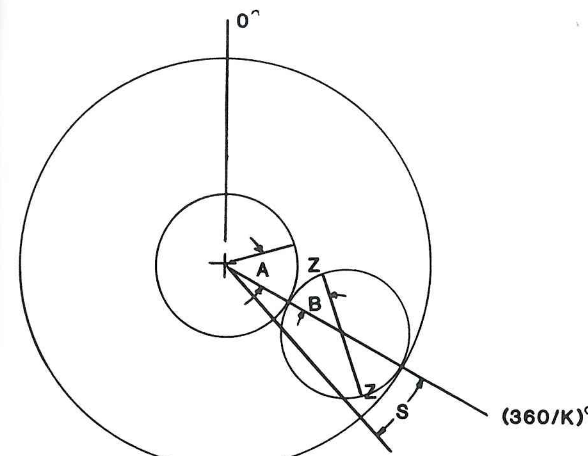

Figure 9.15 Simple planetary gear assembly analysis.

where

- H = a whole number (integer)
- N_s = number of sun gear teeth
- N_r = number of ring gear teeth
- K = number of planets

To understand this equation, refer to Figure 9.15. Assume a three-planet system with a planet gear located at the 0° position as shown. In order to assemble a planet at the 120° position, the centerline of a tooth or tooth space of the sun gear must arrive at the 120° point at the same time as the centerline of a tooth or tooth space of the ring gear. Let A be the angle between the center of that sun gear tooth or space and the 120° line and S be the angle between that ring gear tooth or space and the 120° mark. The following equation can be written for A:

$$A = \frac{360}{K} - \frac{360H_s}{N_s}$$

where

- K = number of planets
- H_s = whole number of sun gear teeth
- N_s = total number of sun gear teeth

Note that the equation is general and can be applied to systems with any number of planets.

A similar equation can be written for the ring gear angle S:

$$S = \left(\frac{360}{N_r} \quad H_r\right) - \frac{360}{K}$$

where

H_r = whole number of ring gear teeth
N_r = total number of ring gear teeth

Line ZZ on the planet (Figure 9.15) passes through either two tooth centers or tooth spaces or one tooth center and one tooth space, depending on whether the planet has an even or odd number of teeth. The planet rotates through an angle B at the same time the sun rotates through angle A and the ring gear through angle S. The relationship between these angles is

$$B = \frac{AN_s}{N_p} = \frac{SN_r}{N_p}$$

where N_p is the total number of planet teeth. Substituting for A and S yields

$$\left(\frac{360}{K} - \frac{360H_s}{N_s}\right)\frac{N_s}{N_p} = \left(\frac{360H_r}{N_r} - \frac{360}{K}\right)\frac{N_r}{N_p}$$

which simplifies to the assembly equation

$$H = \frac{N_s + N_r}{K}$$

The assembly equation for a compound planet system is more complicated and can be understood by considering Figure 9.16. Assume that points M and N are registry teeth on the compound planet. By this is meant that the centerline of a tooth on the primary planet is aligned with the centerline of a secondary planet tooth. The purpose of the registry marks is to enable all compound planets to be manufactured identically.

As was shown in the analysis of the simple planet system, the center of a sun gear space rotated an angle A from the 120° mark must arrive at the 120° mark at the same time as the center of a ring gear space, which is shown rotated an angle S from the 120° tooth mark.

$$A = \frac{360}{K} - \frac{360H_s}{N_s}$$

$$S = \frac{360H_r}{N_r} - \frac{360}{K}$$

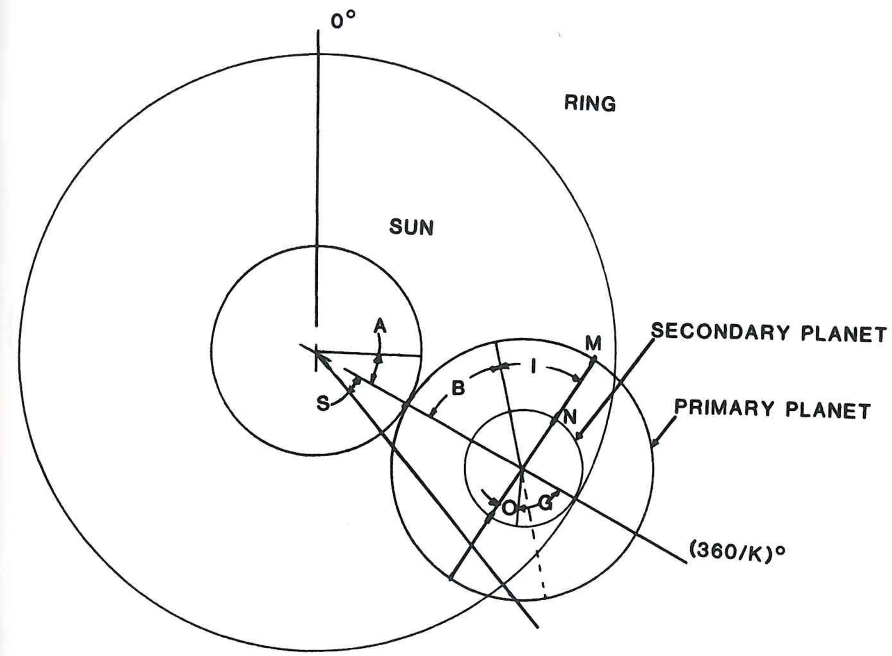

Figure 9.16 Compound planetary gear assembly analysis.

The primary planet rotates through an angle B at the same time the sun rotates through A:

$$B = \frac{AN_s}{N_{p1}}$$

where N_{p1} is the total number of primary planet teeth. The secondary planet rotates through an angle G at the same time the ring gear rotates through S.

$$S = \frac{GN_r}{N_{p2}}$$

where N_{p2} is the total number of secondary planet teeth.

The relationships between the registry teeth and the meshing teeth on the primary and secondary planets are:

$$I = \frac{360H_{p1}}{N_{p1}}$$

$$O = \frac{360N_{p2}}{N_{p2}}$$

where

H_{p1} = whole number of primary planet teeth
H_{p2} = whole number of secondary planet teeth

From Figure 9.16,

$$I = G + O - B$$

Substituting for I, G, O, and B, we have

$$\frac{360H_{p1}}{N_{p1}} = \frac{N_r}{N_{p2}}\left(\frac{360H_r}{N_r} - \frac{360}{K}\right) + \frac{360H_{p2}}{N_{p2}} - \frac{N_s}{N_{p1}}\left(\frac{360}{K} - \frac{360H_s}{N_s}\right)$$

and simplifying yields

$$\frac{N_r + N_sN_{p2}/N_{p1}}{K} + (H_{p1} - H_s)\frac{N_{p2}}{N_{p1}} = H_r + H_{p2}$$

In order to solve this equation the term on the left,

$$\frac{N_r + N_sN_{p2}/N_{p1}}{K}$$

is calculated and whole numbers are substituted for $(H_{p1} - H_s)$. If the left side of the equation works out to be a whole number, the gearset can be assembled.

In a compound planetary each of the planets must be identical not only for assembly purposes but also for proper load sharing. The registry teeth are used to satisfy this requirement and a typical allowable tolerance for the variation of the centerline of the primary planet tooth with the centerline of the secondary tooth is ±0.001 in. Figure 9.17 is a photograph of a compound planetary gear. Compound planets can be either of two-piece or one-piece construction. The processing must be very precise to achieve the registry dimension. In addition to the registry, the tooth thickness and stock removal must be closely controlled. The sequence of operations for a two-piece hardened and ground compound planet are as follows:

1. Cut primary and secondary planet teeth
2. Heat treat
3. Finish grind secondary planet
4. Press primary onto secondary using a fixture to align registry teeth
5. Grind primary planet to final tooth size and registry dimension

The reverted compound planetary gearset (Figure 9.6) with an input and output sun gear has the following assembly equation:

$$\frac{N_0 - N_{s1}N_{p2}/N_{p1}}{K} + (H_s + H_{p1})\frac{N_{p2}}{N_{p1}} = H_{p2} + H_0$$

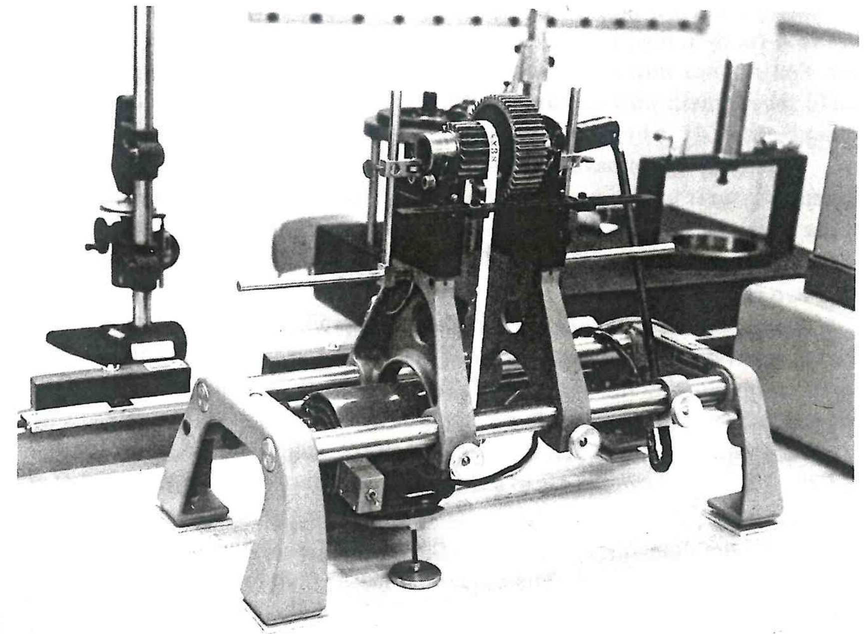

Figure 9.17 Compound planet being balanced.

where

N_0 = total number of low-speed sun gear teeth
H_0 = whole number of low-speed sun gear teeth

It should be noted that it may be physically possible to assemble a gearset that does not satisfy the assembly equations or has mistimed registry markings. If there is sufficient backlash it may be possible to force the components into mesh; however, the gears at some point in the rotation will tend to bind and will have poor load sharing.

Choice of Tooth Numbers

There are three considerations in choosing the numbers of teeth of a planetary in addition to the basic problem of achieving the correct ratio:

1. Assembly considerations as discussed in the preceding section
2. Choosing hunting teeth
3. Achieving sequential mesh rather than simultaneous mesh

By hunting teeth it is meant that the numbers of teeth in a gear mesh be such that each tooth in the pinion at some point in time mesh with each tooth in the gear. For instance with a 20-tooth sun and 40-tooth planet each sun gear tooth would always mesh with the same 2 planet teeth. By changing the number of planet teeth to 41, a hunting tooth mesh is achieved.

The advantage of having a hunting tooth is that the gearset has more of a tendency to wear in. For instance, if one tooth has a high spot and meshes with all the mating gear teeth, there is a better chance that the error will wear in and not cause permanent damage than if the discrepant tooth kept mating with the same two meshing gear teeth.

Sequential mesh occurs in a sun-planet mesh when the number of teeth in the sun is not divisible by the number of planets. When the number of teeth in the sun is divisible by the numbers of planets the mesh is called simultaneous. In this case each sun-planet mesh is occurring at the same point on the gear at the same time. For instance, when the planet at the 0° point is in mesh at its pitch line, the 120° and 240° planets in a three-planet system are also being loaded at the pitch line. This situation should be avoided, if possible, since the load impulse the simultaneous mesh gives to the mechanical system is stronger than that of the sequential mesh. The sequential mesh, therefore, is less likely to generate harmful vibrations.

The mesh frequency of a simple gearset is the number of teeth times the speed. If a planetary gear has sequential meshing, the mesh frequency will be the number of planets times the number of teeth times the speed. The higher sequential mesh frequency is less likely to excite any natural frequencies in the operating equipment.

Although having hunting teeth and sequential mesh is desirable, it is not always achieved and many gearsets operate successfully without fulfilling these conditions. The most obvious way of determining the number of planet teeth is to subtract the number of sun teeth from the number of ring teeth and divide by 2. Let us take an example:

N_s	= number of sun gear teeth	= 20
N_r	= number of ring gear teeth	= 100
CD	= center distance, inches	= 10

The planet gear geometry is

$$N_p = \frac{N_r - N_s}{2} = 40$$

$$DP = \frac{N_p + N_s}{2(CD)} = \frac{N_r - N_p}{2(CD)} = 3.0$$

where DP is the diametral pitch.

$$PD_p = \frac{N_p}{DP} = 13.3333 \text{ in.}$$

What if the number of ring gear teeth minus the number of sun gear teeth is not even? Let us assume the following case:

$$N_s = 20$$

$$N_r = 103$$

$$CD = 10$$

$$N_p = \frac{N_r - N_s}{2} = 41.5$$

If we make the number of planet teeth 41 there is no problem in achieving a practical gear set, but the operating conditions are different at the sun-planet mesh from the planet-ring mesh. At the sun-planet mesh:

$$DP_{s\text{-}p} = \frac{N_s + N_p}{2(CD)} = 3.05$$

The planet pitch diameter is

$$PD_{p(s\text{-}p)} = \frac{N_p}{DP_{s\text{-}p}} = 13.4426$$

At the planet-ring mesh:

$$DP_{p\text{-}r} = \frac{N_r - N_p}{2(CD)} = 3.10$$

$$PD_{p(p\text{-}r)} = \frac{N}{DP_{p\text{-}r}} = 13.2258$$

If the pressure angle ϕ is chosen at 22.5° at the sun-planet mesh, the pressure angle at the planet-ring mesh is calculated as follows:

$$\cos\phi_{s\text{-}p} \cdot PD_{p(s\text{-}p)} = \cos\phi_{p\text{-}r} \cdot PD_{p(p\text{-}r)}$$

and $\phi_{p\text{-}r} = 20.1117°$. The lead of the planet gear is

$$L = \frac{\pi \cdot PD}{\tan\psi}$$

If the helix angle ψ is chosen as 10° at the sun-planet mesh, the helix angle at the planet-ring mesh is as follows:

$$\tan \psi_{(p\text{-}r)} = \tan \psi_{(s\text{-}p)} \cdot \frac{PD_{p(p\text{-}r)}}{PD_{p(p\text{-}s)}}$$

and $\psi_{p\text{-}r} = 9.8419°$. The fact that the operating conditions at the sun-planet mesh are different from those at the planet-ring mesh is advantageous. The sun-planet mesh is critical in terms of compressive stress and flash temperature rise; therefore, it is desirable to have a large pressure angle which reduces both these parameters. Because the ring-planet mesh is internal, compressive stress and flash temperature rise are not usually a problem and the lower pressure angle can be tolerated. In fact, a lower pressure angle results in lower separating forces on the ring gear and this is desirable to reduce ring distortion and stress under load. Even in the case where the number of ring gear teeth minus the number of sun gear teeth is an even number, it can be advantageous to drop one or two teeth from the planet in order to achieve a larger pressure angle at the sun-planet mesh pitch diameter and a smaller pressure angle at the planet-ring mesh pitch diameter.

The question may be asked: How many planets should one use in a particular planetary set? There is a physical limit to the number of planets that can fit into any specific application. Figure 9.18 plots the maximum number of planet gears which can be assembled into a planet system versus the ratio of the sun gear diameter divided by the planet gear diameter. As the reduction ratio becomes smaller, the sun gear becomes larger and more planets can be fit in. As shown earlier, in a three-planet system with a floating sun gear, the load between the planets tends to be equalized. As the number of planets is increased, load sharing may suffer and the full benefit of the additional planets may not be realized. The vast majority of planetary gears use three planets; however, systems with 4, 5, 7, and more planets have been successfully developed.

A point that must be considered when planet gears are idlers meshing with both the sun and ring gear is that the planet teeth experience complete reversal of stress. Allowable bending stresses must be multiplied by 0.7 to compensate for this loading condition.

Planetary Bearing Loads

Quite often the critical components from a life point of view in a planetary system are the bearings. Because the planet gears are loaded both at the sun and ring gear meshing points (Figure 9.4), the planet bearings must react twice the tooth loads.

In an epicyclic system where the planet carrier rotates about the center of the system, the centrifugal force on the planet gears must be reacted by the planet bearings. The centrifugal force on a planet gear is calculated as follows:

$$F_c = mrW^2$$

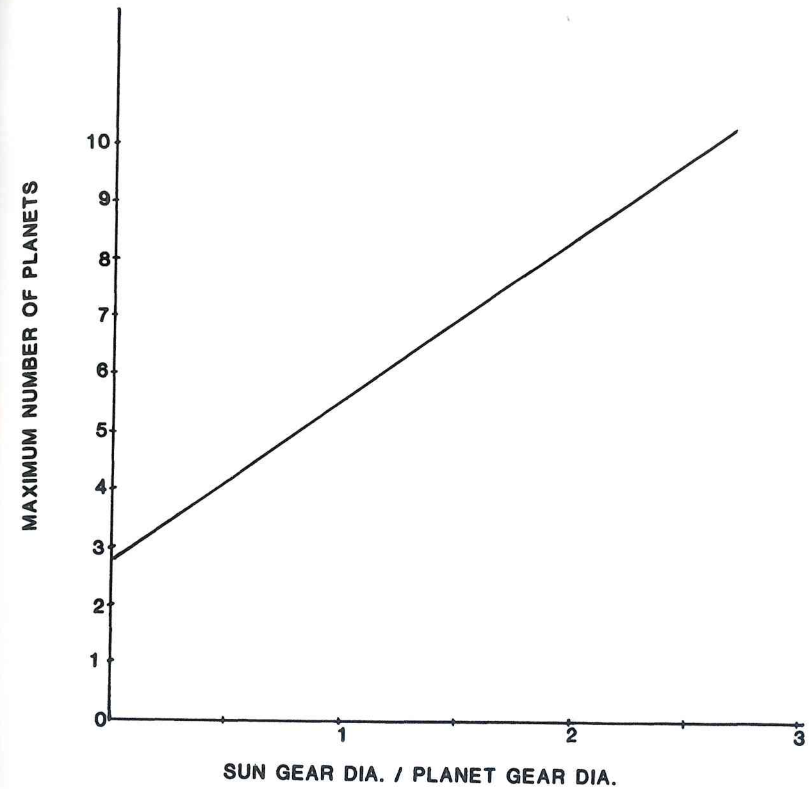

Figure 9.18 Maximum number of planets that can be assembled.

where

F_c = centrifugal force, lb
m = planet mass, lb-sec²/in.
r = radius to planet center, in.
W = carrier angular velocity, rad/sec

Let us work through an example of a helical epicyclic planetary gear set and calculate the planet bearing loads.

PD_s	=	sun gear pitch diameter, in.	= 3.5
PD_p	=	planet gear pitch diameter, in.	= 3.5
HP	=	transmitted horsepower	= 1000
n_s	=	sun rpm	= 7200

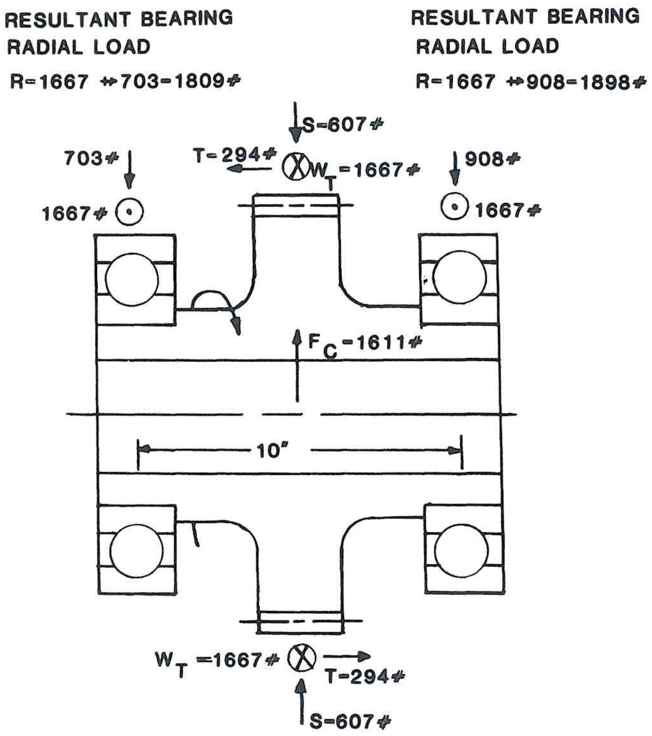

Figure 9.19 Planet bearing loading.

n_c	= carrier rpm	= 1800
ψ	= helix angle, deg	= 10
ϕ	= pressure angle, deg	= 20

Figure 9.19 illustrates the planet configuration and loading. Assume that the planet weighs 5 lb. The centrifugal force is

$$F_c = \frac{5}{386} \quad (3.5) \left(\frac{1800 \cdot 2\pi}{60}\right)^2 = 1611 \text{ lb}$$

The tangential load per planet with a three-planet system is

$$W_T = \frac{63{,}025(1000)}{7{,}200} \left(\frac{2}{3.5}\right) \frac{1}{3} = 1667 \text{ lb}$$

The separating load at each mesh is

$$S = W \tan \phi = 607 \text{ lb}$$

The thrust load at each mesh is

$$T = W_T \tan \psi = 294 \text{ lb}$$

For the planet to be in static equilibrium the bearing loads are as shown in Figure 9.19. It can be seen that the centrifugal load is significant. The limiting factor in an epicyclic system quite often is the carrier speed, which, if excessive, results in overloading of the planet bearings due to centrifugal force.

Planetary Gear Economics

At first glance it would appear that a planetary gearset must be more expensive than a parallel shaft configuration. There is the added complication of a carrier and two or more additional planet gears plus more bearings. Ring gears are usually expensive items.

In fact, the reduced size of planetary components offsets the cost of additional parts and the determination of which design is less costly is not so obvious. An important factor is the quantity to be manufactured and as the quantity of units increases, the savings in material and advantages of handling and machining smaller components begin to outweigh the disadvantages of mechanical complexity.

Also, when high ratios are required (over 15:1) planetaries tend to be more economical, even in relatively small quantities.

In very high horsepower units, the components in parallel shaft boxes tend to become so large that there may be an economic advantage to planetaries even in the ratio range 2:1 to 15:1.

In any case, the technical advantages of planetary gearing are sufficient such that this type of gearing should be considered, especially since the economic penalty may be small or nonexistent.

10

GEARBOX INSTALLATION: MOUNTING, ALIGNMENT, COUPLINGS

Proper installation of the geared system is essential to achieve good performance. The gearbox must be rigidly connected to the foundation, which must also be rigid and have a flat mounting surface. If the foundation or base plate structure is incorrectly designed or constructed vibration, shaft misalignment, bearing damage, and even shaft or housing breakage can result.

Most gearboxes are foot mounted. A flange on the bottom of the unit is doweled and bolted to a base. The doweling is important to ensure that the gearbox does not move during operation. This is a four-point mounting arrangement, and since three points define a plane it is difficult to install the unit such that all four points lie in the same plane. Two major reasons why the mounting points on the base plate are usually not coplanar are:

1. The steelwork warps as a result of poor welding, grouting, or concrete work.
2. The use of multiple steel beams which are not coplanar in the base plate.

When mounting the unit on multiple steel beams a base plate which extends under the entire gearbox and is at least as thick as the gearbox base should be used. Both the gearbox and the base plate should be rigidly bolted to the steel supports.

Shims can be used to bring all mounting surfaces into the same plane. Care must be taken that the shims form a solid tight pack when the bolts are tightened. Prior to final tightening, the shims should be inspected for rust, folds, wrinkles, burrs, tool marks, and dirt. Correct grouting is of great importance when bases are supported on concrete. Figure 10.1 is a checklist to ensure proper grouting.

Proper grouting checklist

1. Use non-shrink grout, as confirmed by test data per ASTM C-827. Volume change after hardening should be the measurement used.
2. Make sure baseplate design permits a complete fill, with adequate pour openings and inspection holes.
3. If epoxy grout is to be used, ask for evidence that long-term cold flow or "compression creep" will not occur.
4. Avoid metallic aggregate grouts where either severe temperature swings or corrosive surroundings may exist.
5. Keep the concrete foundation moist for 24 to 48 hours prior to grouting.
6. Be sure all oil, grease, or dirt is removed from any surface to be contacted by grout.
7. Place grout continuously, as quickly as possible, and work from one side or end across to the other side or end, to avoid trapping air in confined spaces.
8. Ensure that the surrounding temperature and the chemical additives allow ample working time for proper placement before setup (at least 45 minutes for a full foundation).
9. Allow enough curing time before placing the drive in operation; depending on the type of grout, this could be as long as 7 days.
10. When the drive is in service, watch for signs of deterioration of grout.

One result of ignoring the checklist above.

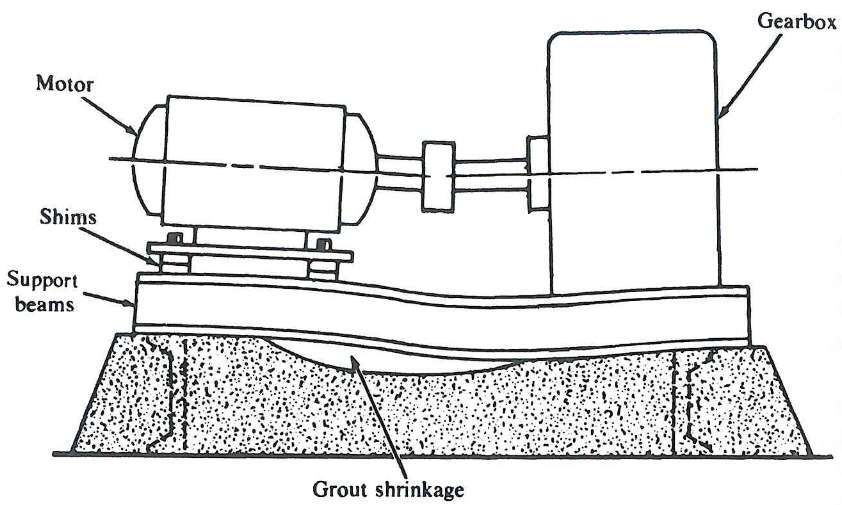

Figure 10.1 Grouting procedures. (From Ref. 1.)

