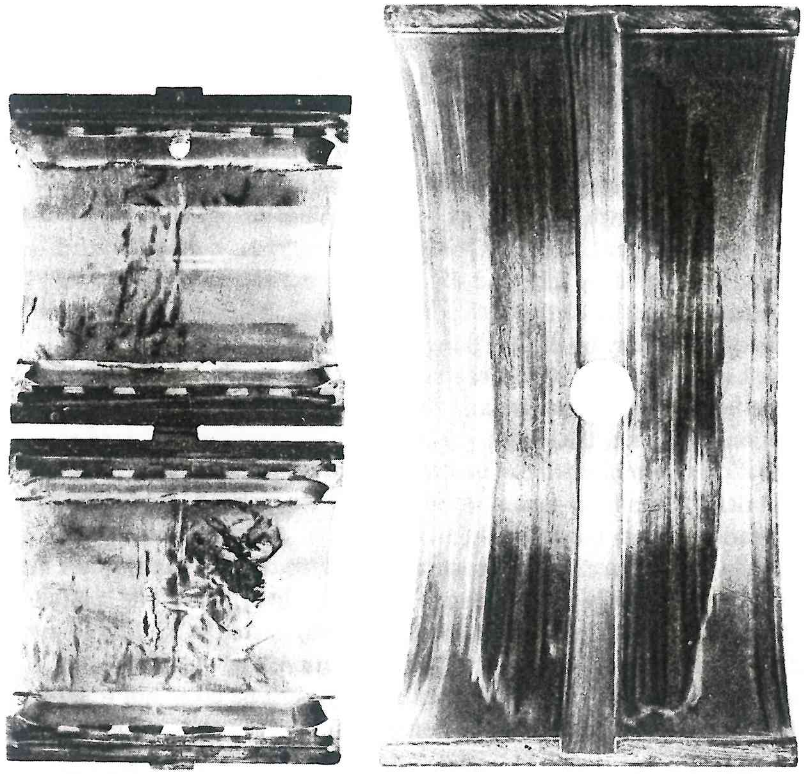

Figure 4.37 Wiping of bearing surface. (Courtesy of Glacier Metal Company Ltd., Wembley, England.)

copper-lead bearing. Lead reacts rapidly with oil oxidation products. To solve this problem oxidation inhibitors are incorporated in the lubricant and protective alloying elements such as tin are included in the lead babbitt formulation.

Faulty Assembly. The fit of the bearing in the housing is a potential source of difficulty. If the bearing is too loose in the housing fretting corrosion can result. This also can occur if the bearing housing design is too flimsy. Too tight a press fit can cause bore distortion. Another cause of bore distortion is entrapment of foreign particles between the bearing and its housing during

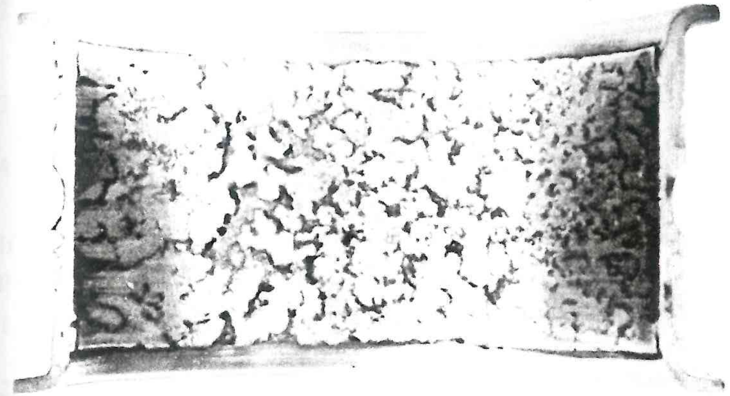

Figure 4.38 Bearing corrosion. (Courtesy of Glacier Metal Company Ltd., Wembley, England.)

assembly. This will result in a localized bore distortion and potential overheating. Misalignment of the shaft with respect to the housing will lead to uneven bearing wear, as shown in Figure 4.39.

Salvage Procedure

Following a journal bearing seizure the shaft may be scored or worn to such an extent that rework is required. There are two potential rework procedures. One is to grind the shaft down and press fit on a sleeve, which is ground to size after assembly. The sleeve must be sufficiently thick to withstand the press fit stresses and the fit should be approximately 0.001 in. per inch of shaft diameter. Another rework method is to grind the shaft down at the distressed area and build it up with chrome plate or a metal spray. Following is a rework procedure for hard chrome plating:

1. Grind the diamter 0.015 in. under the low limit.
2. Bake after grinding for 4 hr at 275°F.
3. Shot peen the rework area using 170 shot to an intensity of 0.012 to 0.014A.
4. Hard chrome plate the rework area and bake after plating for 4 hr at 275°F. Plate to 0.004 in. above the high limit.
5. Grind to finished size.

Care must be taken in grinding the chrome plate since abusive grinding can overheat the base metal, resulting in temper and cracks. These cracks are masked by the chrome plate and cannot be detected by magnaflux inspection. The cracks can propogate through the chrome plate during operation, lifting off the plating and causing bearing failure.