Stability of the foundation is extremely important. Once installed, the base must not deform. One major cause of difficulty is thermal expansion, which can be due either to ambient conditions such as partial sunshine unevenly heating the structure or proximity to hot operating equipment. Of course, the structure must initially be designed with sufficient rigidity to withstand all operating forces without distortions.

It is possible to encounter a resonance condition where the natural frequency of the complete system assembly, including the base plate, coincides with an operating frequency. This type of structural resonance can be corrected in several ways:

1. The natural frequency of the base can be increased by making it more rigid by adding stiffeners or gussets.
2. The natural frequency of the base can be lowered by removing material from the base.
3. The mounting system can be made more elastic to dampen or lower the frequency. An example of this is the use of spring washers at the attachment points.
4. The mass of the rotating system can be changed to modify its natural frequencies.

When handling the unit at installation, care must be taken not to stress parts which are not meant to support the gearbox weight. Gearboxes should be lifted only by the means provided by the manufacturers, such as lifting holes in the casing.

COUPLINGS AND SYSTEM ALIGNMENT

To connect the driving and driven equipment to the gearbox, input and output shaft couplings are used. It would not be practical to align the centerlines of the equipment exactly; therefore, the coupling must have some degree of flexibility to accommodate misalignment. Even if it were possible to perfectly align the equipment at some given operating point, because load, speed, or ambient conditions vary, the alignment will change. Figure 10.2 illustrates the alignment conditions that the equipment shaft ends may be in. The coupling must accommodate these conditions while transmitting torque, yet limit the forces on machine components such as shafts and bearings that result from misalignment. If the coupling had no flexibility, the misalignment would have to be accommodated by bending of the shafting, which would apply loads to the bearings. The maximum amount of misalignment couplings can accommodate must be satisfied at assembly with alignment procedures that will be described below. It is important, however, to attempt to minimize the misalignment to the lowest practical values since coupling life is strongly dependent on how well the system is aligned.

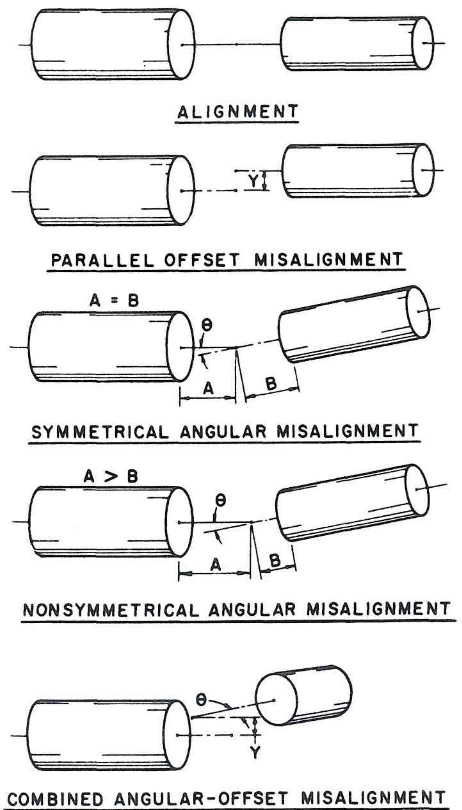

Figure 10.2 Shaft alignment conditions. (From Ref. 2, courtesy of American Gear Manufacturers Association, Arlington, Va.)

Figure 10.3 illustrates the most widely used coupling types. With gear couplings, misalignment is accommodated by sliding of the spline teeth. The sliding results in wear, and excessive wear is the major cause of failure of this type of coupling. In order to limit wear, gear couplings are lubricated, usually with grease. The maximum sliding velocity is [3]

$$V_s = \frac{D}{2} \theta \frac{2\pi \cdot \text{rpm}}{60}$$

where

V_s = sliding velocity, ips
θ = misalignment angle, in./in. (rad)
D = pitch diameter, in.

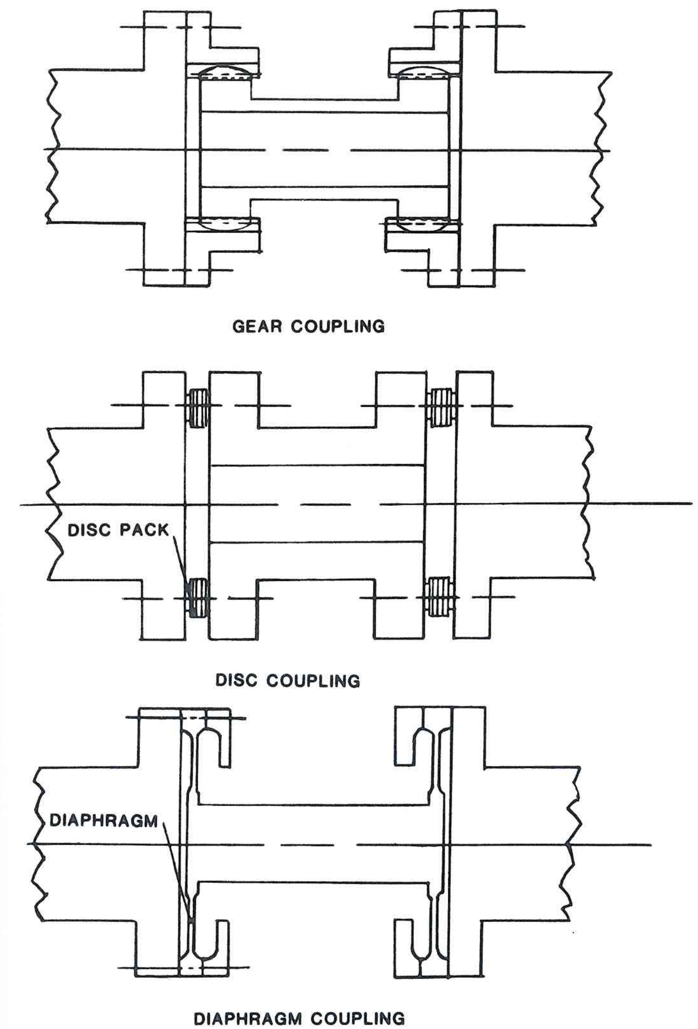

Figure 10.3 Coupling types.

Reference 3 gives the following guidance for allowable sliding velocities:

Less than 1.4 ips: optimum condition
3.0 ips: normal operation
5.0 ips: maximum allowed

Gear couplings can accommodate axial motion by allowing the male splines to float axially within the female splines as shown in Figure 10.3.

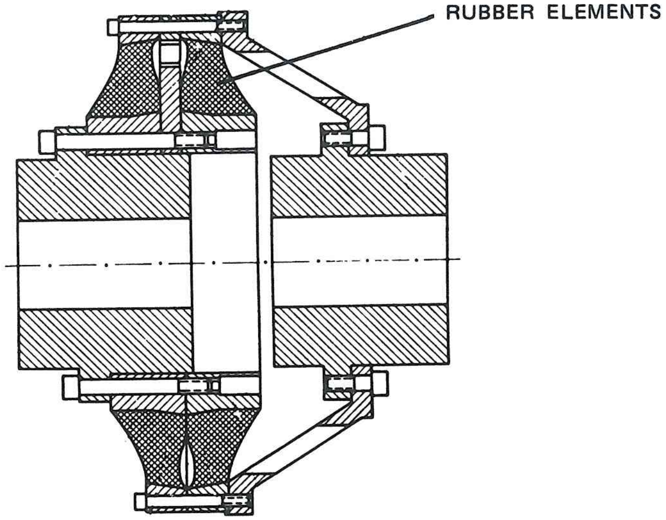

Figure 10.4 Elastic coupling. (Courtesy of American Lohmann Corporation, Hillside, N.J.)

Disk and diaphragm couplings accommodate misalignment and axial motion by flexing of the elastic elements. These couplings generally fail in fatigue and the fatigue life is dependent on misalignment, coupling speed, and the applied steady-state loads due to torque transference. Disk and diaphragm couplings require no lubrication.

Elastic couplings which incorporate rubber elements bonded to a steel backing are sometimes used with diesel drives to damp torsional vibrations. Figure 10.4 illustrates an elastic coupling with rubber elements which dampen vibration and accommodate misalignment.

The maximum amount of angular misalignment flexible couplings can accommodate is in the order of ¼°. The maximum allowable offset is dependent on the distance from pivot to pivot. For instance, if the distance from the center of one spline to the center of the other spline on the gear coupling shown in Figure 10.3 is 10.0 in. and ¼° or 0.0044 in./in. maximum angular misalignment is possible, the maximum inches of parallel offset are

$$0.0044(10.0) = 0.044 \text{ in.}$$

The values above are the maximum flexible couplings can accommodate; however, operation at these limits will shorten coupling life and may be detrimental to the system. During operation misalignment should not exceed

approximately one-fifth of the foregoing values and the system should be carefully aligned prior to startup to achieve this.

ALIGNMENT PROCEDURE

An outline of the steps required to align two pieces of equipment connected by a coupling is as follows:

1. Roughly set up the machines visually and with crude measurements.
2. Accurately align the shafts in the cold condition using precise measurements.
3. Operate the equipment and accurately measure the alignment in the hot condition.
4. Calculate the cold alignment required to achieve alignment in the hot condition. For instance, if it is determined that a motor shaft rises 0.005 in. with respect to the gearbox shaft during operation, the motor shaft should be positioned 0.005 in. below the gearbox shaft during cold alignment.
5. Realign in the calculated cold condition, operate the system, and recheck alignment in the hot condition.

The alignment procedure begins with the coupling hubs mounted on their respective shafts. The equipment is placed in position with the proper axial gap X, as shown on Figure 10.5. In order to roughly align the shafts, crude measurements can be made with a straightedge and calipers. The straightedge checked in

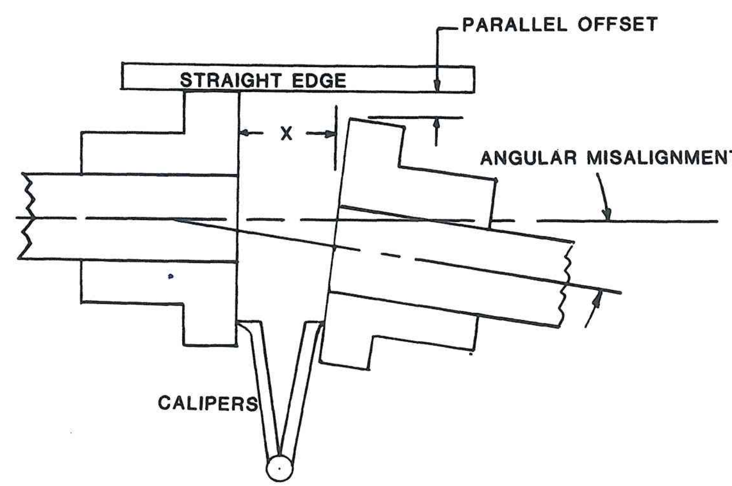

Figure 10.5 Rough alignment of coupling halves.

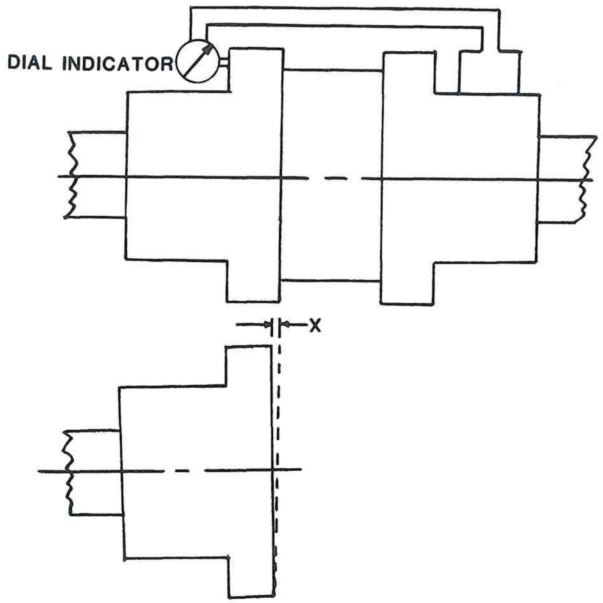

Figure 10.6 Checking for angular misalignment.

two planes will give an indication of the parallel offset. Caliper measurements at four points will give an indication of the angular misalignment.

There are several methods available to accurately align the shaft ends. One method, using a dial indicator, will be described. The first step, as shown in Figure 10.6 is to check for angular misalignment. A dial indicator base is mounted securely on the right-hand hub and the dial indicator stem is placed against a face on the left-hand hub. The connected shafts are rotated several times and the dial indicator checked to ensure ample movement in either direction. It is useful to use a mirror to observe the dial indicator gage as the shaft is rotated. The point at which a minimum reading is registered is found and at this point the dial indicator gage is set at zero. When the shaft is rotated 180° from this point the dial indicator reading will be the total angular misalignment. As an example, refer to Figure 10.6. If the minimum reading occurred at the bottom position, when the coupling is rotated one-half turn and the dial indicator is at the top position as pictured, the indicator will read the dimension X, which means that the distance between the coupling faces is X inches greater at the top than at the bottom. In order to align the faces, the equipment must be shimmed to narrow the distance between the faces at the top to X/2 inches. The distance between the faces at the bottom will increase X/2 inches and the faces will be square.

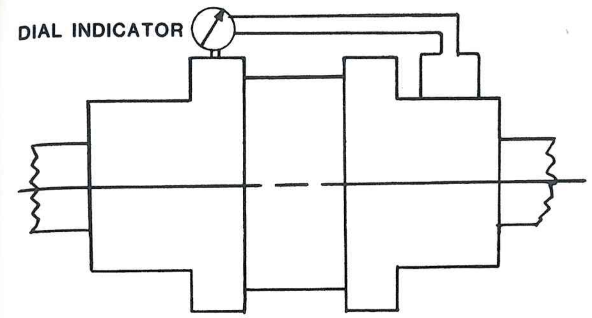

Figure 10.7 Checking for parallel offset.

Now that the angular misalignment is corrected, parallel offset must be measured. Figure 10.7 shows the dial indicator base securely fastened to the right-hand coupling hub with the stem in contact with a smooth outside diameter on the left-hand hub. Again the shafts are rotated several turns, making sure the indicator has travel in both directions and the point where the reading is minimum is found. The dial indicator is set at zero at this point and the shaft rotated 180°. In this position the indicator will read twice the amount of parallel offset. For instance, in Figure 10.7 if the minimum indicator reading was at the bottom position, when the coupling is rotated one-half turn and the indicator stem is at the top position, as pictured, if the reading is 0.050 in. the parallel offset of the center lines of the shaft ends is 0.025 in. with the left-hand shaft higher than the right-hand shaft. The equipment can now be shimmed vertically by 0.025 in. to bring the coupling into alignment.

After correcting for parallel offset the axial spacing and angular measurement should be rechecked to make sure that they were not disturbed. If the bracket holding the dial indicator is not rigid and allows the indicator to sag, an error will be introduced into the alignment readings. Readings in the horizontal plane will be little affected, but in the vertical plane, sag will increase the reading when the indicator is on top and decrease the reading when the indicator is on the bottom. One way to measure if the dial indicator is sagging is to mount the breacket on an accurately machined cylinder. The top reading, T (Figure 10.8), is greater than the nominal reading by the amount of sag and the bottom reading, B, is less than the nominal reading by the amount of sag. The sag is

$$\text{Sag} = \frac{T}{B}$$

The parallel offset determined by the method shown in Figure 10.7 must be corrected for the sag. This can be done by lowering the centerline of the measured shaft by the amount of sag. For instance, if the centerline of the left-hand shaft is measured to be 0.025 in. higher than the right-hand shaft and the

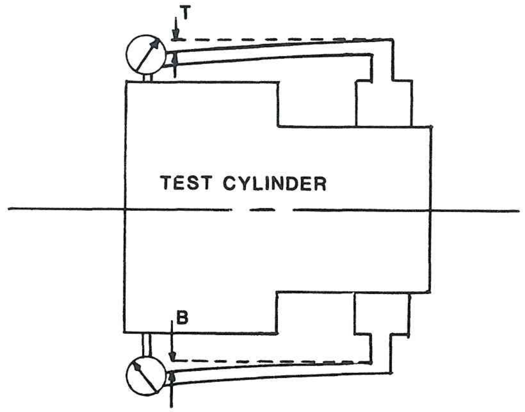

Figure 10.8 Measuring dial indicator bracket sag.

sag is measured to be 0.005 in., the actual parallel offset in the vertical plane is 0.020 in.

In Figures 10.6 and 10.7 both shafts are rotated when making the angular misalignment and parallel offset measurements. It would be possible to rotate only the shaft holding the dial indicator bracket and traverse the face and outside diameter of the other shaft while it is stationary. In this case any outside diameter eccentricity or face runout will be a source of error in the alignment reading. This problem is averted by rotating the measured surface together with the indicator.

Dial indicator methods of measuring misalignment have the disadvantage of not being able to monitor machine condition while operating. There are optical and proximity probe systems which can be used to monitor alignment continuously when an application warrants the expense of such instrumentation [3].

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11
GEAR UNIT OPERATION: TESTING, STARTUP, CONDITION MONITORING

In this chapter three phases of gearbox operation are discussed: testing, initial field startup, and condition monitoring.

TESTING

During the procurement process the gearbox manufacturer and the user must agree on the type of testing the completed unit will be subjected to prior to acceptance. The test program can be as simple as turning the shafting by hand to verify free operation of the internal components or as complicated as full-scale operation of an instrumented gearbox, on the actual application, through a predetermined test schedule.

Several factors must be considered when determining the degree of test program complexity:

1. Cost of testing
2. Confidence in the gearbox design
3. Consequences of gearbox failure

Clearly, a proven gear design, operating in an environment where downtime does not incur a large cost penalty, would not warrant extensive testing. On the other hand, if a gearbox is a critical component in a complex system the cost of testing may be slight compared to the loss in case of failure. Where human safety considerations are involved, testing may be required to limit legal liability. New designs or extrapolations of existing designs warrant sufficient testing to verify the analytical and manufacturing procedures.

Test programs can be extremely valuable in identifying problem areas prior to field operation. Design, manufacturing, and assembly errors can be identified and corrected prior to a costly catastrophic failure. In order to gain the most information, the test plan and instrumentation scheme should be carefully designed and acceptance limits set prior to operation. Data should be taken over the range of speeds, loads, and operating environments anticipated in the application. To identify potential problem areas quickly, the test program should include overspeed and overload operation. Care must be taken not to go too far and develop failure modes that would not occur in the actual application, but overspeeds to 110% and overloads to 125% should be within the capability of the unit.

Some of the common problems that can be identified in initial testing are:

1. *Excessive gearbox heat generation.* This is most commonly caused by oil churning and can be corrected by improving scavenging, reducing oil flow or changing oil type.
2. *Improper gear pattern.* May be caused by gear tooth errors, bore misalignment, or deflections. Can be corrected by tooth modifications or possibly bearing bore relocation.
3. *Overheating of gears or bearings.* Usually caused by insufficient lubrication. Can be corrected by increasing oil flow, retargetting oil jets, or changing oil type.
4. *Excessive noise or vibration.* Caused by unbalance, tooth errors, assembly errors, or operating at critical frequencies.
5. *Oil leakage.* Caused by misassembly of static or dynamic seals or pressurization of gearbox cavity.

Spin Tests

In many cases a full-speed, light-load test is considered sufficient to qualify a gear unit as acceptable. A typical test program might be:

1. Operate the gearbox at maximum continuous speed until bearing and lubrication oil temperature has stabilized.
2. Increase the speed to 110% of maximum continuous speed and run for a minimum of 15 min.
3. Reduce speed to maximum continuous and run for 4 hr at a minimum.

The following measurements should be made during the acceptance test:

Oil inlet temperature
Scavenge oil temperature
Oil feed pressure
Oil flow
Shaft speed

MODEL ________ S/N ________

DATE ________ OPERATOR ________ ENGINEER ________

RUN NO.	TIME OF DAY	SPEED RPM	OIL TEMP., °F			OIL PRESSURE PSIG	OIL FLOW GPM	VIBRATION LEVEL
			1	2	3			

Figure 11.1 Typical test log sheet.

Other parameters that can be measured during gear testing are:

Vibration
Shaft excursion
Noise
Bearing temperatures

A detailed test log should be kept making entries of each measurement at regular intervals such as every 15 min. Figure 11.1 shows a typical test log.

After completion of the mechanical running test, the gear unit should be opened for a visual inspection. Tooth meshes should be inspected for surface damage and proper tooth contact. All bearings and journals should be inspected for signs of surface damage or overheating.

With high-speed gear drives it is not uncommon to conduct the full-speed acceptance test driving the gearbox through a low-speed shaft. This is done if a high-speed prime mover is not available. In this case, the gears are contacting on their normally unloaded faces since the gearbox is being driven backwards. Such a test can still be useful to determine proper operation of the lubrication system and correct alignment of the gear shafts. Also, any gross machining or assembly errors can be identified.

The full-speed "light" load test is widely used since full-load testing at a gear vendor's plant is costly and sometimes a prime mover of sufficient capacity is not available.

Load Testing

Relatively low power gearboxes (up to possibly 200 hp) are load tested with power absorption devices loading the output shaft or shafts. Power absorption

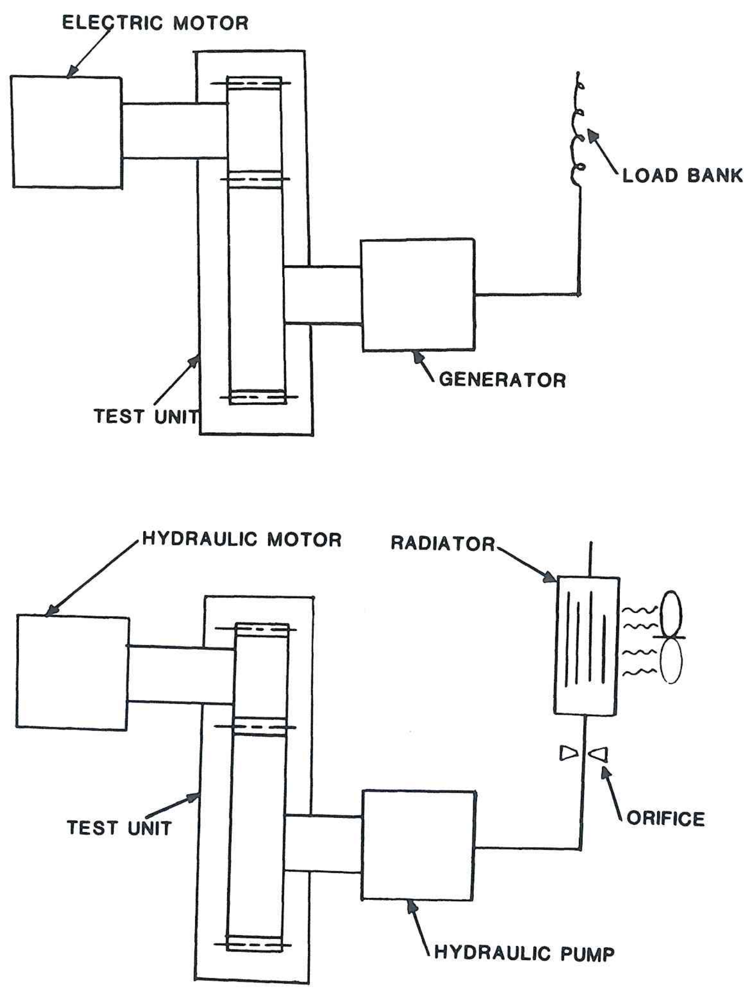


Figure 11.2 Power absorption test setups.

devices generally convert the power into heat and include water brakes, dynamometers, generators with load banks, hydraulic motors driving pumps, and so on. Prime movers include electric, hydraulic, or gasoline motors. Figure 11.2 illustrates some power absorption test setups.

For gear units transmitting high horsepower, power absorption testing is expensive and sometimes impractical. Quite often a suitable prime mover or power absorption device is not available to the gear manufacturer. Also, the power required to conduct tests becomes a major expense. To test large units

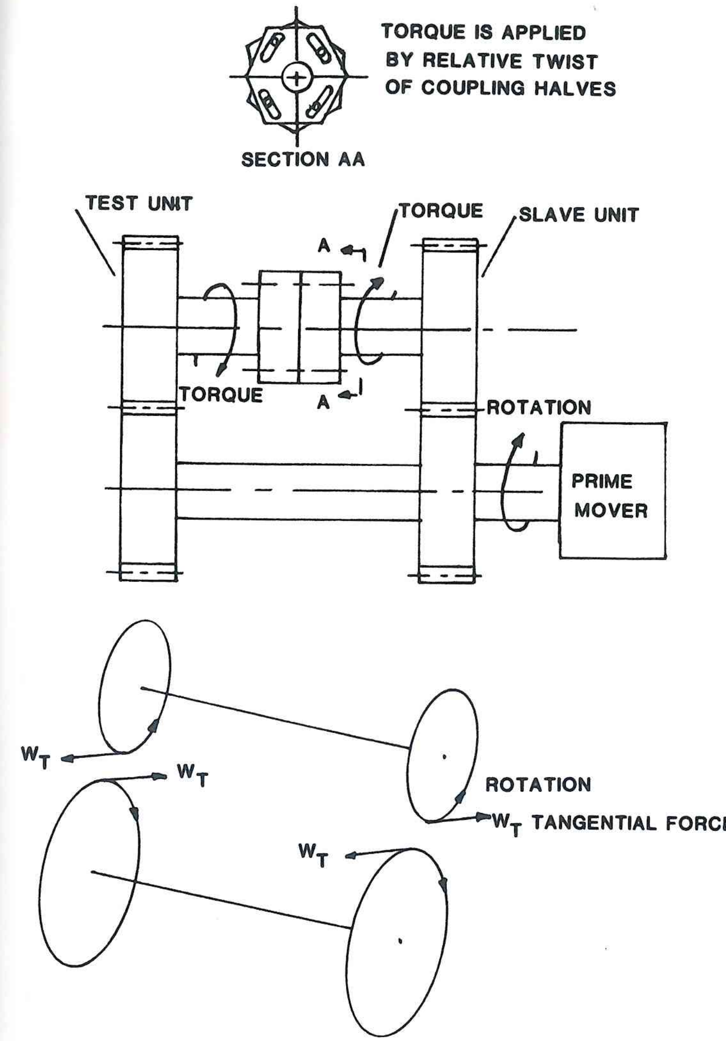


Figure 11.3 Simple regenerative rig.

at full speed and power, regenerative power techniques are used. Some other terms for this method of testing are recirculatory power, four-square rig, or back-to-back testing.

Figure 11.3 illustrates the simplest type of regenerative power test rig. Two identical gear sets are used. The high-speed shafts are coupled together, as are the low-speed shafts. Torque is applied to either the high- or low-speed shaft

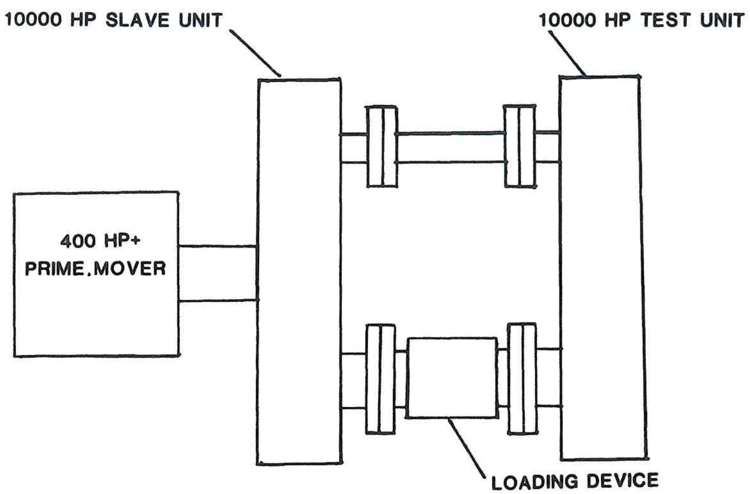


Figure 11.4 Back-to-back gear rig.

by twisting one coupling half flange with respect to the other and the torque is locked in by clamping the coupling halves together. The assembly is then rotated at the desired speed by the prime mover, which need only develop enough power to overcome the friction losses in the system.

To illustrate this point, Figure 11.4 shows two gear units set up in a back-to-back arrangement. If each unit transmits 10,000 hp and is 98% efficient the loss in the gearboxes will be 400 hp. Therefore, the prime mover for this 10,000-hp can be sized at 400 hp plus some margin for losses other than in the gear units.

With the arrangement shown in Figure 11.3, any gear tooth load desired can be developed. One gear pair is designated as the test set and the other the slave set. On the test set the torque and rotation are applied in the same sense as on the actual application. The slave set gear teeth, while loaded on the proper tooth faces, are rotating in the opposite sense. Another way to look at it is that in the test set if the pinion is driving the gear, the opposite is true in the slave set, with the gear driving the pinion. A review of the directions of rotation and torque shown in Figure 11.3 will illustrate this point.

When designing a back-to-back test thought must be given to the bearing and lubrication system operating conditions of the slave set since the opposite sense of rotation may require incorporation of some modifications. For instance, the oil feed groove in a journal bearing may have to be relocated.

The four-square type of rig gives reliable results concerning gear unit deflections under load and can be used to establish tooth modifications to

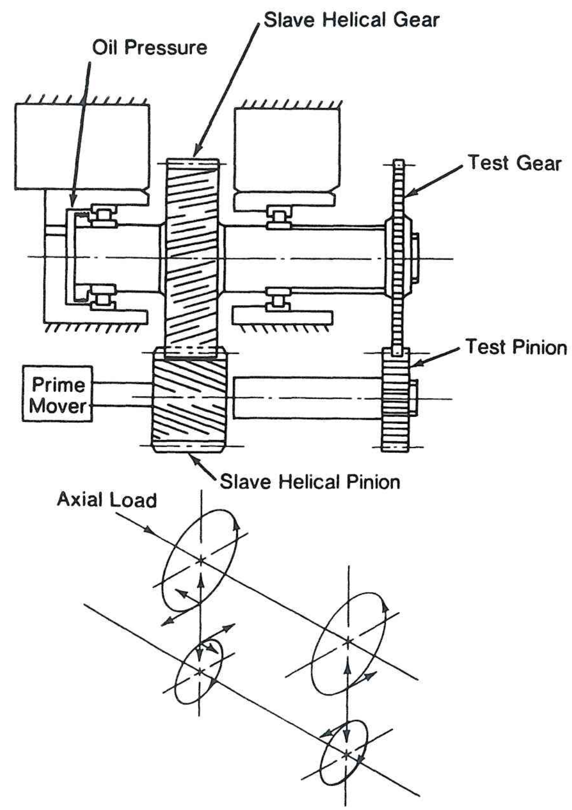


Figure 11.5 Helical gear loading regenerative rig.

improve load distribution. It is also useful in establishing efficiency values and exploring lubrication system problems. Long-term endurance testing can be accomplished at low cost.

The test setup shown in Figure 11.3 has some disadvantages:

1. The load is applied at zero speed when there is no oil film generated at the gear tooth and bearing interfaces. This may cause surface distress of the mating components. Also, it is impossible to simulate actual operating condition of most applications where load increases with speed. Starting torque requirements on the prime mover are high because full load is applied at zero speed.
2. As gears and bearings wear the locked-in load will gradually decrease.
3. The load may vary with operating temperature as components distort. This problem can be resolved by retorquing the coupling when rig temperature stabilizes.

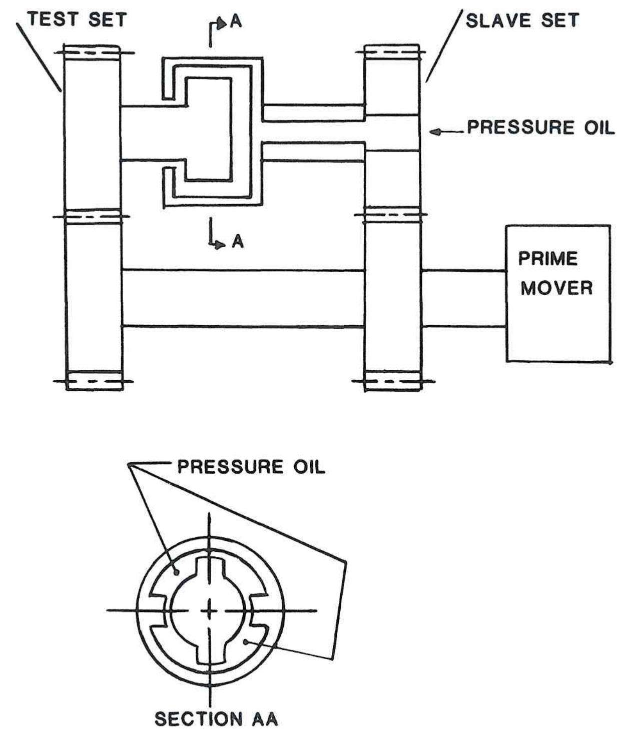


Figure 11.6 Hydraulic rotary torque actuation rig.

From these three points it can be seen that it would be advantageous to be able to apply and control the load while the rig is rotating and this can be accomplished by various means. One method shown in Figure 11.5 makes use of the fact that axial movement of a helical gear will cause rotation of the shafts. As the gear translates, all backlash is taken out of the system and torque is generated. The torque is proportional to the oil pressure applied at the load piston and therefore can be controlled at all speeds. Another method of loading a rig during operation is the rotary torque actuator shown in Figure 11.6. Pressure oil is fed to the actuator and the relative rotation of the vanes generates torque proportional to the oil pressure.

Parallel shaft units lend themselves to a four-square type of rig arrangement. Figure 11.7 shows a planetary gearbox rig where the high-speed shafts of the test and slave gearboxes are connected and operating concentrically inside the low-speed shafts. In order to apply load to the system, one of the stationary planet carriers is mounted on a loading fixture and turned to introduce a torque

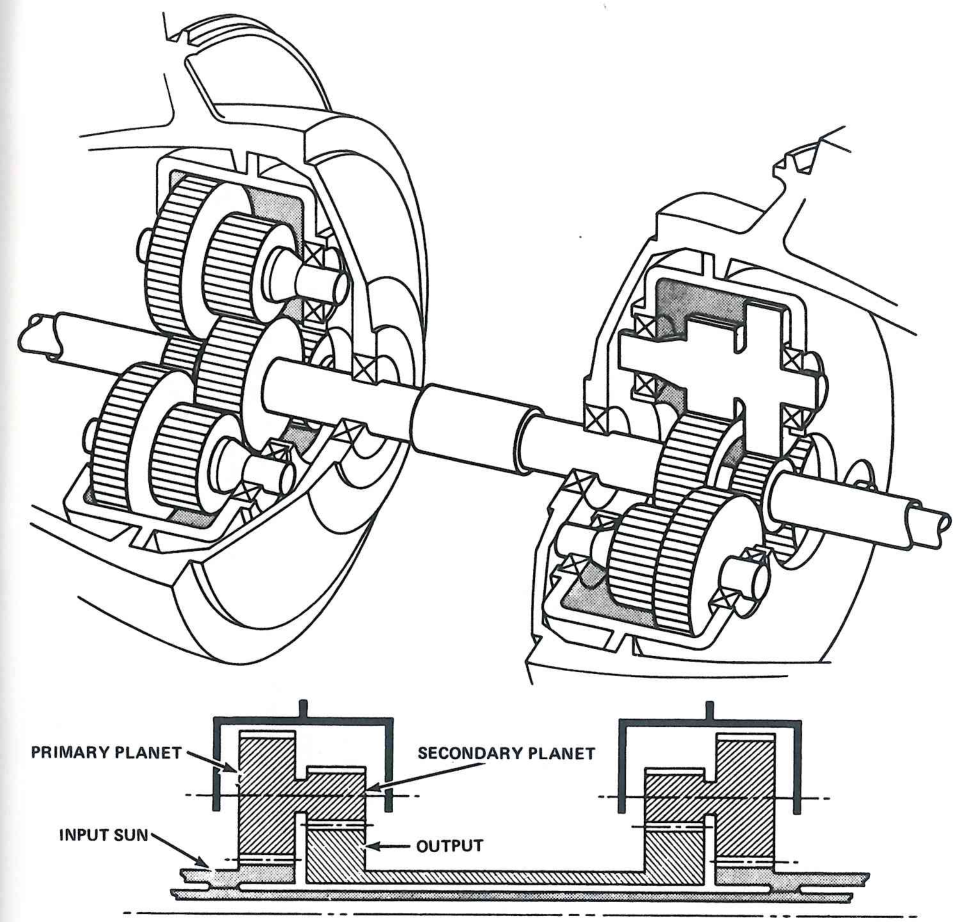


Figure 11.7 Planetary recirculating power rig.

into the system. Figure 11.8 illustrates such a test setup with pneumatic pistons supplying a couple to turn the carrier.

Another type of recirculatory power test setup is shown in Figure 11.9. In this case only one test gearbox is required. The electric motor, through the gearbox, drives a generator, which in turn generates electricity to drive the motor. Additional power is required to offset energy losses in the system. The same technique can be used with a hydraulic motor and pump.

Static Tests

It is possible to apply full load to a stationary gearbox and determine tooth patterns and housing rigidity, as shown in Figure 11.10. The tooth patterns will

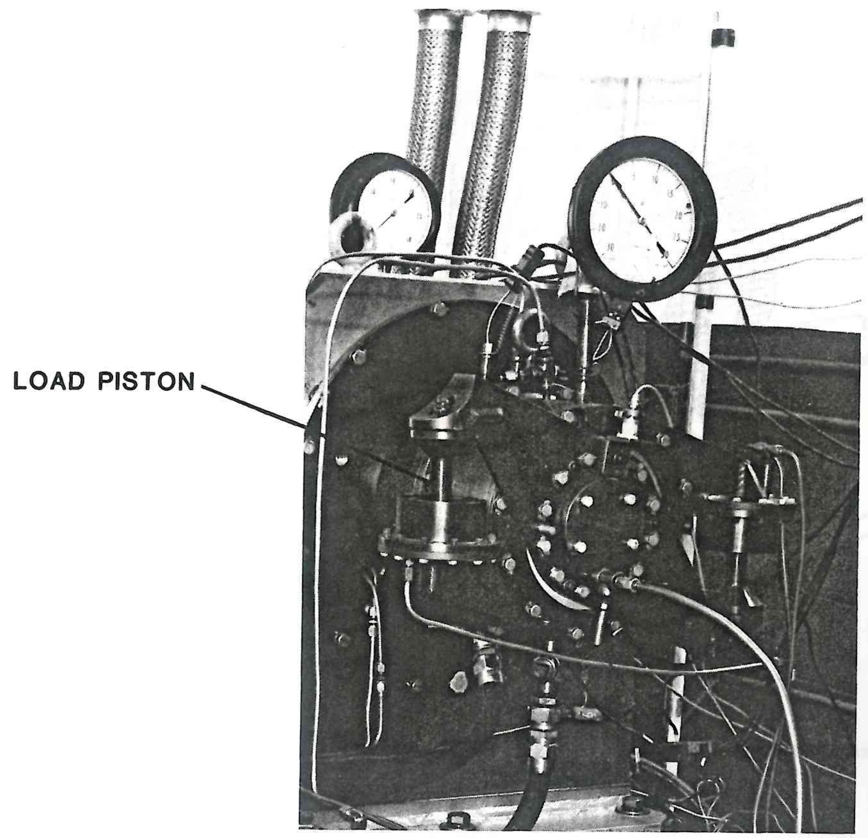


Figure 11.8 Planetary pneumatic load device.

show up if bluing or a paste compound is applied to one gear. If a paste compound is applied, care must be taken to minimize the thickness since paste thicknesses greater than 0.0002 in. will mask the existence of misalignment between the gear teeth.

If equipment is available to conduct a low-speed, full-torque rolling test, this type of operation can be specified. For instance, a 10,000-hp, 20,000-rpm shaft can be operated at 200 rpm with a 100-hp prime mover and develop the same torque, 31,512 in.-lb. This type of test is useful to demonstrate tooth contact, load-carrying capability, and housing rigidity, but yields no information concerning parameters related to speed, such as temperature, vibration, and noise.

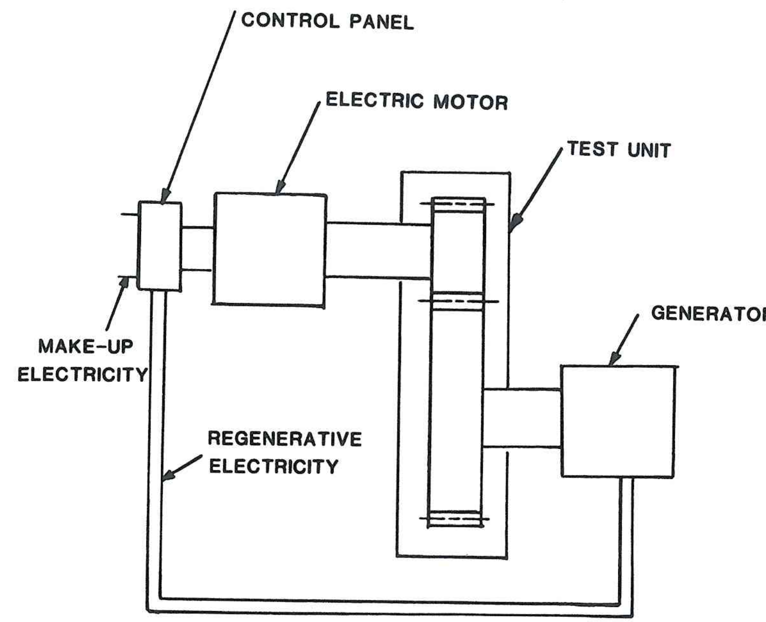


Figure 11.9 Regenerative electric motor drive rig.

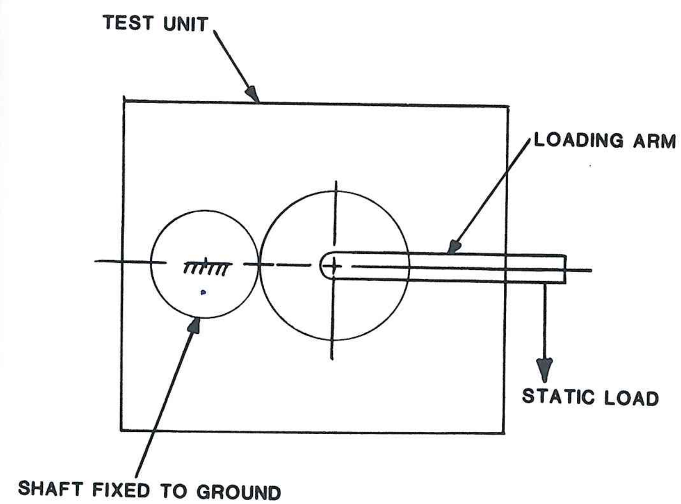


Figure 11.10 Static test rig.

Instrumentation

At a minimum during a gear test torque, speed, oil temperatures, and pressures should be monitored. Also important are vibration and oil flow.

Prior to a test an instrumentation plan should be formulated which includes the type of instrumentation, location, and accuracy. Typical accuracy tolerances of common measurements are as follows:

Speed	±0.5%
Torque	±2%
Power	±2%
Pressure	±2%
Temperature	±2%
Flow rate	±5%
Vibration frequency	±2%
Vibration amplitude (sine)	±10%

Instruments should be calibrated periodically and whenever possible an instrument reading should be verified by an independent measurement to determine if it is operating properly. For instance, if a flowmeter is used to measure oil flow, at some point a known volume of oil should be collected and the time recorded to check the flowmeter accuracy. Instrumentation failure is common and must be anticipated; therefore, redundant readings and backup systems must be planned for.

The most convenient way to measure speed is with a toothed wheel incorporated in the shaft system. An electromagnetic transducer is activated by the teeth passing by and the rpm are digitally displayed via an electronic counter. Care must be taken to set the proper gap between the tooth tips and the transducer. Also, runout of the gear can give erroneous readings. The counter gear usually has 60 teeth, but any number of teeth can be used if the electronics are set up to compensate. Other ways to measure speed are with a hand-held tachometer placed in contact with a shaft, and with strobe lights. When conducting a test, speed should always be measured in two different ways to ensure that a mistake is not made. At initial startup the correct direction of rotation should be verified.

Torque can be measured by monitoring shaft twist with strain gages. An indirect way of determining torque is by measuring the forces at the gearbox mounting pads (Figure 11.11). The housing torque is either the sum or difference of the input and output torques depending on whether the input and output shafts are rotating in the same or opposite directions. In Figure 11.11 assume the following:

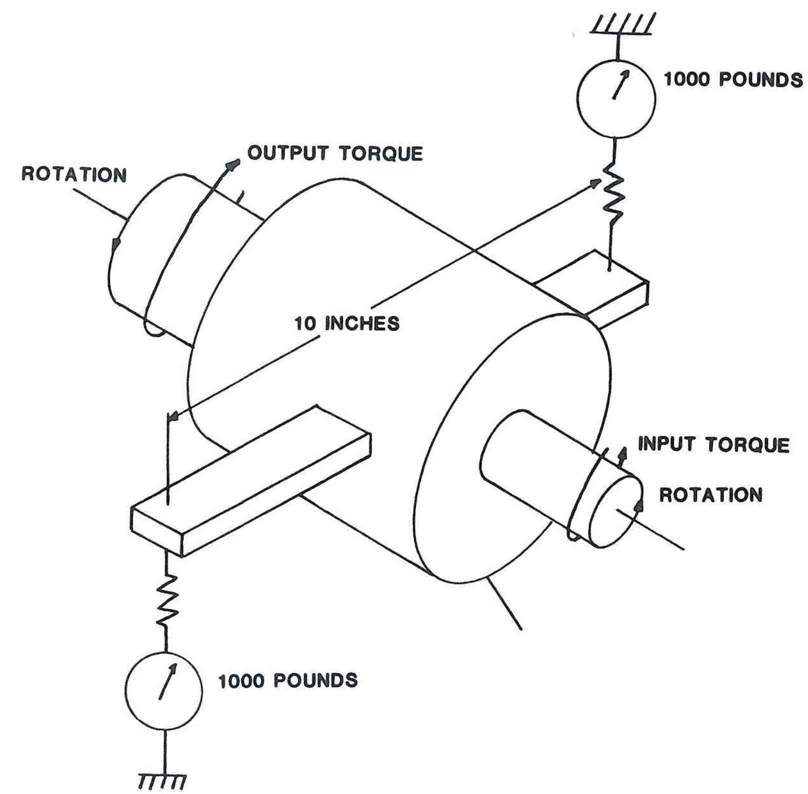


Figure 11.11 Gearbox housing torque measurement.

$$F = 1000 \text{ lb}$$

$$D = 10 \text{ in.}$$

therefore, the housing torque is 10,000 in.-lb. If the gear box ratio is 2:1, the input torque T_i is one-half the output torque T_o, and if the input and output shafts rotate in the same direction:

$$T_{HSG} = T_o - T_i = \tfrac{1}{2}T_o$$

and the output torque is twice the housing torque, or 20,000 in.-lb.

Temperature, pressure, and vibration measurements are covered later in the section "Condition Monitoring."

Reporting

The effort expended in conducting a test may be wasted if the test procedures and results are not carefully documented. The test items should be documented with part and serial numbers. Test setups should be described with photographs and sketches. Test results should be presented completely in the form of tables and graphs. If failures occur, they should be extensively documented as well as the modification incorporated to resolve the problem. The final report may be referred to years after the test program has been completed and the memory dimmed. Following is a general outline for a final report:

Objective
Summary
Test equipment
Test item
Test plan
Results
Discussion
Conclusions and recommendations

SPECIAL TESTS

Sound Testing

Measurement of the sound level of gearboxes is becoming increasingly important as government standards concerning equipment noise generation become more stringent. In addition to noise testing a brief discussion of sound fundamentals is presented in this section.

As a gearbox vibrates, a pressure oscillation in the surrounding medium (usually air) is generated. The transmission of the pressure vibration is called a sound wave and as it travels through the medium it can be detected by some form of receiver such as a microphone or the human ear. The sound pressure level of a gearbox is conventionally specified in decibels (dB) at a given distance from the gearbox. The sound pressure level L_p is the ratio of the pressure of the sound being measured to a reference pressure:

$$L_p = 20 \log_{10} \frac{p}{p_0}$$

where

p = sound pressure being measured, N/m^2
p_0 = reference pressure, N/m^2

(*Note*: psi $\times$ 6893 = N/m^2.)

The reference pressure is conventionally taken as 20 $\mu N/m^2$ (20×10^{-6} N/m^2), which is approximately the threshold of normal hearing at a frequency of 1000 Hz. As an example, if the sound pressure 2 ft from a gearbox is measured as 0.0010 psi (6.893 N/m^2), the sound pressure referred to 20 $\mu N/m^2$ is

A, B, AND C ELECTRICAL WEIGHTING NETWORKS FOR THE SOUND-LEVEL METER

These numbers assume a flat, diffuse-field response for the sound-level meter and microphone.

FREQUENCY Hz	A-WEIGHTING RELATIVE RESPONSE, dB	B-WEIGHTING RELATIVE RESPONSE, dB	C-WEIGHTING RELATIVE RESPONSE, dB	FREQUENCY Hz	A-WEIGHTING RELATIVE RESPONSE, dB	B-WEIGHTING RELATIVE RESPONSE, dB	C-WEIGHTING RELATIVE RESPONSE, dB
10	−70.4	−38.2	−14.3	500	−3.2	−0.3	0
12.5	−63.4	−33.2	−11.2	630	−1.9	−0.1	0
16	−56.7	−28.5	−8.5	800	−0.8	0	0
20	−50.5	−24.2	−6.2	1,000	0	0	0
25	−44.7	−20.4	−4.4	1,250	+0.6	0	0
31.5	−39.4	−17.1	−3.0	1,600	+1.0	0	−0.1
40	−34.6	−14.2	−2.0	2,000	+1.2	−0.1	−0.2
50	−30.2	−11.6	−1.3	2,500	+1.3	−0.2	−0.3
63	−26.2	−9.3	−0.8	3,150	+1.2	−0.4	−0.5
80	−22.5	−7.4	−0.5	4,000	+1.0	−0.7	−0.8
100	−19.1	−5.6	−0.3	5,000	+0.5	−1.2	−1.3
125	−16.1	−4.2	−0.2	6,300	−0.1	−1.9	−2.0
160	−13.4	−3.0	−0.1	8,000	−1.1	−2.9	−3.0
200	−10.9	−2.0	0	10,000	−2.5	−4.3	−4.4
250	−8.6	−1.3	0	12,500	−4.3	−6.1	−6.2
315	−6.6	−0.8	0	16,000	−6.6	−8.4	−8.5
400	−4.8	−0.5	0	20,000	−9.3	−11.1	−11.2

FREQUENCY RESPONSES

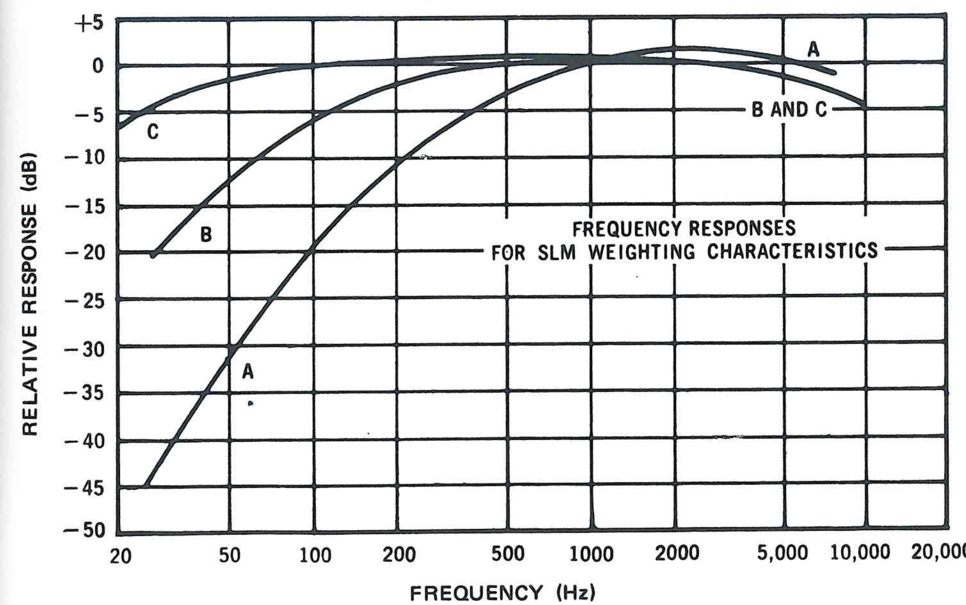


Figure 11.12 Sound meter frequency response. (From Ref. 1.)

$$L_p = 20 \log_{10} \frac{6.89 \text{ N/m}^2}{20 \times 10^{-6} \text{ N/m}^2} = 110.7 \text{ dB}$$

The sound-level meters that are used to measure decibels have a frequency response which is better than the human ear. In other words, at specific frequencies the meter will record sound a human being would not hear. To compensate for this fact, various filters have been incorporated into sound meters so that the measurements will approximate what the ear would record. As shown in Figure 11.12, three weighing scales, A, B, and C, have been established. The A scale matches the human ear's response at sound levels below 55 dB, the B scale at levels between 55 and 85 dB, and the C scale above 85 dB. The A scale is most commonly specified due to its use by OSHA for measurements up to 115 dB. Therefore, in the example above, the gearbox sound level measured by a meter set on the A scale would be defined as 110.7 dBA.

In order to analyze the source of gearbox noise, sound meter readings can be filtered to register only a limited range of frequencies. Frequently, octave and 1/3-octave bands are specified, as defined in Table 11.1.

Figure 11.13 presents the results of an octave band analysis of a two-stage parallel shaft gearbox with the high-speed mesh operating at 18,000 rpm. The lower curve is a measure of the background noise, including the prime mover decoupled from the gearbox. When even finer analysis of noise data is required, filters with bandwidths down to 2 Hz are used. A real-time analyzer can be employed to look at all frequencies simultaneously.

American Gear Manufacturers Association (AGMA) Standard 295.04 [2] defines the instrumentation and procedures to be used for sound measurement of high-speed helical and herringbone gear drives and presents typical maximum sound levels as shown in Figure 11.14.

Sound test speed and load conditions, according to the standard, are to be agreed upon by the manufacturer and purchaser. The microphone is to be located perpendicular to the center of the gear unit's vertical surface, but not less than 1 ft above the floor or plate. The distance between the gear unit's vertical surface and microphone is to be the normal working distance of the closest employee or as in the following table:

Gearbox center distance (in.)	Microphone distance (ft)
4 and below	3
4-15	5
15 and over	10

Table 11.1 Continuous Octave and One-Third Octave Frequency Bands

	Frequency (Hz)					
	Octave			One-third octave		
Band	Lower band limit	Center	Upper band limit	Lower band limit	Center	Upper band limit
12	11	16	22	14.1	16	17.8
13				17.8	20	22.4
14				22.4	25	28.2
15	22	31.5	44	28.2	31.5	35.5
16				35.5	40	44.7
17				44.7	50	56.2
18	44	63	88	56.2	63	70.8
19				70.8	80	89.1
20				89.1	100	112
21	88	125	177	112	125	141
22				141	160	178
23				178	200	224
24	177	250	355	224	250	282
25				282	315	355
26				355	400	447
27	355	500	710	447	500	562
28				562	630	708
29				708	800	891
30	710	1,000	1,420	891	1,000	1,122
31				1,122	1,250	1,413
32				1,413	1,600	1,778
33	1,420	2,000	2,840	1,778	2,000	2,239
34				2,239	2,500	2,818
35				2,818	3,150	3,548
36	2,840	4,000	5,680	3,548	4,000	4,467
37				4,467	5,000	5,623
38				5,623	6,300	7,079
39	5,680	8,000	11,360	7,079	8,000	8,913
40				8,913	10,000	11,220
41				11,220	12,500	14,130
42	11,360	16,000	22,720	14,130	16,000	17,780
43				17,780	20,000	22,390

Source: Ref. 1.

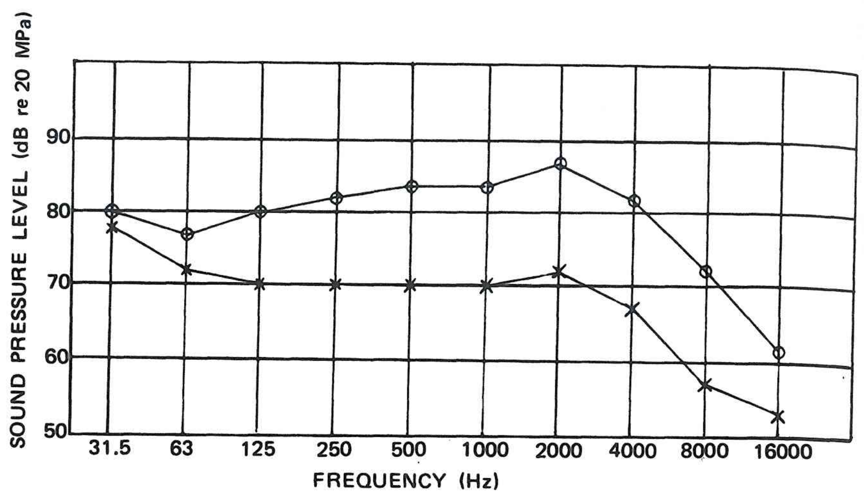


Figure 11.13 Octave band analysis (end location at 3 ft and 14,200 rpm).

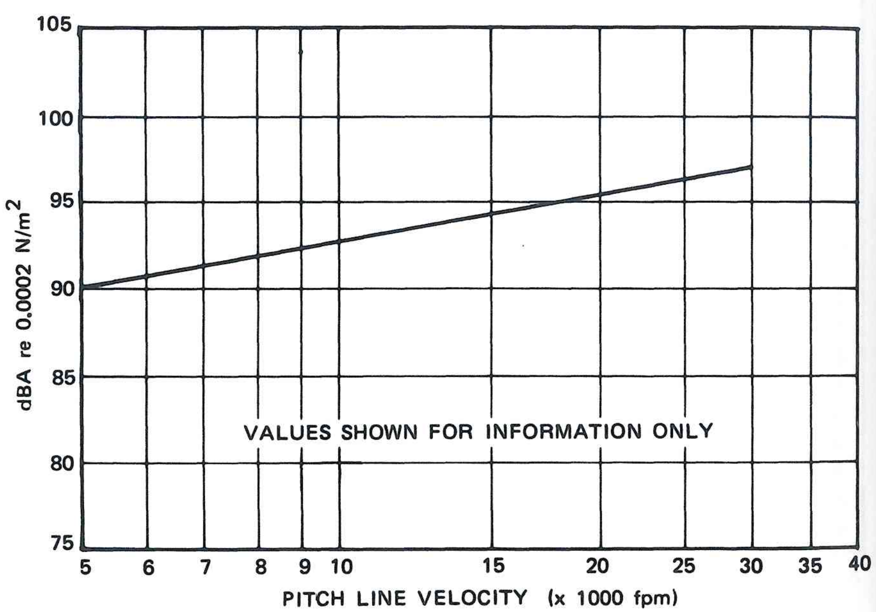


Figure 11.14 Typical maximum sound pressure levels versus high-speed mesh pitch line velocity. Note: In case of multireduction or increasing gear sets within one housing use the pitch line velocity of the highest speed set. (From Ref. 2.)

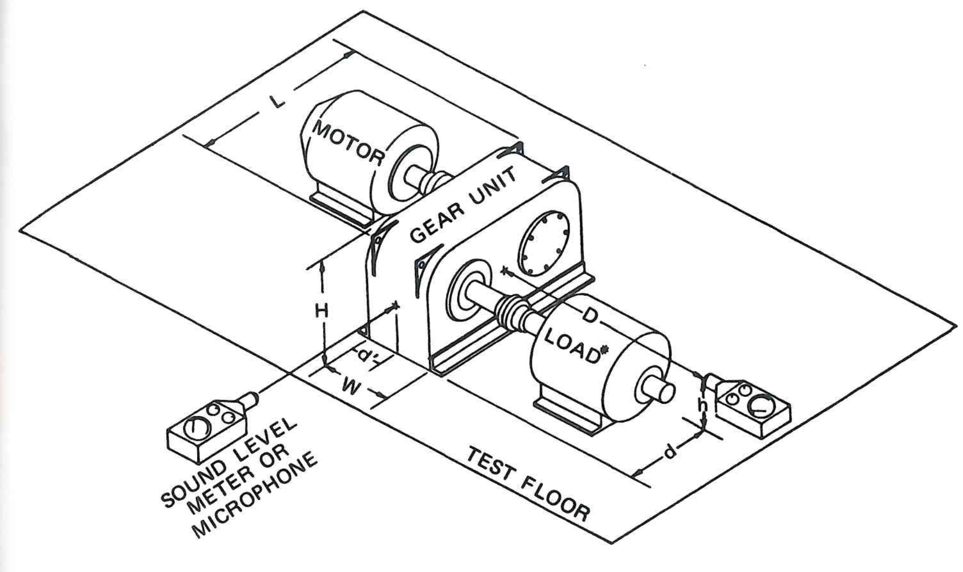


Figure 11.15 Sound test microphone position. Key: L, length of gear unit; H, height of gear unit; W, width of gear unit; D, distance of microphone perpendicular to unit, as specified in standard for size, h, height of microphone perpendicular to floor (H/2); d, distance of microphone from corner of unit (1/2) or (W/2). *Note: Load is optional for factory testing. (From Ref. 3.).

Figure 11.15 shows the sound test microphone position. The sound level is to measured with and without the gear unit operating so as to correct the unit rating for the ambient sound pressure level. Corrections are presented in Table 11.2.

Table 11.2 Corrections for Ambient Sound Pressure Levels

Difference between gear unit and ambient sound pressure levels (dB re 20 $\mu N/m^2$)	Correction to be subtracted from gear unit sound pressure level (dB re 20 $\mu N/m^2$)
3 or less	3
4 and 5	2
6–9	1
10 or greater	0

Source: Ref. 2.

The overall sound level of a mechanical system is made up of the sound levels of the components therein. For instance, the components of a generator drive system when measured individually may exhibit the following noise levels:

Prime mover:	88 dBA
Gearbox:	82 dBA
Generator:	95 dBA

The following expression is used to combine the component noise levels to analytically define an overall noise level:

$$L_p = 10 \log_{10} \sum_{i=1}^{N} 10^{0.1 a_i}$$

where

N = number of components
a_i = sound level of each component

In the example,

$$L_p = 10 \log_{10} (10^{8.8} + 10^{8.2} + 10^{9.5})$$
$$= 96 \text{ dBA}$$

It can be seen that in order to reduce the noise level of this system, the major contributor, the generator, must be worked on.

Noise generation is influenced by the design and manufacture of the gearbox components. The ideal situation in a gear mesh would be to transmit power with no change in the angular velocity of the gear shaft. In such a perfect gear mesh there would be no accelerations or decelerations of the gear shafts, which provide the energy for vibration and noise generation. Of course, there are no perfect gear meshes, and errors in tooth spacing, profile, or runout will always be present, resulting in accelerations and decelerations of the gear shafts and noise.

Design of gear teeth can reduce noise levels. For instance, increasing overlap or changing from spur to helical gearing will reduce noise. In these cases the changes result in smoother load distribution from tooth to tooth as they go through the mesh. Changes in design to reduce noise, however, might adversely affect other parameters. For instance, increasing contact ratio by going to a finer pitch might compromise bending strength. Other design changes that tend to reduce noise are:

Reduced pitch line velocity
Proper profile and lead modifications
Lower pressure angles

It must be realized that detail gear tooth geometry changes can only account for changes in sound level of up to approximately 4 dBA. Also, no matter how optimum the design, the quality of the gearing will determine the sound level. Table 11.3 lists the common sources of sound in a gear drive system.

Sound can be transmitted through the air or be structure borne. Structure-borne noise may travel through the support structures and radiate at some point other than its source. Because of this the noise measurements near a gearbox in an operating system may be quite different from those measured during a bench test. Also, when connected to the prime mover and load, the torsional response of the system may be different from the bench test and result in different noise measurements.

In many cases when it is determined that the noise generation of a system is excessive yet not an indication of some malfunction, the most practical method of noise reduction is the use of an acoustical enclosure. The effectiveness of such an enclosure is very dependent on eliminating all openings. Such an enclosure will also affect the ambient temperature and housing heat dissipation ability of the gearbox and means must be provided to cool the unit properly.

Efficiency Testing

Gearboxes are an extremely efficient means of transmitting power. Depending on the operating conditions, design, and manufacturing techniques, the loss per mesh of spur, helical, or bevel gears will vary between approximately ½ to 3%. Analytical means are available to calculate gearbox efficiency, but in order to be confident that an accurate measure of a unit's efficiency is known, the value must be arrived at by testing.

Obviously, the efficiency of the gearbox affects the total system efficiency and the rating of the prime mover required. The power lost in the gearbox, however, also has a large impact on system design since it must be dissipated as heat.

Many low-speed gearboxes have a self-contained splash lubrication system and the power rating of the unit may be limited by the oil temperature rise rather than a mechanical limit. This is the reason gearbox catalogs list a thermal rating in addition to a mechanical rating. When the heat generated in a unit is greater than can be dissipated through the casing an external lubrication system including a cooler is required. The heat generation or efficiency of the gearbox will determine the size of cooler required.

The heat generation can be calculated as follows:

$$Q = MC_p \, \Delta T$$

where

Q = heat generated, Btu/min
C_p = specific heat, Btu/lb-°F

Table 11.3 Common Sources of Airborne and Structure-borne Sounds Generated in Gear Drive Systems[a]

Instruments that provide the operator with not only the amplitude of the vibration or noise, but also the predominant frequencies can be a tremendous aid in determining sources. These causes normally present themselves as follows:

1. *Balance*: Residual unbalance presents itself at a frequency equal to once per shaft revolution and it will increase in amplitude as speed is increased.
2. *Alignment*: Misalignment will present itself at once or sometimes twice and three times per shaft revolution. However, the amplitude will remain fairly constant with speed changes.
3. *Friction*: This is difficult to pinpoint by vibration and noise frequency. Amplitude may be very high when continuous sliding occurs. It may also be random, high-amplitude, shock-type pulses, as in hydrodynamic bearing rubbing. It may be irregular and often violent.
4. *Looseness*: This may cause unbalance, misalignment and friction rubbing at moderate and high speeds. At low speeds, it may display itself as an irregular rattle. Often it shows up at twice shaft rotational speed.
5. *Distortion*: This is often an indirect cause of vibration and noise, which also leads to unbalance, misalignment, or friction. It will tend to change in amplitude with load or operating temperatures, when speed is held constant.
6. *Critical speeds*: These occur through any given speed range and are points at which a rotating system likes to vibrate torsionally or laterally at a particular frequency. Rotors characteristically show violent increase in amplitude at particular critical speeds but are fairly stable above and below these speeds. A critical speed may change frequency with load and temperatures.
7. *Resonances*: These also display themselves as frequencies at which system members like to vibrate. The distinction from critical speeds is that resonances occur in other than rotating members, and affect alignment. Resonances occur at fixed frequencies and change in amplitudes with load and temperature.
8. *Tooth mesh (i.e., tooth contact)*: This will show up at tooth mesh frequency (i.e., rotating speed times number of teeth) and multiples of this mesh frequency.
9. *Bearing instability*: Bad antifriction bearings will cause high-frequency vibration at several times rotational speed, also friction vibration will occur. Hydrodynamic bearings, lightly loaded, will tend to whirl at 0.43 to 0.47 times the rotational speed. This so-called "half-frequency whirl" will "on-set" violently with speed or temperature changes and may continue until the rotor is completely stopped.
10. *System pulses*: These may occur in many types of systems, such as the vane-pass frequency of a pump or compressor (rotational speed times the number of vanes), and the beating of reciprocating engines which cause frequencies at one-half and one-quarter rotational speed at various amplitudes.
11. *Windage*: Couplings and other rotating parts generally create broadband noise, but can be at a bolt pass frequency or fan blade pass frequency.

[a]All of these types of vibrations and noise frequencies can be generated in a gear drive. Major frequencies can interact and cause frequency modulation and phase shifts. Any combination, sum difference, and multiple (harmonics) of prime frequencies can occur if the forcing magnitude and system freedoms are such that they will cause and allow the generated vibration to become predominant. Generally, only the prime frequencies will present themselves as problem modes. However, sometimes very elusive frequencies appear, such as *periodic cutting machine error appearing on one of the gears.*

Source: Ref. 3.

M = oil flow, lb/min
ΔT = oil temperature rise across the gearbox, °F

Note:

$$\frac{\text{Btu/min}}{42.44} = \text{horsepower}$$

C_p of oil $\cong$ 0.5 Btu/lb-°F

1 gpm $\cong$ 7.5 lb/min of oil

By accurately measuring the temperature of the oil entering and leaving the gearbox and the oil flow, the heat rejected to the oil can be calculated as shown above.

For example, a gearbox with oil in temperature of 130°F, oil-out temperature of 160°F and oil flow of 20 gpm will reject 848 Btu/min (20 hp) to the oil. If the gearbox is transmitting 1000 hp, the efficiency can be calculated as follows:

$$E = \frac{P_t - P_1}{P_t} = \frac{1000 - 20}{1000} = 98\%$$

where

P_t = total hp transmitted
P_1 = power loss

It must be realized that in addition to the heat rejected to the oil, heat is dissipated to the atmosphere by radiation and convection through the casing. To arrive at an estimate of the heat dissipated through the casing, the following heat transfer equation may be used:

$$H = CA(Tc - Ta)$$

where

H = heat dissipated by the gear casing, Btu/hr
C = combined coefficient of radiation and convection, Btu/hr/ft/°F
Tc = casing temperature, °F
Ta = ambient temperature, °F

The value of C depends on the material, roughness, and color of the housing and the velocity of air around the housing and can vary from approximately 0.5 to 3.0. To eliminate the variable of casing radiation and convection, the gear housing can be insulated for an efficiency test. In this way all the heat generated in the unit will be carried away by the oil.

The heat balance is the most widely used method of measuring gear unit efficiency. A more direct way is to measure input and output torque, but this method requires exact and expensive instrumentation. In the case of a regenerative or back-to-back test, only input torque need be measured and this will monitor the torque required to drive the two locked-up gear trains. Since the prime mover supplies only enough power to make up the losses in the system, the torque lost in each gearbox is one-half the input torque.

Power losses in the gearbox can be divided in two categories, friction losses in the gear mesh and at the bearing interfaces and windage or churning losses. In high-speed boxes, windage and churning losses can be considerable amounting to up to half the total power loss. An easy way to determine these losses is to operate the gearbox at full speed, no load, and then perform a heat balance.

A high-speed box is considered to be one with pitch line velocity ranging from 5000 to 20,000 fpm. Above 20,000 fpm windage and churning can be a limiting factor to the success of the design. For instance, some wide-face-width helical gearboxes operating at pitch line velocities above 20,000 fpm have experienced severe heat generation problems due to pumping of oil and air along the face width. In high-speed double helical gears the hands of helix should always be selected so as to pump oil away from the apex.

When it is determined that a gearbox has excessive windage and churning losses, the situation can be relieved by strategically placing baffles or screens in the unit. In low-speed units using splash lubrication it is beneficial to shroud the gears dipping into the sump oil. With high-speed units the sump oil level should be below the gears and the sump should be separated from the gear meshes by a baffle plate or screen. Minimizing windage and churning losses is more an art than a science and can require trial-and-error solutions verified only by testing.

Lubrication system parameters can have a profound effect on gearbox efficiency. For instance, efficiency will increase with reduced oil flow; obviously, this is due to reduced churning. The limit to this technique is the increase in temperature rise across the box and in the individual components as oil flow is reduced. Increasing oil in temperature will increase efficiency since the oil viscosity will decrease and therefore reduce churning. Changing to a lower-viscosity oil will have the same effect. Proper scavenging of the unit is extremely important in minimizing churning; therefore, great care must be taken in the location and size of drain holes. Also, back pressure must be minimized.

INITIAL FIELD STARTUP

Prior to starting the equipment, the following preliminary checks should be performed:

1. Check oil level and ensure that the proper oil is being used.
2. Tighten all pipe connections.
3. Check all electrical connections.
4. Tighten all mounting and gearbox bolts with proper torque.
5. Check mounting of all gauges, switches, and so on.
6. Check all couplings for proper installation and alignment.
7. Check inspection cover installations.

The following instructions pertain to the initial startup:

1. The unit should be preoiled to ensure lubrication of the journal bearings at startup.
2. The gearbox should be started slowly under as light a load as possible. Observe that the rotation is in the proper direction. Check the system oil pressure.
3. After starting, when the oil has been circulated, the unit should be stopped and sufficient oil added to bring the sight gage oil level up to the specified amount.
4. As the unit is brought up to operating speed, it should be continuously monitored for excessive noise, vibration, or temperature. If any of these occur, shut down immediately, determine the cause, and take corrective action. Also check for oil leaks as the unit is initially operated.
5. If possible, operate at half load for the first 10 hr to allow final breaking in of the gear tooth surfaces.
6. After the initial 50 hr of operation the oil in a new unit should be drained and the case flushed with SAE 10 straight mineral flushing oil containing no additives. Drain the flushing oil and refill with the recommended lubricant to proper level.
7. After the initial 50 hr of operation check all coupling alignments and retorque all bolts. Check all piping connections and tighten if necessary.

If starts are made in a cold environment, consideration should be given to preheating the lubricant. Load should not be applied until the lubricant has attained operating temperature.

CONDITION MONITORING

On an operational gearbox the question always arises as to how to evaluate the condition of the unit and when to disassemble the equipment and inspect or replace internal components. Figure 11.16 presents a hazard function or what is traditionally called the "bathtub curve," which describes the failure rate of a unit at any particular point in its operating history. Initially, there is a high "infant mortality" period, which reflects failure due to assembly or

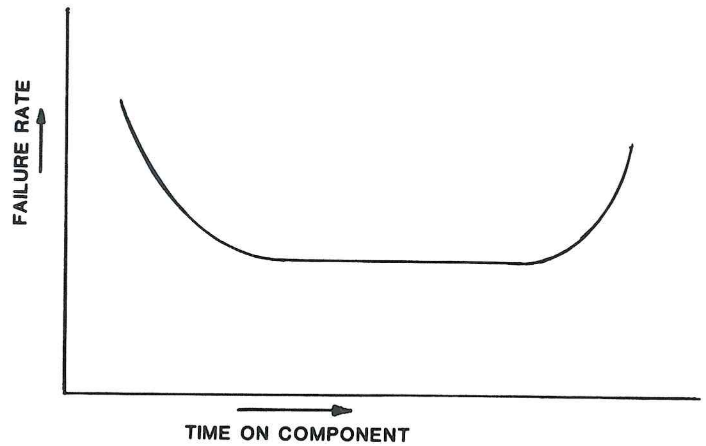


Figure 11.16 Hazard function.

manufacturing errors. For instance, an oil jet might have been clogged or a bearing preloaded at assembly. These early failures can often be screened out during an acceptance test program. Following the "infant mortality" phase, there is a period of constant failure rate where failure modes related to time appear. Examples of these are bearing or gear tooth fatigue, spline wear, and so on. At first glance it would appear reasonable that the unit be removed from service at the point in time where the failure rate begins to increase and the components inspected and reworked or replaced as necessary. This is not necessarily so, since removal and disassembly of the unit begins another operating cycle and the gearbox again would be subject to the high "infant mortality" failure rate. As a practical matter it is difficult to establish the point where the failure rate increases sufficiently to warrant overhaul since this requires an extensive data base. Gear units are subject to many different modes and within any given failure mode there will be significant scatter.

The most efficient method of determining when a gear unit requires service is to base repairs on the condition of the gearbox rather than overhaul it at an arbitrary time period. The basic idea is not to disturb equipment that is operating properly but inspect only items that exhibit potential failure symptoms. This type of predictive maintenance is termed "on condition" and requires instrumentation of a gear unit and the proper interpretation of the data provided. The goals of this technique are threefold:

1. Detect gear units which are operating abnormally.
2. Diagnose which internal component is deteriorating.
3. Predict how long the unit can function in this condition before corrective action must be taken.

Condition monitoring systems are expensive and require discipline to set up and use effectively. It is difficult to define the normal operating parameters of a system and this initial step is vital in an "on condition" program. Sometimes the normal operating parameters are termed the "signature" of a component. Once this is established it is relatively easy to identify abnormal operation. When an "on condition" program is achieved, the benefits are as follows:

1. Reduction in downtime since potentially catastrophic failures are detected and corrected. This also leads to a reduction in repair costs.
2. Enables efficient scheduling of equipment shutdown since warning of a failure is detected well in advance of the event.
3. Shutdown periods are shorter because unnecessary work is avoided and the work required is known in advance so that preparations can be made.

In the following pages the various methods used to monitor gearbox condition will be discussed. These include:

Vibration
Noise
Shaft position
Oil temperature
Oil pressure
Oil analysis
Chip collection

Oil Analysis

Gearbox condition can be monitored by the analysis of oil samples. One widely used technique is SOAP (Spectrographic Oil Analysis Program). An oil sample is taken periodically from the gearbox sump and sent to an analytical laboratory, where it is burned and the light waves passed through a spectrometer. The spectrum of light waves given off by the sample yields information as to the types of wear metals suspended in the oil and their quantity.

In principle, if it is possible to identify and measure the wear metals present in the lubricant, one can identify which internal gearbox component is deteriorating. Also, by monitoring the rate of change of wear metal from sample to sample, conclusions can be drawn concerning the gearbox condition and when maintenance will be required. Figure 11.17 illustrates the information presented in a typical SOAP laboratory analysis.

To use SOAP properly, a gearbox history base must be established to determine what type and quantity of wear metals the unit normally generates. Also, a listing of materials used in the gearbox must be available. With this information SOAP can be used to predict failures. For instance, an increase in silver might mean that a silver-plated antifriction bearing retainer is wearing. Tin and

DATE		HOURS		OIL ADDED	PHYSICAL PROPERTY TESTS			SPECTROCHEMICAL ANALYSIS METALS:PARTS PER MILLION BY WEIGHT																
SAMPLED	RECEIVED	OIL	UNIT		VIS. AT 100°F	TOTAL ACID NO.	WATER	IRON	LEAD	COPPER	CHROMIUM	ALUMINUM	NICKEL	SILVER	TIN	SILICON	BORON	PHOSPHORUS	SODIUM	ZINC	CALCIUM	BARIUM	MAGNESIUM	TITANIUM
1/25	1/28	48.3	48.3	–	29.2	0.0	<.05	02.	02.	.0	02.	.0	.0	.0	.0	.0	.0	63.	.0	20.	.0	.0	.0	.0
2/1	2/10	76.3	76.3	–	28.9	0.03	<.05	04.	02.	.0	02.	.0	1.0	.0	.0	11.	1.	81.	7.	20.	50.	20.	.0	.0

Figure 11.17 Typical oil analysis form.

lead traces could indicate a journal bearing problem. A large percentage of iron might point to gear or bearing distress. Of course, practically all gearbox components contain some iron, but if other elements are also indicated, the cause of distress might be narrowed down to a specific component.

SOAP oil samples should always be taken from the same area of the unit. Oil sampling periods are in the order of every 10 to 100 hr of operating time and each oil sample should be placed in a clean vial marked with an identifying number, the date, and the number of operating hours of the unit. If readings indicate a problem, the sampling interval should be reduced. There is a time delay in getting results from the laboratory and a typical turnaround time might be 3 days.

SOAP readings can be influenced by external factors. For instance, oil changes or oil additions must be accurately recorded since they will reduce the percentage of wear metals. Also, the degree of filtration will affect the wear metals suspended in the fluid. Very fine filtration such as 3 μm will effectively remove all wear metals and make SOAP meaningless. It should also be recognized that large particles will settle out and not be detected by SOAP analysis. The effective detection range of the SOAP procedure is with particles in the size range 1 to 10 μm.

Increases in the wear metal percentage between samplings are more significant than the total wear metal content since an increase signifies a trend of abnormal operation. Therefore, a rapid increase in wear metal should be investigated even if the absolute percentage does not exceed the acceptance criterion.

When performing a SOAP analysis other oil characteristics are commonly monitored, such as acid number, viscosity, and water content. The acid number is relatively easy to check and will increase due to the presence of wear metals or due to overheating. Occasionally, one can detect acid oil by the odor alone, which is distinctive and unpleasant. Changes in oil viscosity indicate lubricant deterioration and can be monitored. Monitoring of acid number and/or viscosity can be done more quickly and economically than SOAP but does not yield as much information.

Vibration

Gear boxes are mass-elastic systems and therefore will vibrate when excited by internal or external influences. Vibration of gear units can be measured in several different ways. The motion of the shafts, both radial and axial, can be observed with noncontacting sensors. Casing motions can be measured using velocity or acceleration pickups. Generally, the following parameters are monitored:

1. Peak-to-peak mils displacement
2. Peak inches per second velocity

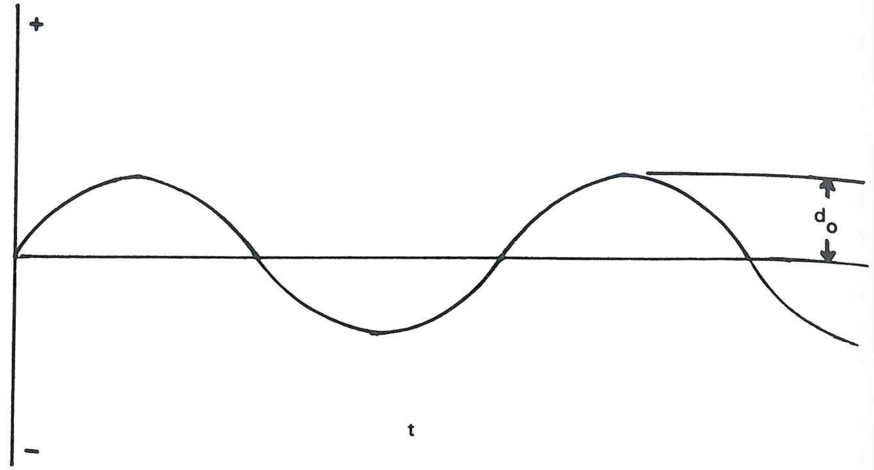


Figure 11.18 Simple harmonic motion.

3. Peak g's acceleration
4. Frequency of vibration

The relationship of vibratory displacement, velocity, and acceleration is as follows:

A vibration of simple harmonic motion which has a pure sinusoidal waveform has a displacement d (see Figure 11.18):

$$d = d_0 \sin \omega t$$

where

d_0 = ½ peak-to-peak displacement reading, in.
ω = frequency, rad/sec
= (2π)rpm/60
t = time, sec

Differentiating for the velocity v:

$$v = \omega d_0 \cos \omega t$$

and differentiating once more for the acceleration a:

$$a = -\omega^2 d_0 \sin \omega t$$

To calculate the G loading, let $\sin \omega t = 1$, and divide a by the acceleration of gravity:

$$G = \frac{\omega^2 d_0}{g}$$

where g is 386 in./sec^2.

The major contributors to gearbox vibration are:

1. Rotating shaft unbalance
2. Shaft assembly problems such as loose connections, bent shafts, and misalignment
3. Gear tooth inaccuracies
4. Component wear

AGMA Standard 426.01 [4] presents acceptable vibration levels, as shown in Figure 11.19. If the displacement amplitude is not obtainable at discrete frequencies, the standard allows either of the following:

1. A nominal unfiltered velocity level of 0.3 in./sec but not exceeding a maximum 2 mil displacement. (*Note*: a mil is 0.001 in.)
2. An unfiltered displacement level determined from Figure 11.19 using the shaft rotation speeds as discrete frequencies (i.e., 120 rpm = 20 Hz).

A gearbox may be acceptable according to the criteria of Figure 11.19 in acceptance testing at the manufacturer's facility, yet exhibit higher vibration levels in field service. The vibration levels may be adversely affected by factors not under control of the gear manufacturer, such as:

1. Inadequate foundation
2. Excessive shaft misalignment
3. Coupling components not tested with the gear unit
4. Resonance of base or other supporting structure
5. System torsional vibration
6. Motor magnetic center wobble
7. Unbalance or other forced vibration from other components in the system

Shaft vibration levels as shown in Figure 11.19 are measured by the use of noncontacting, eddy current proximity devices. A probe is positioned approximately 0.050 in. from a shaft and the distance (gap) between the probe tip and the shaft is translated to a voltage.

A proximity probe is basically a small coil of wire embedded in a ceramic tip on the end of a holder. This coil is supplied with a high-frequency voltage which produces a magnetic field radiating from the probe tip. Any conducting surface that lies within this magnetic field absorbs some of its power, and the closer to the tip the more power absorbed. The RF signal to the coil is supplied by a proximitor which has the secondary function of relaying the resultant field strength back to the readout instruments by reconverting it to a voltage.

Proximitors and probes are calibrated to give a linear relationship between readout voltage and distance of observed material. The range of this linear relationship depends on the type of probe and supply voltage, but is on the order of

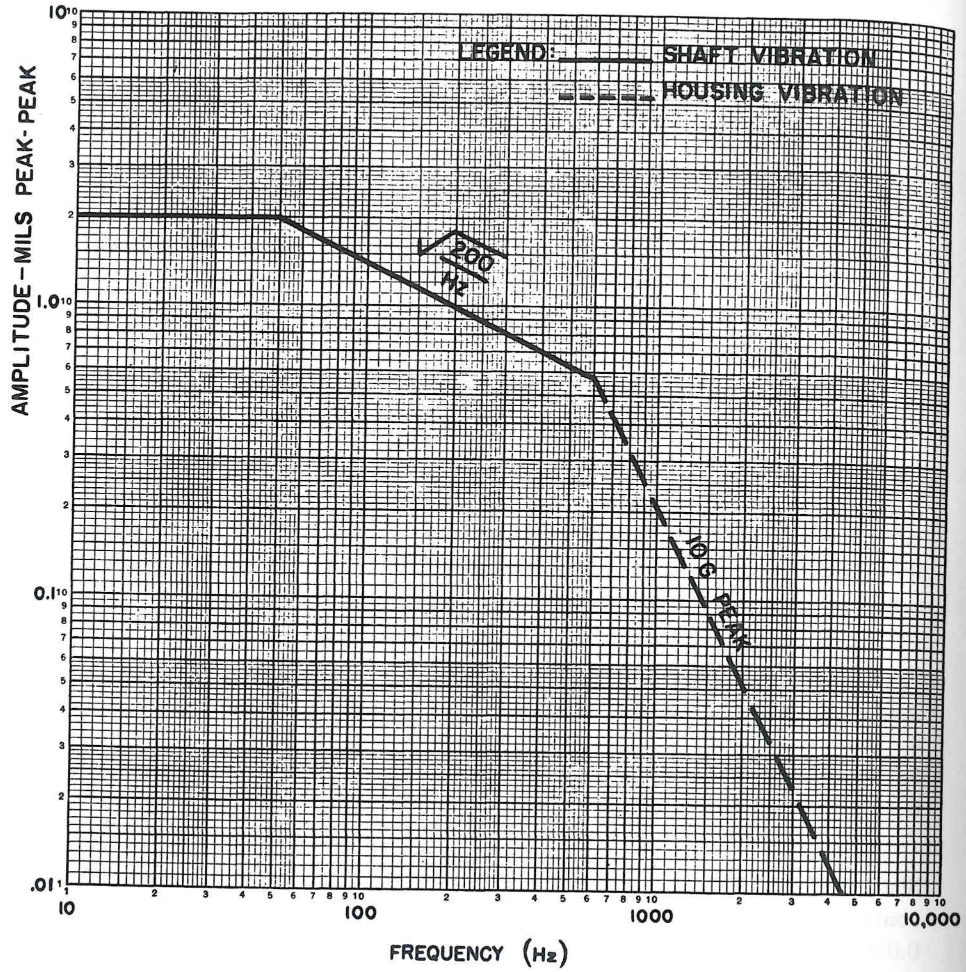


Figure 11.19 Acceptable shaft vibration levels. (From Ref. 4.)

80 mils. The readout obtained from a radially mounted probe is the electrically measured difference between the minimum and maximum distances of the probe from the shaft material (peak-to-peak amplitude).

The American Petroleum Institute has issued a standard [5] which defines the use of proximity probes. Figures 11.20 to 11.22 illustrate radial and axial probe arrangements and their application in a gearbox.

Note that in Figure 11.22 two radial probes 90° apart are mounted at the bearing positions. It is possible to have totally different vibration in two

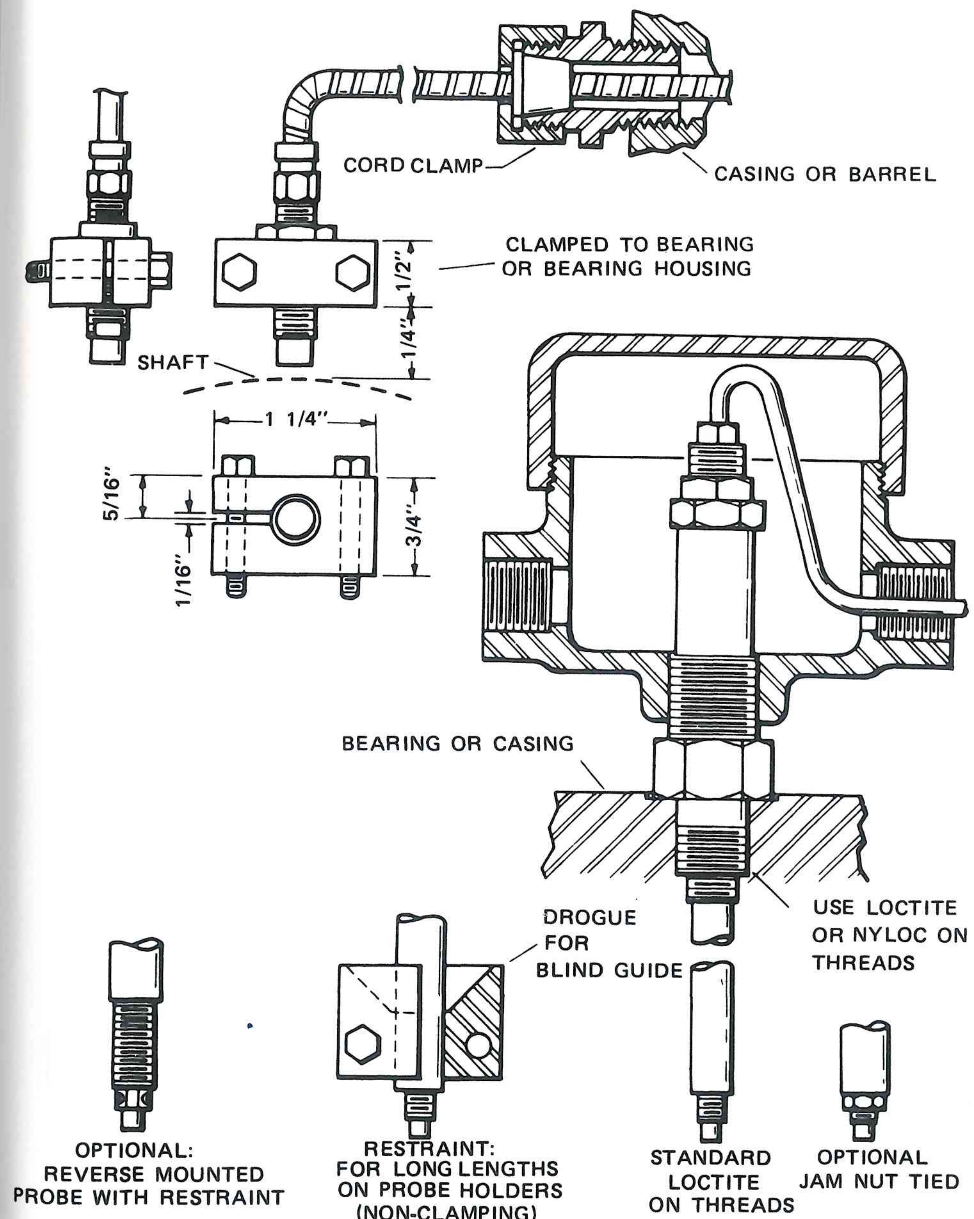


Figure 11.20 Typical radial probe arrangement. (From Ref. 5.)

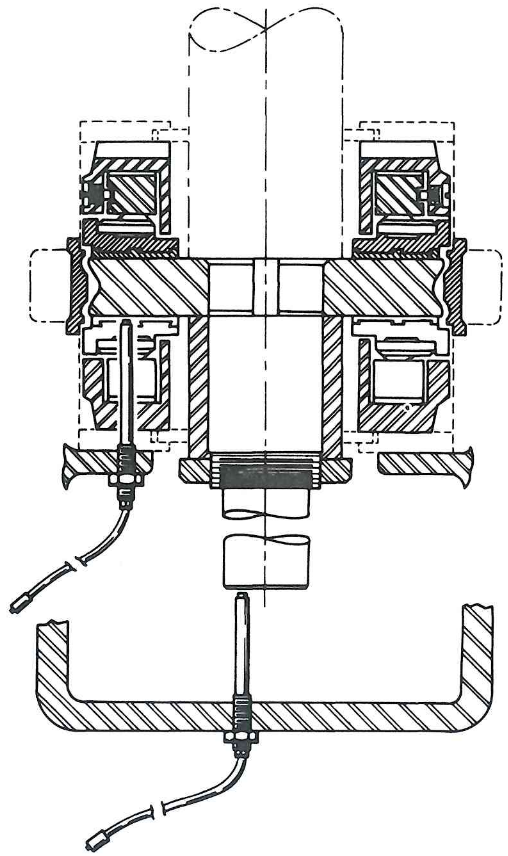
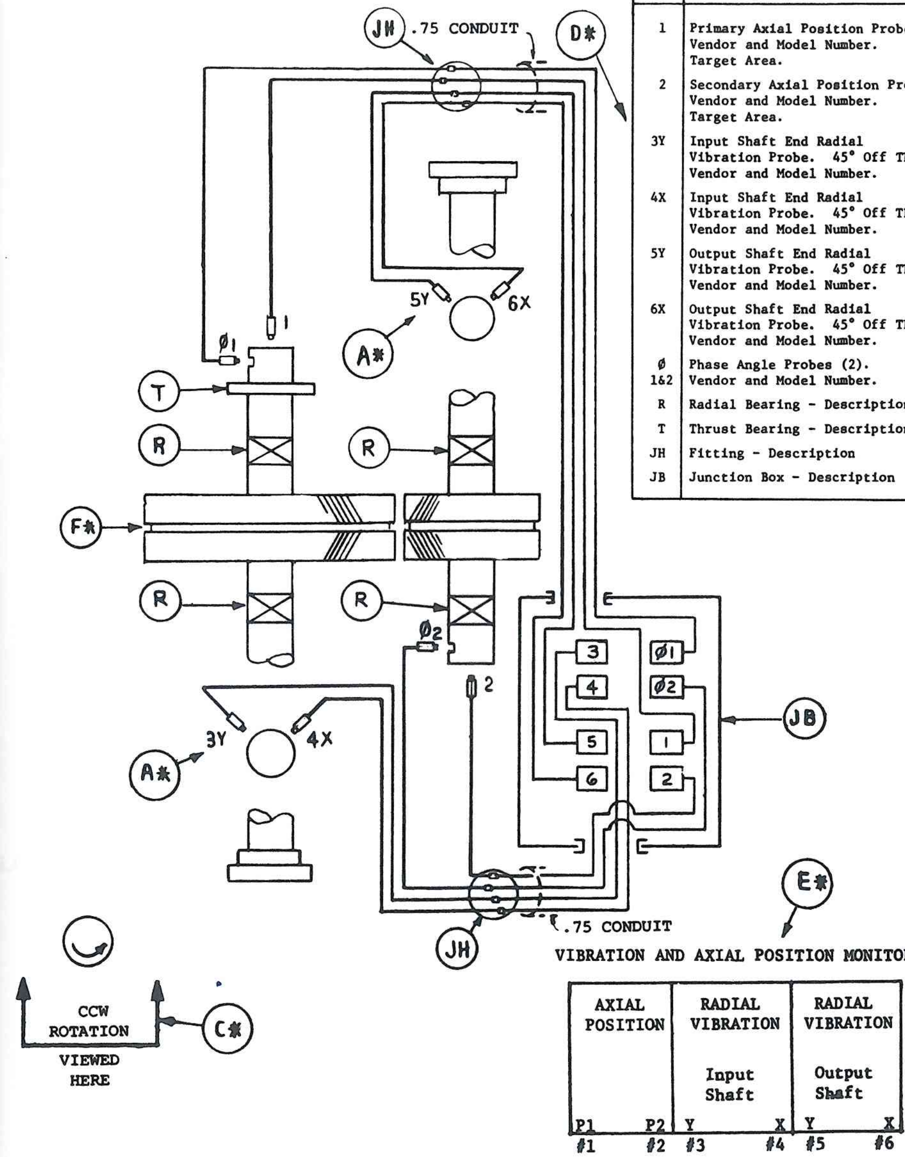


Figure 11.21 Typical axial probe installation. (From Ref. 5.)

perpendicular directions at one particular bearing. This is generally true of any vibration measurement, whether it be at the bearing or at the casing and readings in two planes should always be taken. Gearbox bearing probes are mounted 45° from the vertical center since quite often there is a split line at the horizontal.

On Figure 11.22 a phase angle probe is shown on each shaft. This is a transducer that observes a once-per-turn event such as a keyway. Its function is to provide a reference mark and timer for speed, phase angle, frequency measurements, and all data acquistiion. Figure 11.23 shows a phase angle probe installation. Each time the keyway passes the probe a voltage pulse results. This pulse provides a physical reference on the shaft which can be used to measure the high spot of the shaft.



ITEM	DESCRIPTION
1	Primary Axial Position Probe. Vendor and Model Number. Target Area.
2	Secondary Axial Position Probe. Vendor and Model Number. Target Area.
3Y	Input Shaft End Radial Vibration Probe. 45° Off TDC. Vendor and Model Number.
4X	Input Shaft End Radial Vibration Probe. 45° Off TDC. Vendor and Model Number.
5Y	Output Shaft End Radial Vibration Probe. 45° Off TDC. Vendor and Model Number.
6X	Output Shaft End Radial Vibration Probe. 45° Off TDC. Vendor and Model Number.
Ø 1&2	Phase Angle Probes (2). Vendor and Model Number.
R	Radial Bearing - Description
T	Thrust Bearing - Description
JH	Fitting - Description
JB	Junction Box - Description

Figure 11.22 Typical system arrangement for double helical gear. (From Ref. 5.)

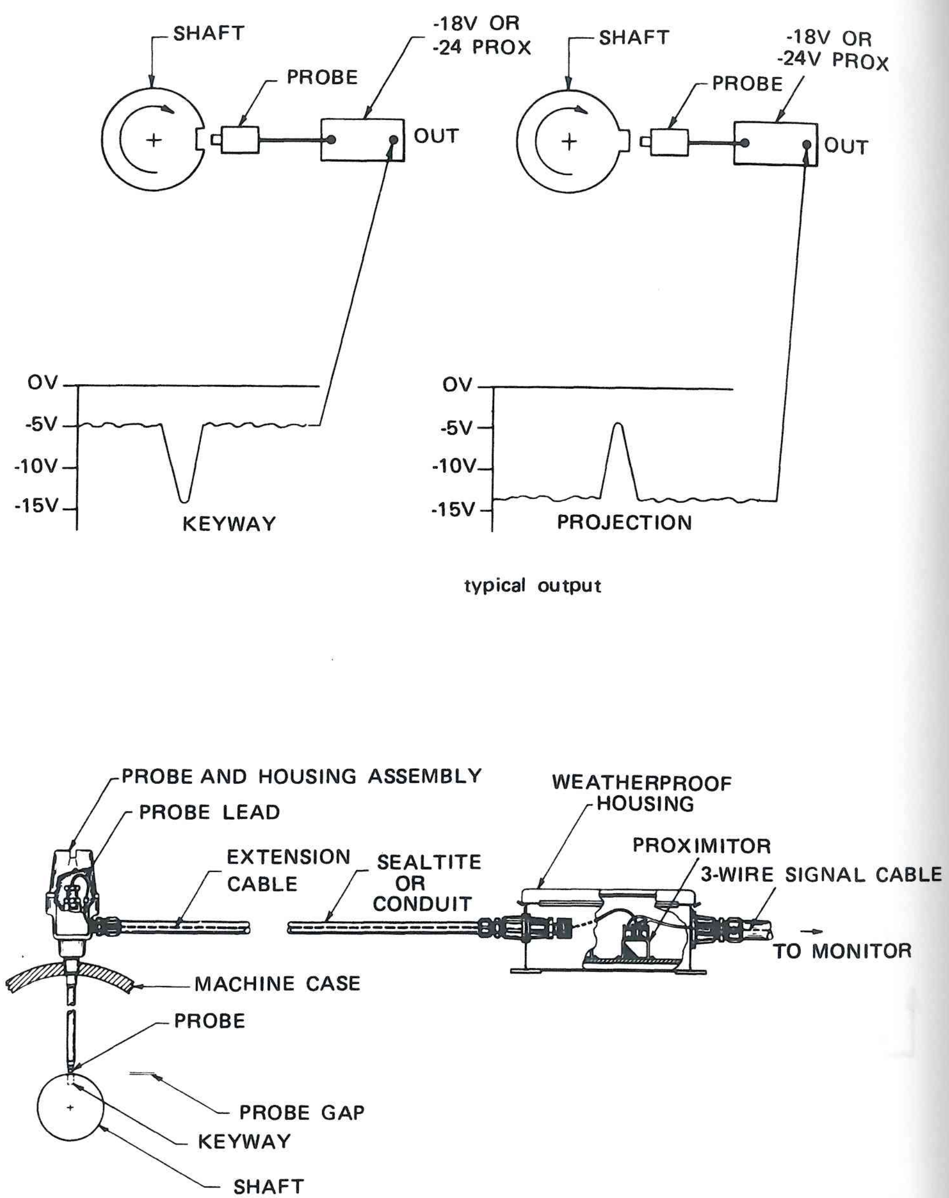


Figure 11.23 Phase angle probe. (Courtesy of Bently Nevada, Minden, Nevada.)

To illustrate the type of data acquired by proximity probes, refer to Figure 11.24. The oscilloscope photograph displays unfiltered time-domain traces from both horizontal and vertical probes, as well as the same signals filtered at rotational frequency. The inclusion of a phase angle mark, KØ, identifies one complete revolution of the shaft. In the middle of the picture the shaft orbit can be observed as an unfiltered motion, and filtered exactly at running speed. In addition, a spectrum analysis of each signal is presented, and the various components identified.

There are two types of oscilloscope readings presented in the center of Figure 11.24. One is the time-base mode, where the sinusoidal-type waveform representing the shaft motion is displayed. This shows the position of the shaft relative to the input transducer versus time horizontally across the cathode ray tube of the oscilloscope. The other display is the orbit presentation where the output from two separate proximity probes at 90° to one another are shown in the X-Y mode of the oscilloscope. In this mode the centerline motion of the shaft is displayed. If the probes are mounted at the bearing, the orbit is a presentation of the motion of the shaft centerline with relationship to the bearing.

When measuring shaft vibration with proximity probes it is possible to get erroneous readings caused by physical and mechanical deformities in the shaft material. These erroneous readings are sometimes referred to as glitches.

Obviously, there will always be some mechanical runout in the shaft. This may be measured by a dial indicator mounted in the probe area of the shaft, which itself is supported at the bearing journals in V blocks or rollers. The shaft runout at the probe locations may be subtracted from the vibration readings to arrive at actual vibration levels, provided that the runout is shown to be in phase with the vibration.

In addition to runout, other mechanical problems that will cause glitches are bowed shafts and surface imperfections such as scratches, dents, and burrs. To correct these problems the shaft surface may have to be remachined or reground. If the surface irregularities are very minor, a redressing of the surface with an oil-wetted fine emery cloth may eliminate the surface imperfections.

In addition to mechanical runout there is the possibility of electrically induced runout due to residual magnetism which can be corrected by having the shaft degaussed. Other potential causes of glitch could be metallurgical segregation or residual stress concentrations.

Before assmbling a shaft that has probe areas a bench test should be conducted to ensure that glitch has been held to minimum (in the order of 0.00025 in.) so as not to influence vibration readings excessively. After functional inspection it is advisable to coat the probe areas with an epoxy resin which can remain in position for the life of the machine. This coating will not affect probe readings but will protect the probe area from corrosion and minor mechanical damage. The shaft position probes provide a vibration measurement of the

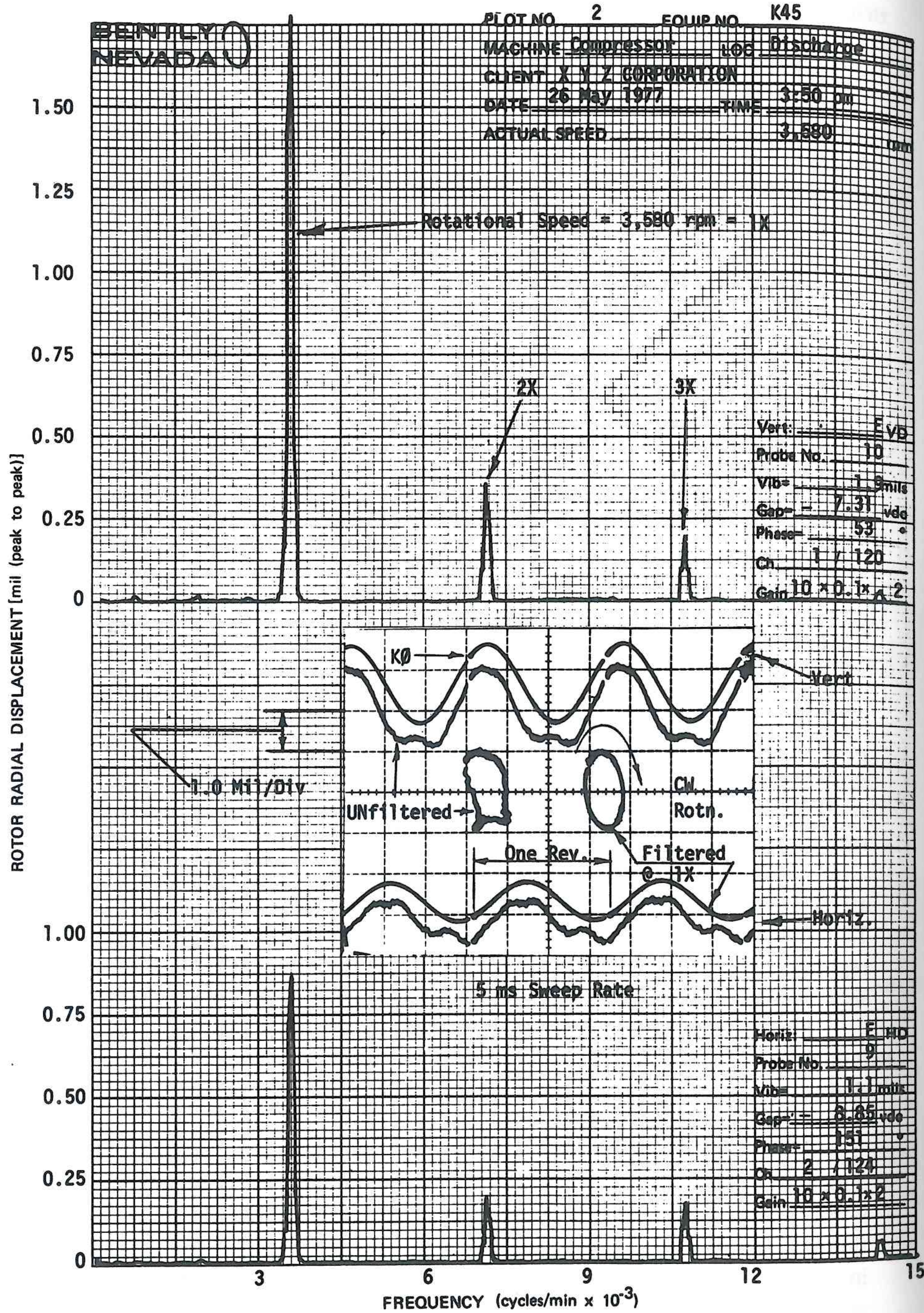


Figure 11.24 Shaft displacement data. (Courtesy of Bently Nevada, Minden, Nevada).

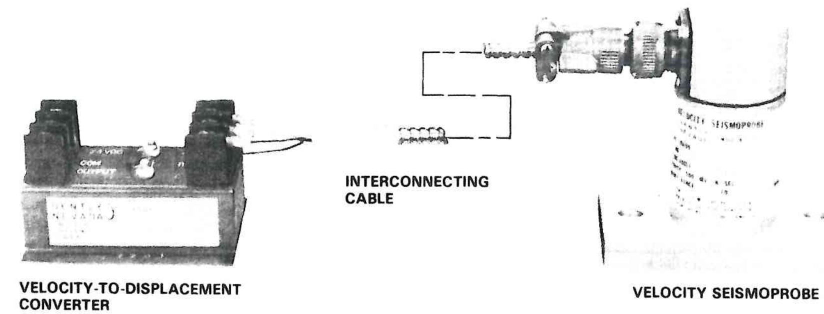


Figure 11.25 Velocity transducer system. (Courtesy of Bently Nevada, Minden, Nev.)

relative motion between the shaft and the mounting of the proximity probe. The probe is usually mounted rigidly to a bearing cap or the gear casing. In order to establish motion of the casing itself, velocity pickups are attached to the casing.

Figure 11.25 illustrates a velocity transducer system. The purpose is to measure gearbox casing or structural vibration velocity and convert the vibration velocity into an electrical signal that represents the displacement of the casing.

One type of velocity transducer works on the principle that as a magnet moves with respect to a relatively stationary coil, a current is induced in the coil. The magnet is rigidly attached to the pickup case and therefore vibrates along with the casing. The coil is suspended by sensitive springs inside the pickup case.

Figure 11.26 depicts the casing velocity characteristics emitted by a three-stage speed-increasing gearbox. The time-domain signal inset in the figure is quite complicated due to the transducer responding to a large number of excitations. A spectrum analysis of the signal reveals Fourier components that can be related back to the operational characteristics of the machine. Peaks can be seen at the rotational speeds of each gear plus the second harmonics of each fundamental frequency. As a rule of thumb, peak vibration velocity readings over approximately 0.5 in./sec signify extremely rough operation and warrant shutdown of the machinery. Readings below 0.1 in./sec indicate smooth, well-balanced, and well-aligned equipment. A reading of over 0.3 in./sec might be a good level at which to consider taking corrective action.

Casing vibration should be measured in the vertical, horizontal, and axial directions. The measurements should be taken on a rigid section on the housing. It should be noted that a measurement taken on a rigid casing can identify such things as structural or piping resonance, loose or cracked foundations, or external vibration input sources but may not transfer vibration amplitudes due

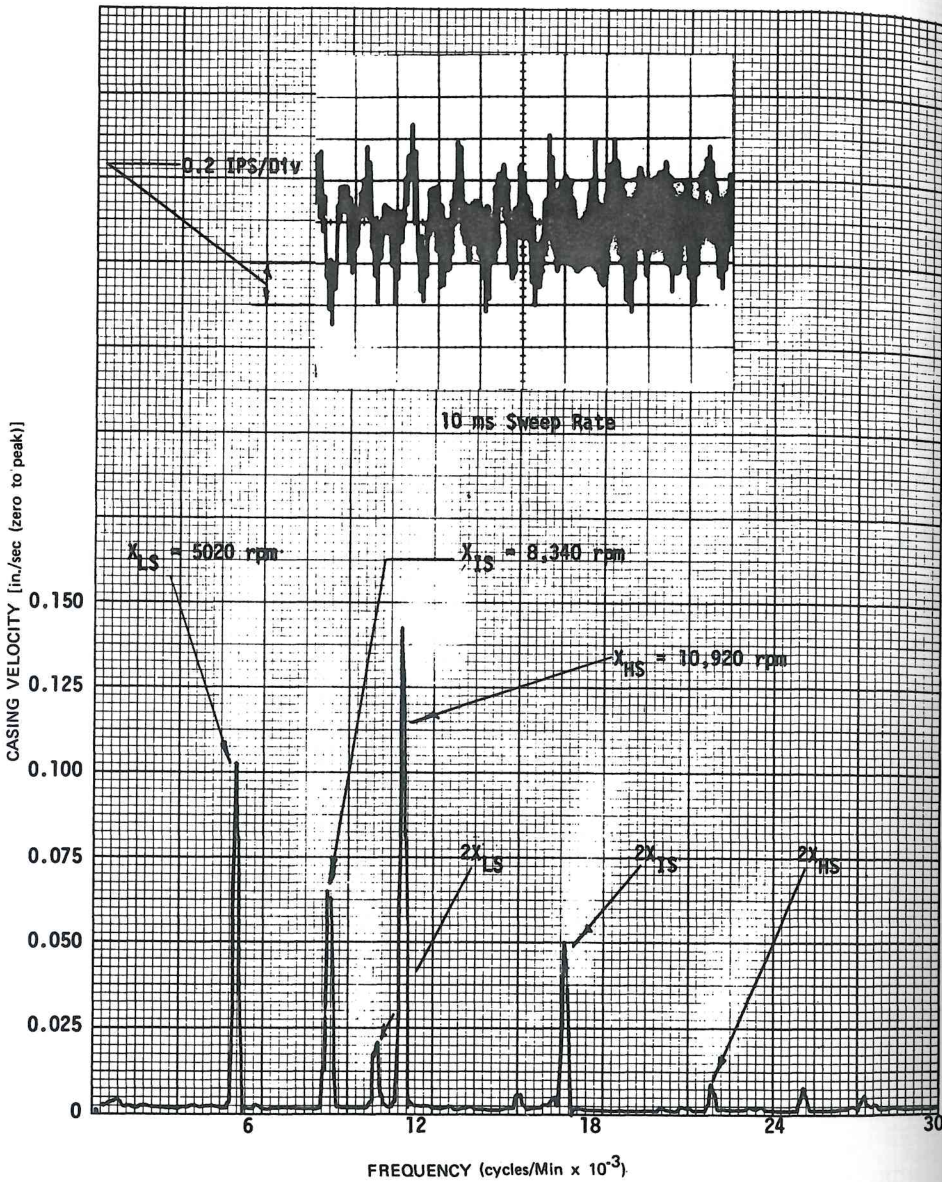


Figure 11.26 Gear box casing velocity characteristics. From Ref. 6; courtesy of Bently Nevada, Minden, Nev.)

Table 11.4 Vibration Conversion Factors

	Multiply the following by the numerical values:			
To obtain	Average	Rms	Peak	Peak to peak
Average	1.0	0.900	0.636	0.318
Rms	1.111	1.0	0.707	0.354
Peak	1.571	1.414	1.0	0.500
Peak to peak	3.142	2.828	2.0	1.0

to shaft motions. The casing may be too stiff to move as a result of shaft motion; therefore, on critical applications shaft position sensors should be incorporated in addition to casing pickups. When a casing transducer is located in the same plane as a shaft proximity probe, a vector summation of the two outputs can give the absolute shaft motion in addition to the shaft motion relative to the casing.

On occasion, rather than defining vibration in peak values, rms or average values are required. Table 11.4 presents the conversion factors. For high-frequency excitations such as gear meshing frequencies, accelerometers are used as transducers. Figure 11.27 shows an acceleration transducer system. The accelerometer portion of the acceleration transducer system is a "contacting" transducer that is physically attached to the vibrating machine part. The accelerometer uses a piezoelectric crystal situated between the accelerometer

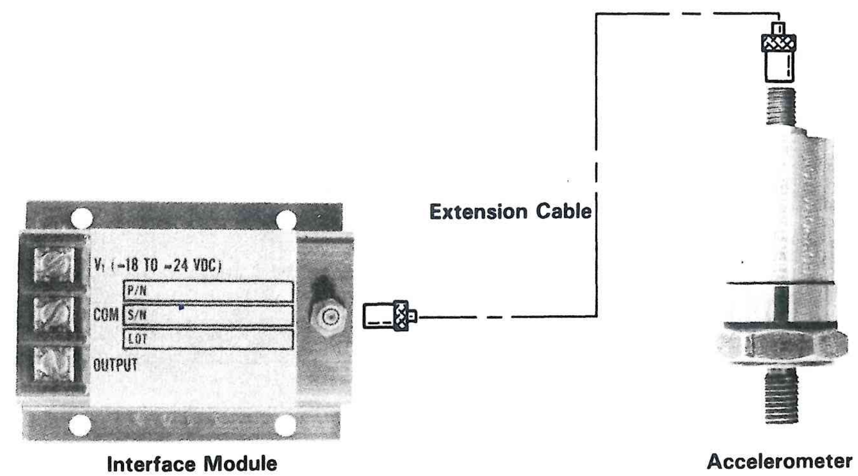


Figure 11.27 Acceleration transducer system. (Courtesy of Bently Nevada, Minden, Nev.)

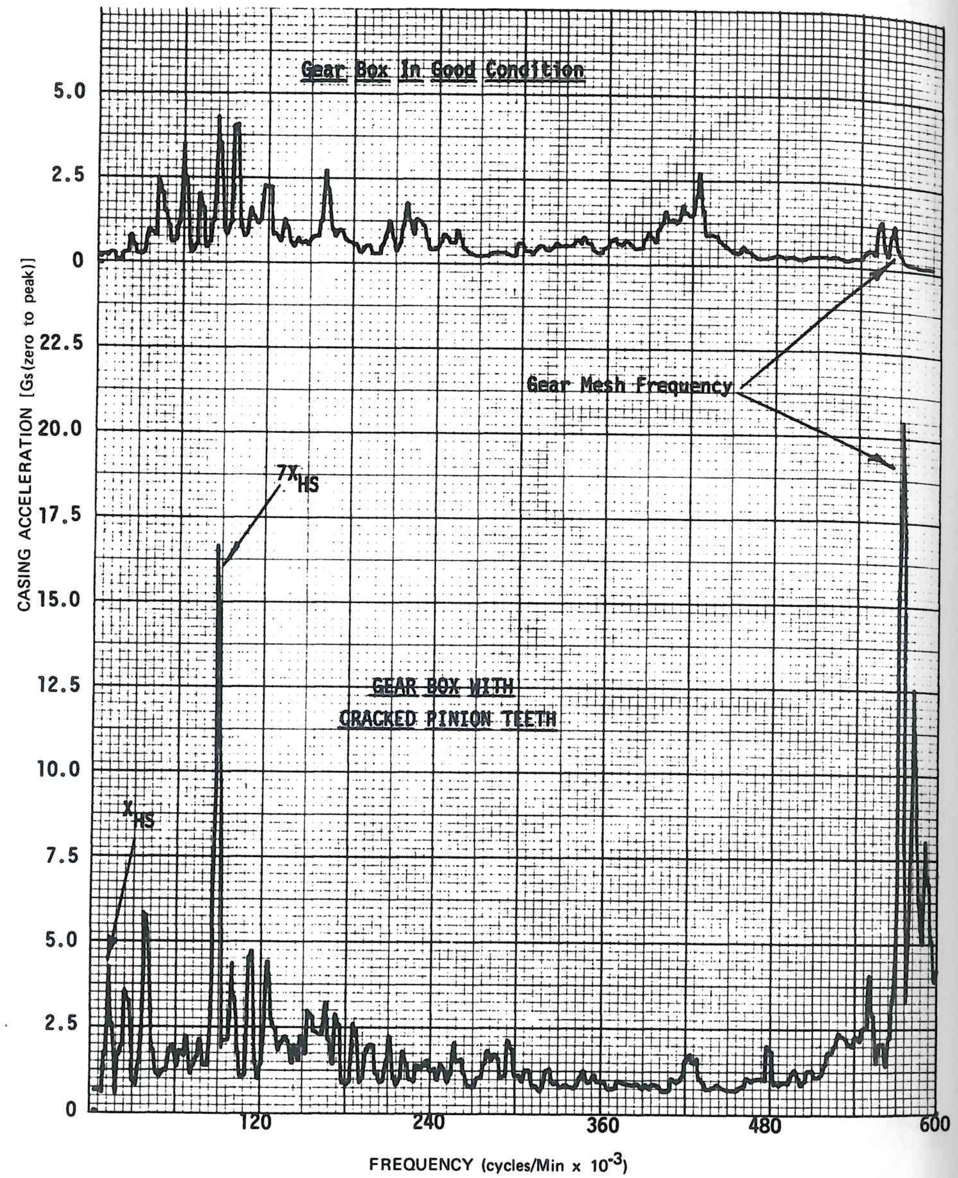


Figure 11.28 Gear box casing acceleration characteristics. (From Ref. 6; courtesy of Bently Nevada, Minden, Nev.)

base and an inertial reference mass. When the crystal is strained (compression or tension force), a displaced electric charge is accumulated on the opposing major surfaces of the crystal. The crystal element performs a dual function. It acts as a precision spring to oppose the compression or tension force and it supplies an electric signal proportional to the applied force.

Frequencies up to 30 kHz can be handled by accelerometers. Figure 11.28 illustrates gearbox casing acceleration characteristics. The top spectrum plot depicts a gearbox in good mechanical condition with reasonably low acceleration levels and a normal mixture of components. A similar measurement made on a unit that had cracked pinion teeth is presented on the bottom plot. High G loadings are exhibited at the gear mesh frequency, the seventh harmonic of pinion rotational speed, and the pinion running speed, X_{HS}.

Figure 11.29 Magnetic chip detector. (Courtesy of Technical Development Company, Glenolden, Pa.)

Figure 11.30 Chip collector with wear debris. (Courtesy of Technical Development Company, Glenolden, Pa.)

Chip Collection

A typical magnetic chip collector is shown in Figure 11.29. Figure 11.30 is a photograph of a chip collector with typical wear debris. The chip collector is installed in the oil sump or a scavenge return line. Figure 11.31 shows typical installations. A self-sealing valve allows withdrawal of the magnetic probe and visual inspection of the collected debris with loss of only a few drops of oil.

A system can be initiated where the chip collector is periodically inspected, every 25 to 50 hr, and records kept characterizing the particles collected. Once a baseline is defined, any change in the debris collected during a

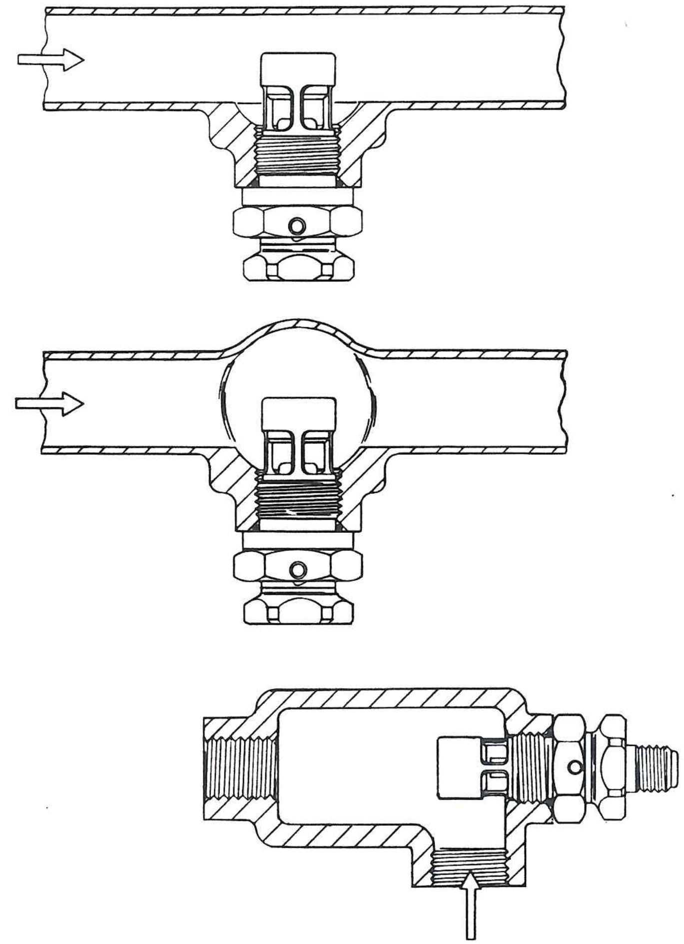


Figure 11.31 Chip collector installations. (Courtesy of Technical Development Company, Glenolden, Pa.)

given interval, such as rate of collection or size of particle, identifies a potential problem. The particles can be analyzed by electron spectroscopy to determine what elements are present and yield clues as to which components are deteriorating [7].

Because the particles trapped are conductors the magnetic probe can be designed such that a gap is bridged between two electrodes when sufficient particles land on the sensor. When this occurs a warning light can be activated. Figure 11.32 shows two such electric chip detectors with radial and axial gaps.

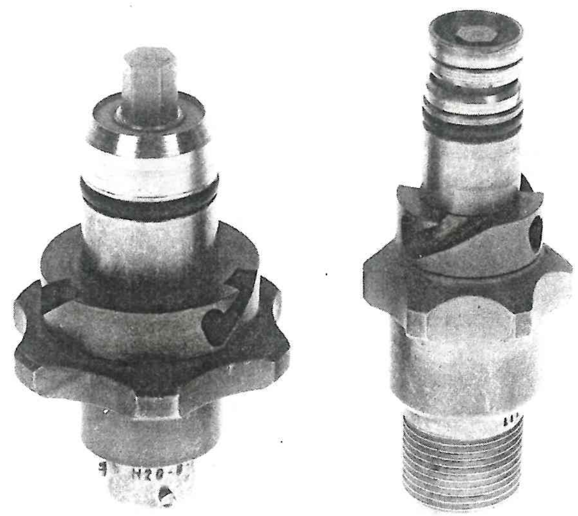


Figure 11.32 Electric chip detector. (Courtesy of Technical Development Company, Glenolden, Pa.)

This type of detector is widely used in aircraft and turbine engine transmissions. Some development of the proper gap size is required since too small a gap will lead to many "nuisance" indications and too large a gap will be insensitive to failure indications. Gaps used are on the order of 0.050 to 0.150 in. With chip collectors or detectors care must be taken to locate them in an area where they will be exposed to as many particles as possible. Also, the area should be relatively still, so the particles can settle.

In forced-flow lubrication systems where particles tend to be dragged along with the oil system, full-flow debris detectors are sometimes used. These configurations have screens through which the total scavenge oil flows. The screens retain the debris. A chip detector can be incorporated for indication purposes. Figure 11.33 shows a full-flow debris detector. An advantage of this type of monitor is that nonmagnetic as well as magnetic debris will be trapped.

Another type of full-flow monitoring device is the indicating screen (Figure 11.34). It is woven from wire strands and when a conductive particle bridges the gap between wires, a warning light can be activated. This device can be used as a pump inlet screen. The indicating screen is sensitive to magnetic and nonmagnetic debris but is more difficult to remove and inspect than a chip detector.

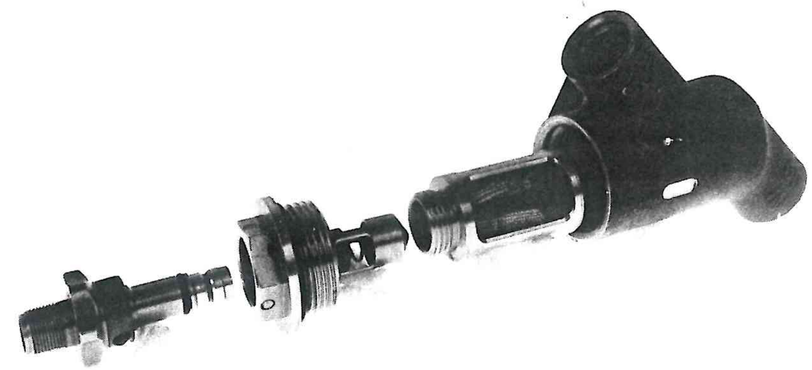


Figure 11.33 Full-flow debris collector. (Courtesy of Technical Development Company, Glenolden, Pa.)

Oil Pressure and Temperature

Oil pressure and temperature measurements are relatively easy to accomplish and yield significant information concerning the operating condition of a gear unit.

The oil pressure should be monitored at the entrance to the unit downstream of any pressure regulation device. Low oil pressure can indicate internal leakage such as a cut static seal or possibly low flow due to pump distress. This can lead to oil starvation of components; therefore, low oil pressure must be

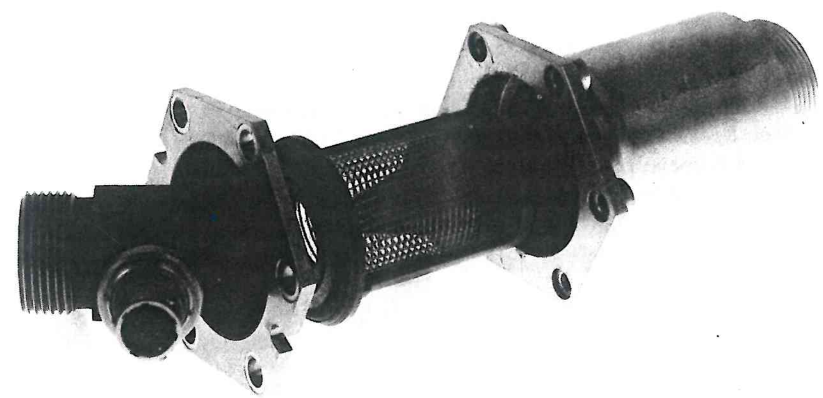


Figure 11.34 Indicating screen debris monitor. (Courtesy of Technical Development Company, Glenolden, Pa.)

investigated and corrected. High oil in pressure can be an indication of downstream blockage such as a clogged oil jet and therefore must also be investigated and corrected. A fluctuating oil pressure can be an indication of pump cavitation due to poor inlet conditions such as excessive pressure drop or air leakage in the pump inlet line. When setting oil pressure limits it must be remembered that the pressure will be affected by variables such as oil temperature and viscosity and jet size variation downstream of the pressure measurement. Pressure in the gearbox cavity will also directly affect the feed pressure.

Oil temperatures should be monitored at the gearbox inlet and outlet at a minimum. The change in temperature across the unit is a good indication of gearbox condition. Scavenge oil temperature of individual bearings can be monitored. On journal bearings, temperature sensors can be embedded under the surface of the babbitt at a depth of 0.030 to 0.060 in. This is a very positive way of monitoring bearing condition. More than one sensor should be installed in case of sensor failure.

Two types of temperature sensors are available: the resistance temperature detector (RTD) and the thermocouple. The RTD is basically a precision resistor where the resistance changes linearly with temperature. RTDs are superior to thermocouples in terms of accuracy, stability, and interchangeability.

REFERENCES

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12

MAINTENANCE AND FAILURE ANALYSIS: SCHEDULED MAINTENANCE ACTIONS

The objective of a maintenance program is to ensure satisfactory gearbox performance at all times and to maintain the transmission in a state of readiness if it is not in operation. A program should be planned which includes regular maintenance actions and periodic monitoring of operating characteristics to determine whether any problems are developing.

Every gearbox should come with a set of maintenance instructions developed by the manufacturer. These instructions should include the following information:

General description of the equipment
Specifications as to speed and power ratings, lubrication, dimensions, and so on
Installation information
Lubrication instructions
Maintenance requirements
Assembly and disassembly instructions
Drawing and parts list

The following section presents some general comments on maintenance of gearboxes. A specific unit may require special maintenance procedures and the manufacturer's instructions should be adhered to.

There are several maintenance actions that should be affected during the initial operation of a gearbox. After approximately 50 hr of running time, the following should be accomplished:

Check coupling alignment and correct if necessary.
Check bolt torques and retighten if necessary.

Table 12.1 Regular Gearbox Maintenance Actions

Frequency	Maintenance item	Corrective action
Daily	Check oil temperature and pressure at operating conditions.	If there is a drastic change from previous readings, stop unit and determine cause.
	Check for noise, vibration, and oil leaks.	
	Check sump oil level.	Add oil if necessary.
Weekly	Check oil filter.	Change filter element if necessary.
Monthly	Check lubricating oil for contamination.	Drain and refill lube system if necessary. Change oil filter.
	Check all gages, controls, and alarm systems.	
	Clean breather element.	
Every 2500[a] hours or 6 months	Change lubricating system oil.	

[a]If operating conditions are unusually severe, such as high-temperature or high-moisture atmospheres, oil change requirements might be more frequent. Changes can be based on inspection of the oil for viscosity or acid number in such cases.
Source: *Sawyers Turbomachinery Maintenance Handbook*, 1st ed., Vol. III, Turbomachinery International Publications, Norwalk, Conn., 1980.

Check piping connections and retighten if necessary.
Change oil and clean sump.

The oil, after 50 hr, need not be discarded but can be drained, filtered through an element with a micrometer rating no greater than the gearbox filter, and reused. Particles may be found in the oil and the sump due to normal wearing in processes. At this point the sump or reservoir should be thoroughly cleaned. After draining the original oil it is recommended that the gearbox and lubrication system be flushed out with a flushing oil. If possible, bring the unit up to operating speed at light load after filling with flushing oil. Shut down immediately after achieving full speed. The drain the flushing oil and refill with the recommended lubricant to the proper level.

After the 50 hr maintenance a regular program should be followed as outlined in Table 12.1. Logs should be kept of instrument readings and maintenance actions to keep a running account of gearbox condition.

When performing maintenance operations every precaution must be taken to prevent foreign matter from entering the gearbox. The introduction of moisture, dirt, or fumes can lead to sludge formation and deterioration of the lubrication and cooling system.

STORAGE

Quite often the gearbox is delivered before the complete system is ready for assembly and it must be stored for some period of time prior to operation. When operation is delayed more than 1 month after shipment, special precautions must be taken to prevent rusting of the components. If possible, the gearbox should be completely filled with oil during storage. Where this is not practical, all exposed metal parts, both inside and outside the unit, should be sprayed with a heavy-duty rust preventative. The gearbox should be stored in a dry area remaining at approximately constant temperature, preferably indoors. If stored outdoors, the gearbox should be raised off the ground and completely enclosed by a protective covering such as a tarpaulin. If possible, the unit should be rotated at weekly intervals while in storage.

OVERHAUL AND SPARE PARTS

Generally, gearboxes do not have a specific time period after which the unit is disassembled and overhauled. It is more common to observe deterioration of components such as bearings and gears during operation and replace the particular component at a convenient time. Usually, the gearbox is delivered with an operating and maintenance manual which describes how to disassemble and assemble the unit. If the user is not completely familiar with the equipment, it would be prudent to have a factory representative accomplish any major component replacements. Spare gears or bearings for gearboxes are not necessarily readily available from the manufacturers. Journal bearings, unlike antifriction bearings, are not usually stocked by distributors. In many cases the bearings are customized for the specific gearbox and therefore are even harder to replace. Finished gears are rarely stocked by manufacturers, and lead times on gears or bearings might be 20 weeks or more. When purchasing the drive the user should request a recommended spares list and determine what the availability of these parts will be. The user and manufacturer can then arrive at some agreement over what spares will be available and where they will be stored.

TROUBLESHOOTING

The major causes of gearbox failure are improper lubrication and overload. Care must be taken to check for proper oil level before operation. Excessive oil

Table 12.2 Gearbox Troubleshooting Chart

Problem	Recommended inspection	Corrective action
Overheating	1. Oil cooler operation	Check flow of coolant and oil flow. Measure oil temperature into and out of cooler. Check cooler internally for buildup of deposits from coolant water.
	2. Is oil level too low or too high?	Check oil level indicator.
	3. Bearing installation	Make sure that bearings are not pinched and properly adjusted.
	4. Grade and condition of oil	Check that oil is specified grade. Inspect oil to see if it is oxidized, dirty, or with high sludge content.
	5. Lubrication system	Check operation of oil pump. Make sure that suction side is not sucking air. Measure flow. Check if oil passages are free. Inspect oil line pressure regulator, nozzles, and filters to be sure they are free of obstruction.
	6. Coupling float and alignment	Check coupling alignment and adjust end float.
Shaft failure	1. Type of coupling	Rigid couplings between rigidly supported shafts can cause shaft failure. Replace with coupling to provide required flexibility and lateral float.
	2. Coupling alignment	Realign equipment as required.
	3. Excessive overhung load	Reduce overhung load. Use outboard bearing or replace with higher capacity unit.
	4. High transient loading	Apply couplings capable of absorbing shocks. Use couplings with shear pins.

Table 12.2 (Continued)

Problem	Recommended inspection	Corrective action
Shaft failure (continued)	5. Torsional or lateral vibrations	Adjust system mass elastic characteristics to control critical speed location. Possibly, coupling geometry can be modified.
	6. Cracks due to fretting corrosions	Note cause of fretting and correct. Press fits between gear and shaft.
Oil leakage	1. Exceed oil level	Check oil level indicator.
	2. Is breather open	Check oil breather.
	3. Are oil drains open	Check that all oil drain locations are free and clean.
	4. Oil seals	Check oil seals and replace if worn. Check condition of shaft under seal and polish if necessary.
	5. Plugs at drains, levels, and pipe fittings	Apply sealant and tighten fittings.
	6. Housings and caps	Tighten cap screws or bolts. If not effective, remove housing cover and caps. Clean mating surfaces and apply new sealing compound. Reassemble. Check compression joints by tightening fasteners firmly.

Source: *Sawyers Turbomachinery Maintenance Handbook*, 1st ed., Vol. III, Turbomachinery International Publications, Norwalk, Conn., 1980.

volume can be as detrimental as lack of lubrication and will result in churning and overtemperature of components. Overload can be a result of vibration, shock loads, or high torque at low speed. If there is a possibility that operating loads will exceed rated gearbox loads, the manufacturer should be consulted. Table 12.2 is a troubleshooting guide that gives some guidance in how to identify and correct some of the common problem areas occurring during gearbox operation.

Gear teeth should be inspected for nicks, burrs, and scratches, which may be repaired by blending provided that they are minor and not on the working surfaces of a tooth. The blend may be accomplished using a small file and an India or carborundum stone. Crocus cloth should be used for the final polishing. All repairs must be finished smoothly. Power tools are not permitted for blend repairs.

In many gearboxes the teeth can be visually inspected by removing inspection covers bolted into the casing. When opening these inspection covers care must be taken to ensure that no foreign material enters the gearbox. Gear teeth should be examined under good lighting and be wiped clean of oil to prevent a false diagnosis. The content pattern should cover approximately 80% of the tooth. The following section discusses gear tooth failure modes and describes various conditions that may be encountered in the field.

Gear Tooth Failure Modes

The major modes of gear tooth failure are breakage, wear, pitting, and scoring. When these problems are encountered in the field it is important to accurately define the condition and causes of failure to be able to determine corrective actions for the particular units in difficulty and also to modify analytical and manufacturing methods for future gearboxes so that they will not suffer the same problems. Also, an accurate diagnosis of field problems will enable the user to determine if a gearbox requires immediate modification or replacement or if it can be expected to continue to function for some period of time, when repairs can be made more conveniently.

Breakage

Breakage of gear teeth is the most catastrophic form of failure. It occurs precipitously with no advance warning. If a number of teeth break, load transmission is no longer possible. If only one tooth or a portion of a tooth breaks, there is the possibility of secondary damage if the broken part interferes with other components in the system. Also, dynamic loading will increase and if operation continues, other teeth will soon fail. Breakage of gear teeth is caused by excessive bending stress in the root imposed by the transmitted load. Tooth breakage can be the result of a fatigue mechanism (Figure 12.1) or an overload which exceeds the gear tooth fracture strength (Figure 12.2). A fatigue break initiates as a small crack which, over a large number of load cycles, propagates until a portion of, or a whole tooth, separates from the gear. Failures of this nature may be a result of system overloads greater than the design load, such as torsional vibrations. Manufacturing discrepancies such as tool marks or metallurgical discrepancies can also lead to fatigue failures by initiating cracks. In surface-hardened gears, if the case in the root is not correct either due to heat-treat problems or excessive machining, bending failures can result.

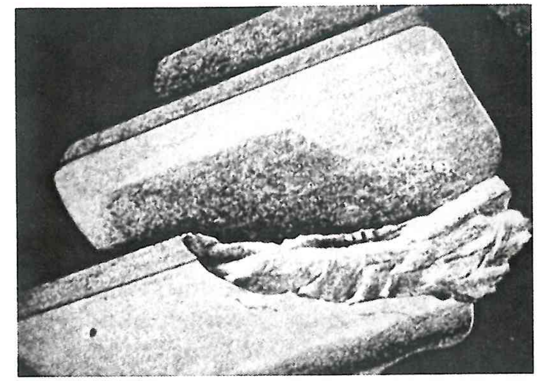


Figure 12.1 Fatigue breakage. (Courtesy of American Gear Manufacturers Association, Arlington, Va.)

As shown in Figure 12.1, the smooth appearance of the fracture surface attests to the fact that considerable working of the cracked surfaces occurred prior to final separation. Overload breakage occurs in relatively few cycles; therefore, the fracture surfaces are rougher than those of a high-cycle fatigue failure. Some causes of overload breakage are:

Large particles passing through the mesh
Sudden misalignments such as when a coupling fails
Bearing seizures
Shock loads such as short circuits in a generator drive

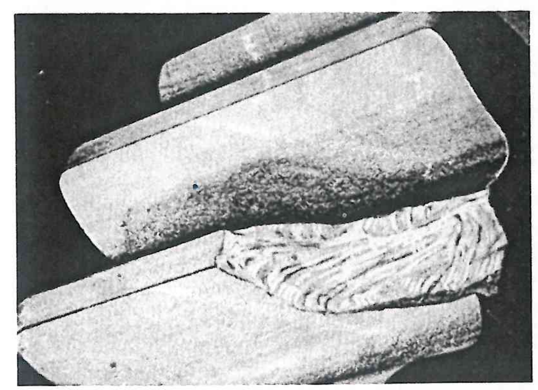


Figure 12.2 Overload breakage. (Courtesy of American Gear Manufacturers Association, Arlington, Va.)

In some cases other failure modes such as wear or pitting may increase dynamic loading or weaken the tooth to the extent that breakage ultimately occurs.

Wear

Wear may be defined as the loss of metal due to the rubbing action of two surfaces moving in relation to one another when the oil film is not of sufficient thickness to separate them [1]. One form of rubbing wear is adhesive wear characterized by metal particles from one gear tooth adhering to the mating gear tooth by a welding action and subsequently detaching. The other form of rubbing wear is abrasive wear caused by abrasive action between the sliding gear teeth or by the presence of abrasive particles between them. These particles may be contaminants in the oil or pieces detached from the tooth surfaces themselves. Figure 12.3 illustrates some worn areas on a helical gear.

Gear teeth do not necessarily wear during operation. If the oil film thickness is sufficient to separate the mating tooth surfaces, millions of cycles can be accumulated with no measurable wear. In some cases there may be an initial wearing in of gear teeth and if rubbing wear diminishes with time it may not be detrimental; therefore, when wear is first noted the gearset should be closely monitored to determine the rate at which wear progresses such that a determination can be made as to whether it will be damaging. The wear-in phenomenon is more common with through-hardened gears (Rc 32 approximately) than surface-hardened gearing (approximately Rc 60).

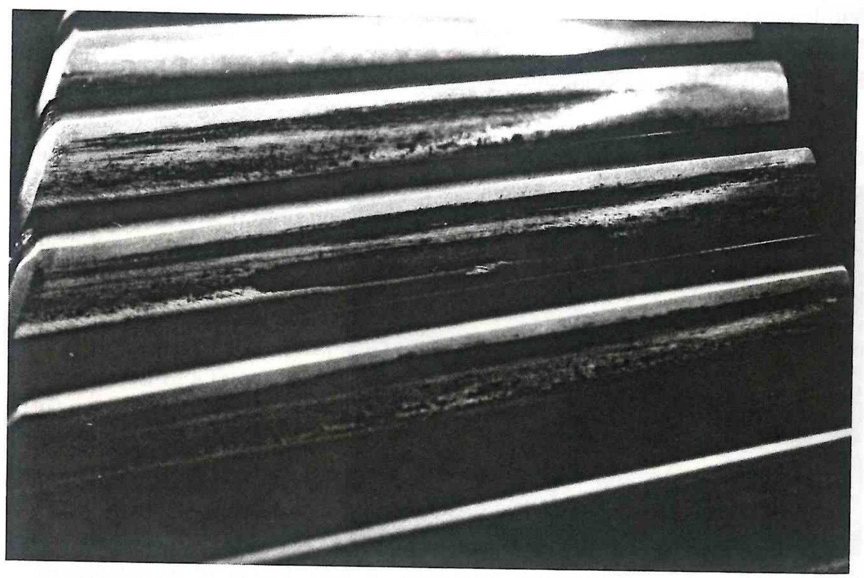


Figure 12.3 Tooth wear.

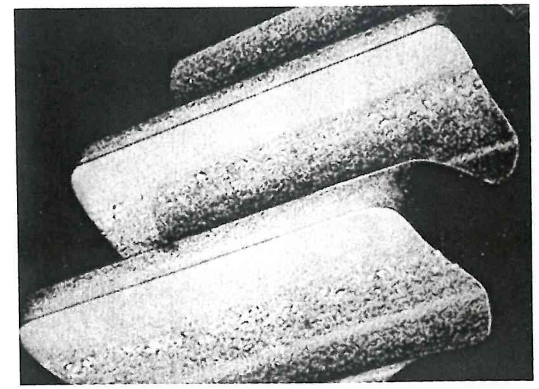


Figure 12.4 Initial pitting. (Courtesy of American Gear Manufacturers Association, Arlington, Va.)

Pitting

Pitting manifests itself in several forms. Initial pitting, sometimes called frosting, may occur during early operation. This is a surface-oriented failure mode where local high spots contact and exhibit distress, as shown in Figure 12.4. As the high spots are removed, the load is more evenly distributed and the pitting action diminishes. This condition may be acceptable.

Destructive fatigue pitting is a result of repeated stress cycling of the tooth surface beyond the material's endurance limit. Surface or subsurface cracks initiate, propagate, and eventually material detaches from the tooth surface, leaving pitted areas (Figure 12.5). Pitting may progress to a point where large areas are broken out, as shown in Figure 12.6. This condition is referred to as spalling.

Scoring

Scoring is a form of surface damage on the tooth flank which occurs when overheating causes the lubricant film to become unstable, allowing metal-to-metal contact. Local welding is initiated and the welded junctions are torn apart by the relative motion of the gear teeth, resulting in radial score marks. Figures 12.7 to 12.9 illustrate degrees of the scoring phenomenon. Light scoring which does not progress may be acceptable, but heavier scoring can destroy the tooth profile and lead to pitting and breakage. The scoring type of failure mode generally occurs in high-speed applications using low-viscosity lubricants.

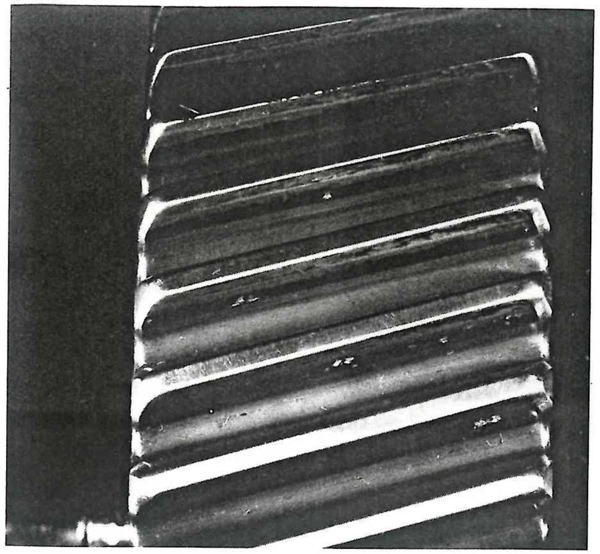


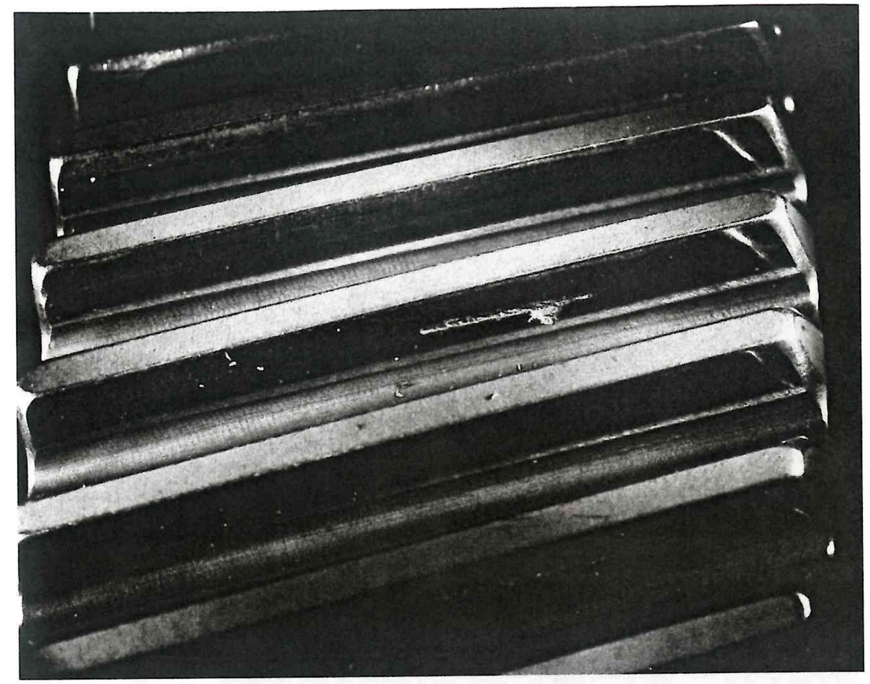
Figure 12.5 Pitting.

Evaluation of Surface Distress

Tooth breakage is the only failure mode that demands immediate action be taken. Wear, pitting, and scoring are progressive and may occur over a long period of time without significantly impairing the function of the unit. As an example, Figures 12.10 to 12.16 show the progressive deterioration of a gear during 1000 hr of operation. At 50 hr there is a band of wear in the dedendum of the teeth with a depth of 0.0001 in. at the right hand end increasing to 0.0006 in. at the more heavily worn left-hand end. Some pitting is also noticeable. At 120 hr the wear has spread toward the tip of the tooth. More pits have appeared. At 200 hr the tooth tip has worn further, to a depth of approximately 0.0006 in. At 400 hr the wear band has spread to the right-hand end. During this segment of testing the right-hand end wore 0.0006 in. From 400 to 1000 hr there were no major changes in gear tooth condition.

The purpose of this discussion is not to advocate running gears in a worn and pitted condition. The system noise and vibration level will increase as the tooth surface deteriorates and abrasive particles are being introduced into the lubrication system. There is always the danger of catastrophic breakage of a weakened tooth. It can be seen, however, that even when the tooth surface is severely deteriorated, gear teeth are capable of operating.

As a general guideline, if surface distress is noted early in operation and it is light in nature, time can be taken to observe the progression and if distress



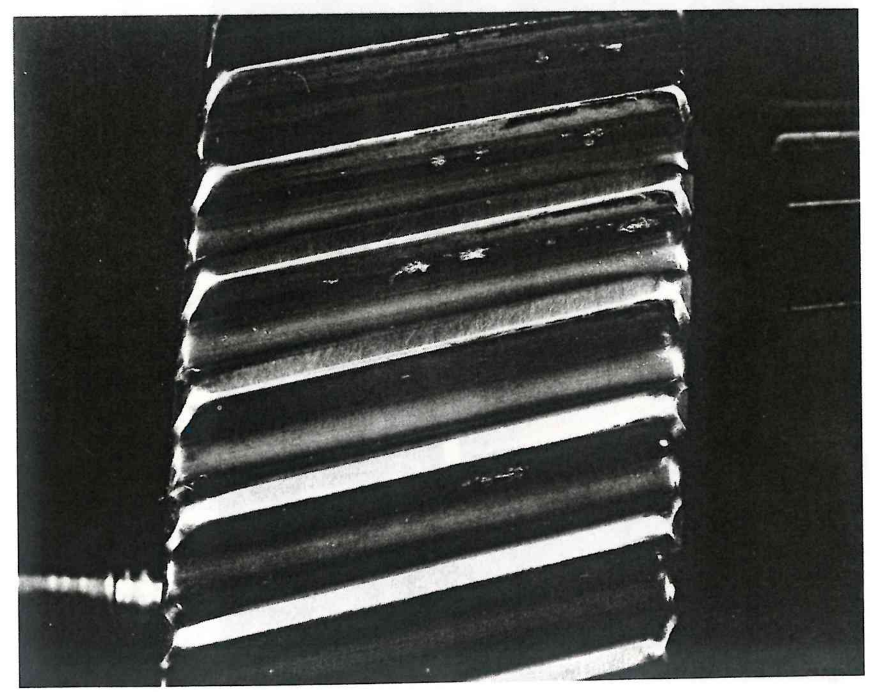


Figure 12.6 Spalling.

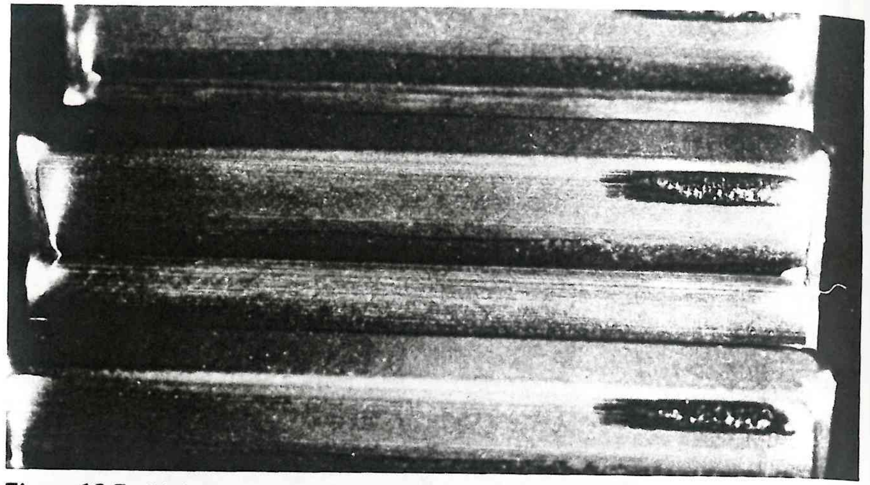


Figure 12.7 Light scoring.

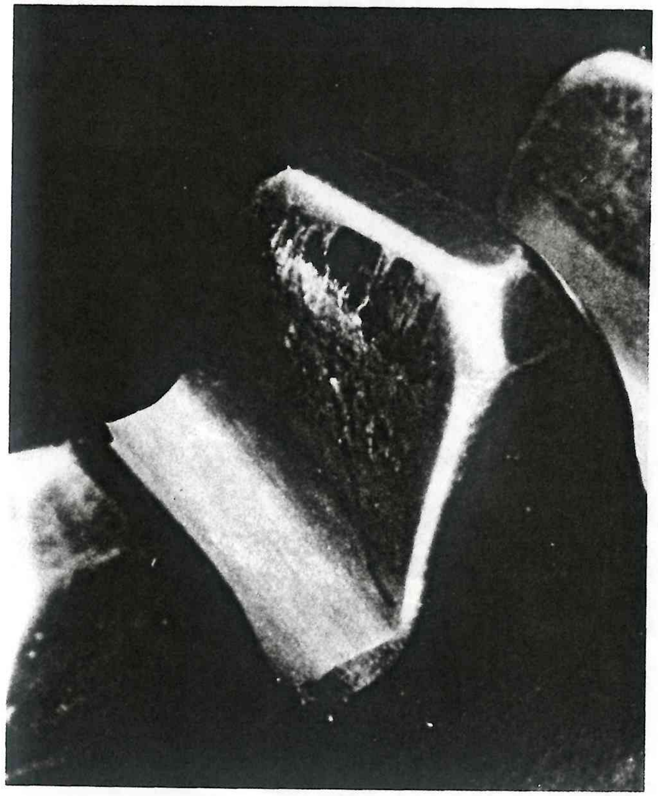


Figure 12.8 Medium scoring.

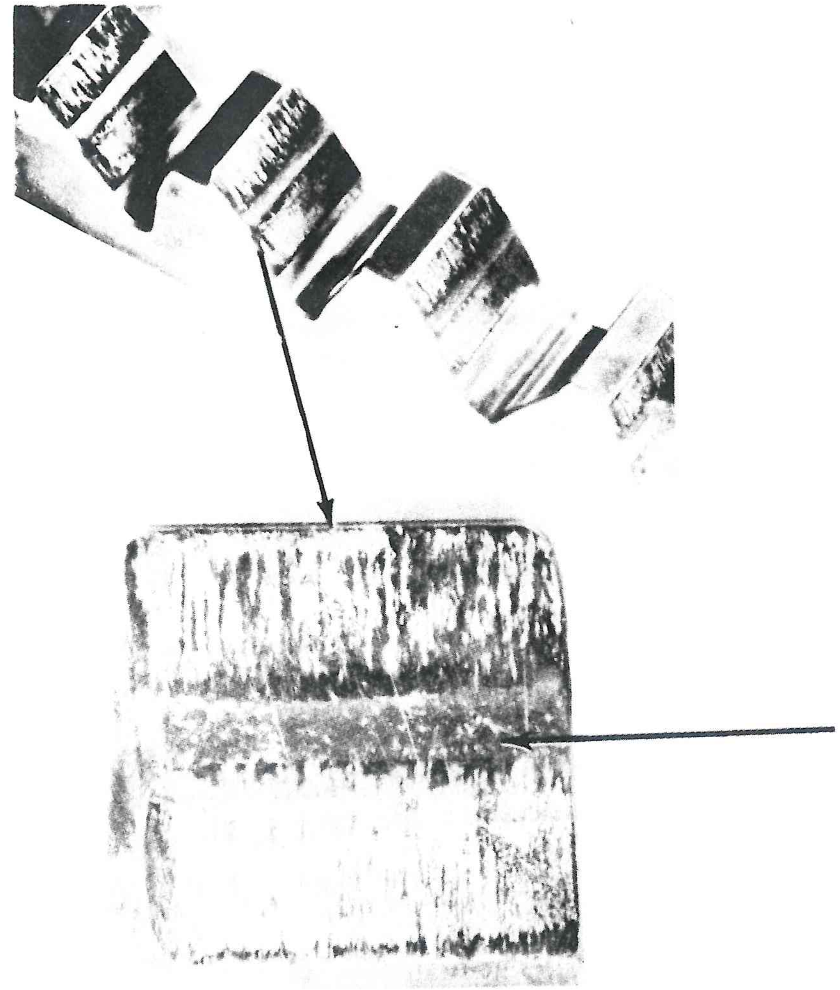


Figure 12.9 Heavy scoring.

ceases to increase, the gearset is probably satisfactory for continued operation. If distress occurs after long operation, it is probably an indication that something has changed in the operating conditions. Possibly loads have increased or the lubricant has deteriorated. When surface distress appears there are several potential changes in operating procedures that can be made to prolong gear life:

It may be possible to reduce the load on the gear teeth. Sources of external loading such as coupling unbalance or misalignment should be checked. If

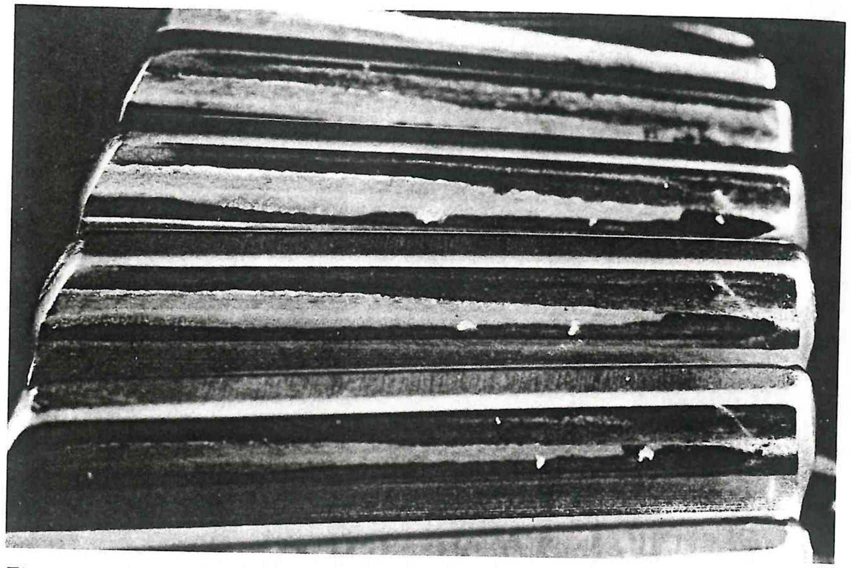


Figure 12.10 Tooth condition after 50 hr.

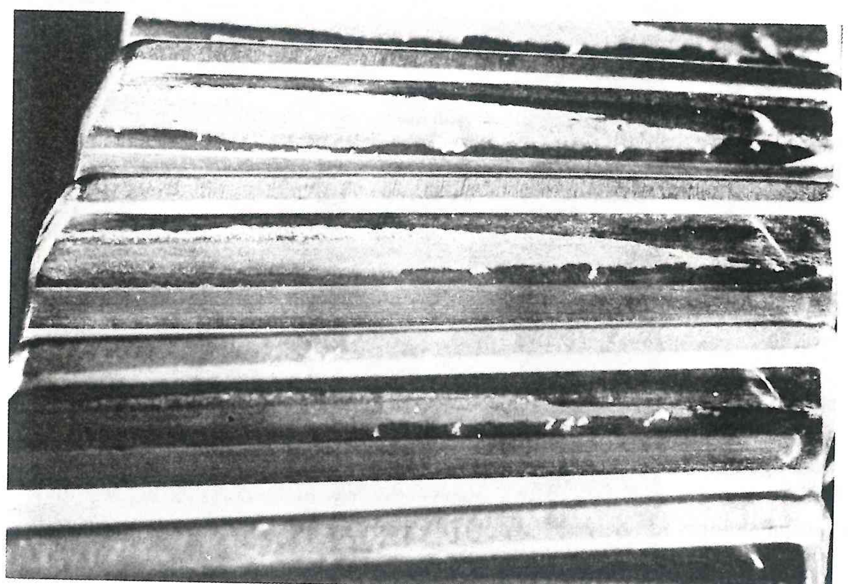


Figure 12.11 Tooth condition after 120 hr.

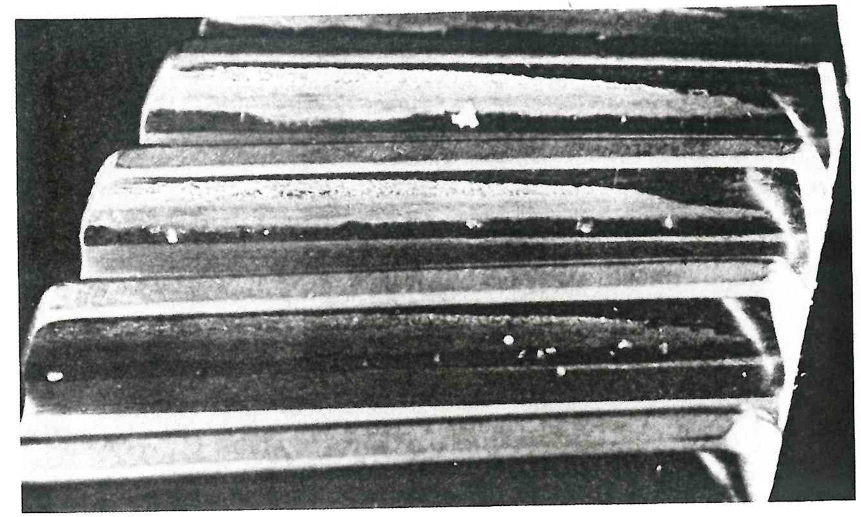


Figure 12.12 Tooth condition after 200 hr.

torsional vibrations are suspected, measurements should be taken. It may be possible to derate the system and operate at lower loads.

An oil with higher load capacity may be available. EP additives can possibly be of value. The manufacturer should be consulted before changing lubricants.

Operating the unit with cooler oil which will be at a higher viscosity creating a thicker film may be beneficial when surface distress is a problem.

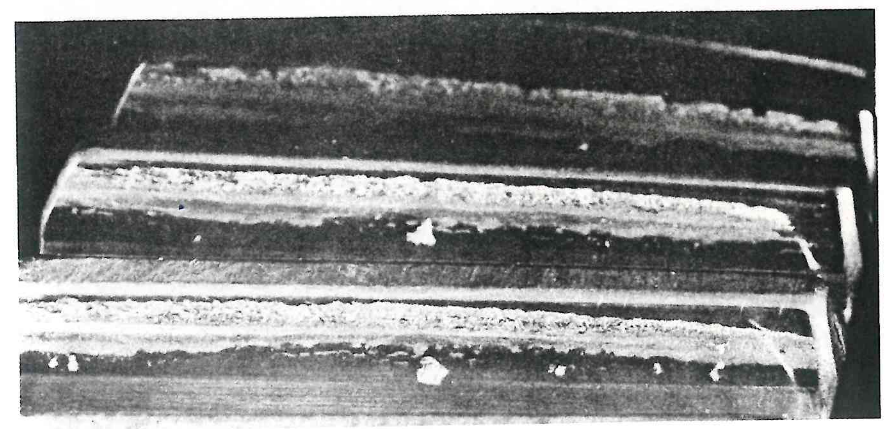


Figure 12.13 Tooth condition after 400 hr.

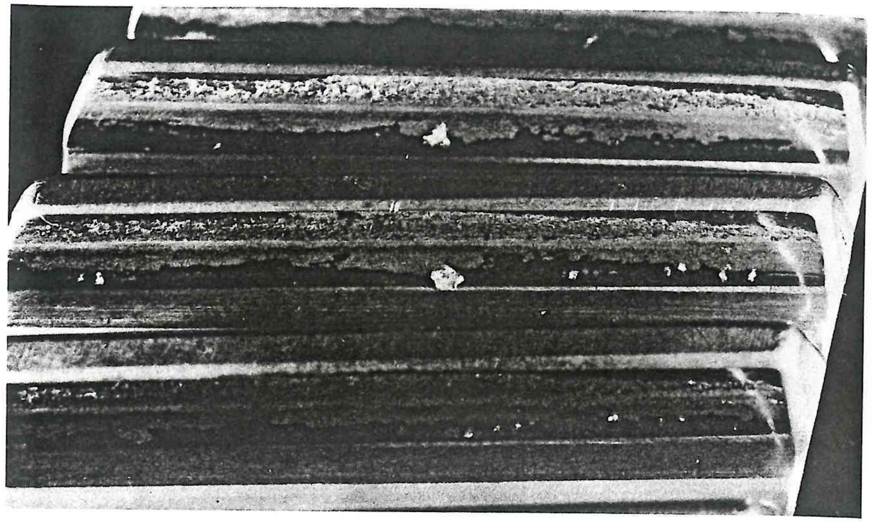


Figure 12.14 Tooth condition after 600 hr.

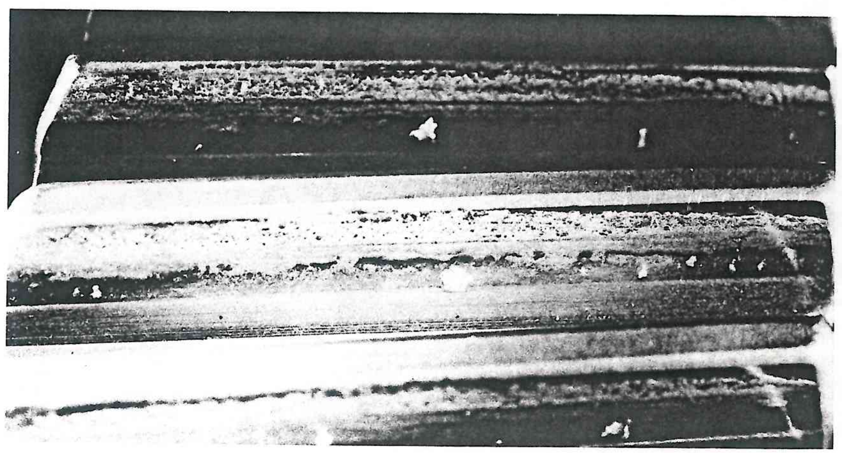


Figure 12.15 Tooth condition after 800 hr.

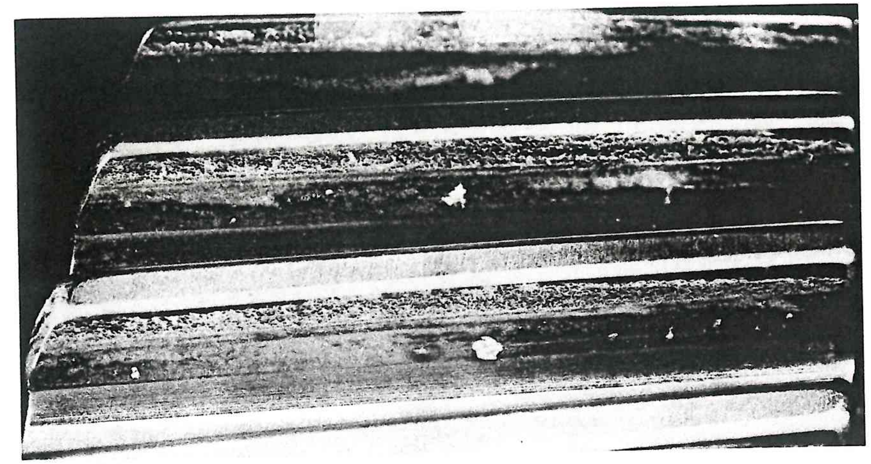


Figure 12.16 Tooth condition after 1000 hr.

Cracking

On occasion, cracks are observed on gear teeth as shown in Figure 12.17. Cracks are usually a result of incorrect heat treatment or abusive grinding. In some cases improper heat treatment results in brittle tooth tips or ends that chip off during operation.

Cracks on the tooth flanks are dangerous since they can propagate and lead to catastrophic failure; therefore, gears exhibiting cracking should be replaced. Chipping of gear teeth at the tips or ends may be repairable by blending. The danger here is that if more chipping occurs, the pieces may interfere with rotating components.

Improper heat treatment of case-hardened gears can result in case-core separation. Cracks originating in the core material propagate along the case-to-core boundary and then work out to the surface. When this occurs large areas of material are removed.

Oil Starvation

Quite a few gearboxes have been lost because the units were run without lubrication. Sometimes when viewing the components after failure it is not obvious what the cause was. For instance, Figure 12.18 shows the condition of components of a planetary gear set after 1.75 min of operation at full load and speed without oil. The smaller sun gear has failed catastrophically, shedding all its teeth. Other gears and antifriction bearings are in reasonably good condition. The sun gear, having the least area to dissipate heat, reached the melting point

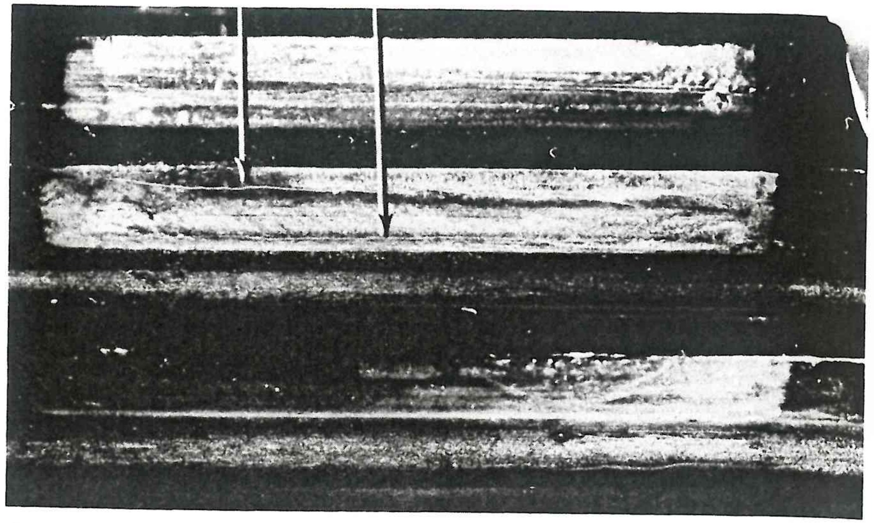


Figure 12.17 Tooth cracking.

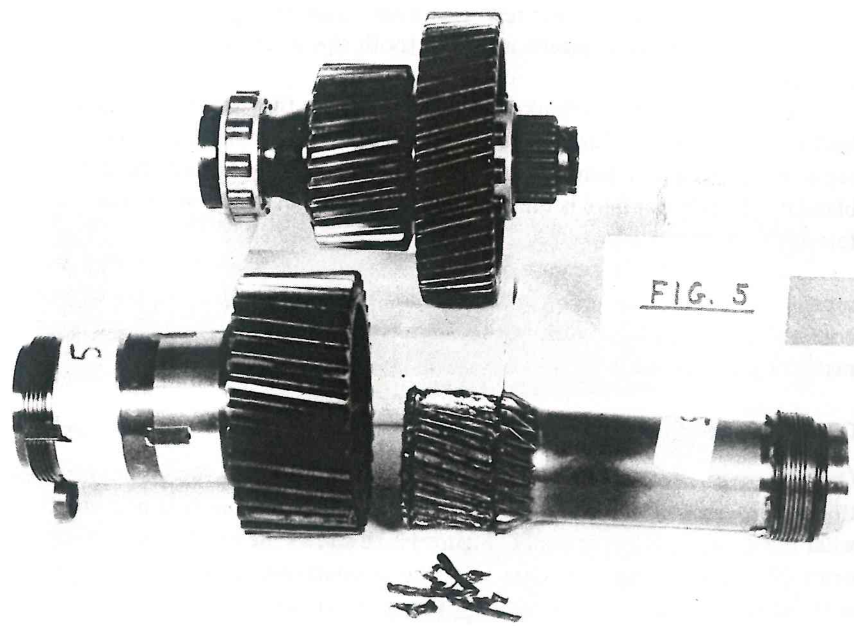


Figure 12.18 Oil starvation failure. (Courtesy of AVCO, Lycoming Corporation, Stratford, Conn.)

of its teeth and suffered the most damage. It was not obvious from the component condition that oil starvation was the cause of failure. With journal bearing gearboxes the diagnosis is easier since the bearings will sieze long before the melting point of gear teeth is reached.

REFERENCE

1. Southwest Research Institue, Gear Tooth Scoring Investigation, Report USAAMRDL-TR-75-33, San Antonio, Tex., July 1975.

APPENDIX

INVOLUTE TABLES (INV $\phi = \tan\phi - \phi$)

Angle (deg)	Cosine	Involute	Angle (deg)	Cosine	Involute
17.0	0.956305	0.009025	19.2	0.944376	0.013134
17.1	0.955793	0.009189	19.3	0.943801	0.013346
17.2	0.955278	0.009355	19.4	0.943223	0.013562
17.3	0.954761	0.009523	19.5	0.942641	0.013779
17.4	0.954240	0.009694	19.6	0.942057	0.013999
17.5	0.953717	0.009866	19.7	0.941471	0.014222
17.6	0.953191	0.010041	19.8	0.940881	0.014447
17.7	0.952661	0.010217	19.9	0.940288	0.014674
17.8	0.952129	0.010396	20.0	0.939693	0.014904
17.9	0.951594	0.010577	20.1	0.939094	0.015137
18.0	0.951057	0.010760	20.2	0.938493	0.015372
18.1	0.950516	0.010946	20.3	0.937889	0.015609
18.2	0.949972	0.011133	20.4	0.937282	0.015850
18.3	0.949425	0.011323	20.5	0.936672	0.016092
18.4	0.948876	0.011515	20.6	0.936060	0.016337
18.5	0.948324	0.011709	20.7	0.935444	0.016585
18.6	0.947768	0.011906	20.8	0.934826	0.016836
18.7	0.947210	0.012105	20.9	0.934204	0.017089
18.8	0.946649	0.012306	21.0	0.933580	0.017345
18.9	0.946085	0.012509	21.1	0.932954	0.017603
19.0	0.945519	0.012715	21.2	0.932324	0.017865
19.1	0.944949	0.012923	21.3	0.931691	0.018129

Angle (deg)	Cosine	Involute	Angle (deg)	Cosine	Involute
21.4	0.931056	0.018395	25.3	0.904083	0.031130
21.5	0.930418	0.018665	25.4	0.903335	0.031521
21.6	0.929776	0.018937	25.5	0.902585	0.031917
21.7	0.929133	0.019212	25.6	0.901833	0.032315
21.8	0.928486	0.019490	25.7	0.901077	0.032718
21.9	0.927836	0.019770	25.8	0.900319	0.033124
22.0	0.927184	0.020054	25.9	0.899558	0.033534
22.1	0.026529	0.020340	26.0	0.898794	0.033947
22.2	0.925871	0.020629	26.1	0.898028	0.034364
22.3	0.925210	0.020921	26.2	0.897258	0.034785
22.4	0.924546	0.021216	26.3	0.896486	0.035209
22.5	0.923880	0.021514	26.4	0.895712	0.035637
22.6	0.923210	0.021815	26.5	0.894934	0.036069
22.7	0.922538	0.022119	26.6	0.894154	0.036505
22.8	0.921863	0.022426	26.7	0.893371	0.036945
22.9	0.921185	0.022736	26.8	0.892586	0.037388
23.0	0.920505	0.023049	26.9	0.891798	0.037835
23.1	0.919821	0.023365	27.0	0.891007	0.038287
23.2	0.919135	0.023684	27.1	0.890213	0.038742
23.3	0.918446	0.024006	27.2	0.889416	0.039201
20.4	0.917755	0.024332	27.3	0.888617	0.039664
20.5	0.917060	0.024660	27.4	0.887815	0.040131
20.6	0.916363	0.024992	27.5	0.887011	0.040602
20.7	0.915663	0.025326	27.6	0.886204	0.041076
20.8	0.914960	0.025664	27.7	0.885394	0.041556
20.9	0.914254	0.026005	27.8	0.884581	0.042039
24.0	0.913545	0.026350	27.9	0.883766	0.042526
24.1	0.912834	0.026697	28.0	0.882948	0.043017
24.2	0.912120	0.027048	28.1	0.882127	0.043513
24.3	0.911403	0.027402	28.2	0.881303	0.044012
24.4	0.910684	0.027760	28.3	0.880477	0.044516
24.5	0.909961	0.028121	28.4	0.879649	0.045024
24.6	0.909236	0.028485	28.5	0.878817	0.045537
24.7	0.908508	0.028852	28.6	0.877983	0.046054
24.8	0.907777	0.029223	28.7	0.877146	0.046575
24.9	0.907044	0.029598	28.8	0.876307	0.047100
25.0	0.906308	0.029975	28.9	0.875465	0.047630
25.1	0.905569	0.030357	29.0	0.874620	0.048164
25.2	0.904827	0.030741	29.1	0.873772	0.048702

Angle (deg)	Cosine	Involute	Angle (deg)	Cosine	Involute
29.2	0.872922	0.049245	33.1	0.838671	0.074188
29.3	0.872069	0.049792	33.2	0.836764	0.074932
29.4	0.871214	0.050344	33.3	0.835807	0.075683
29.5	0.870356	0.050901	33.4	0.834848	0.076439
29.6	0.869495	0.051462	33.5	0.833886	0.077200
29.7	0.868632	0.052027	33.6	0.832921	0.077968
29.8	0.867765	0.052597	33.7	0.831954	0.078741
29.9	0.866897	0.053172	33.8	0.830984	0.079520
30.0	0.866025	0.053751	33.9	0.830012	0.080305
30.1	0.865151	0.054336	34.0	0.829038	0.081097
30.2	0.864275	0.054924	34.1	0.828060	0.081894
30.3	0.863396	0.055518	34.2	0.827081	0.082697
30.4	0.862514	0.056116	34.3	0.826098	0.083506
30.5	0.861629	0.056720	34.4	0.825113	0.084321
30.6	0.860742	0.057328	34.5	0.824126	0.085142
30.7	0.859852	0.057940	34.6	0.823136	0.085970
30.8	0.858960	0.058558	34.7	0.822144	0.086804
30.9	0.858065	0.059181	34.8	0.821149	0.087644
31.0	0.857167	0.059809	34.9	0.820152	0.088490
31.1	0.856267	0.060441	35.0	0.819152	0.089342
31.2	0.855364	0.061079	35.1	0.818150	0.090201
31.3	0.854459	0.061721	35.2	0.817145	0.091066
31.4	0.853551	0.062369	35.3	0.816138	0.091938
31.5	0.852640	0.063022	35.4	0.815128	0.092816
31.6	0.851727	0.063680	35.5	0.814116	0.093701
31.7	0.850811	0.064343	35.6	0.813101	0.094592
31.8	0.849893	0.065012	35.7	0.812084	0.095490
31.9	0.848972	0.065685	35.8	0.811064	0.096395
32.0	0.848048	0.066364	35.9	0.810042	0.097306
32.1	0.847122	0.067048	36.0	0.809017	0.098224
32.2	0.846193	0.067738	36.1	0.807990	0.099149
32.3	0.845262	0.068432	36.2	0.806960	0.100080
32.4	0.844328	0.069133	36.3	0.805928	0.101019
32.5	0.843391	0.069838	36.4	0.804894	0.101964
32.6	0.842452	0.070549	36.5	0.803857	0.102916
32.7	0.841511	0.071266	36.6	0.802817	0.103875
32.8	0.840567	0.071988	36.7	0.801776	0.104841
32.9	0.839620	0.072716	36.8	0.800731	0.105814
33.0	0.838671	0.073449	36.9	0.799685	0.106795

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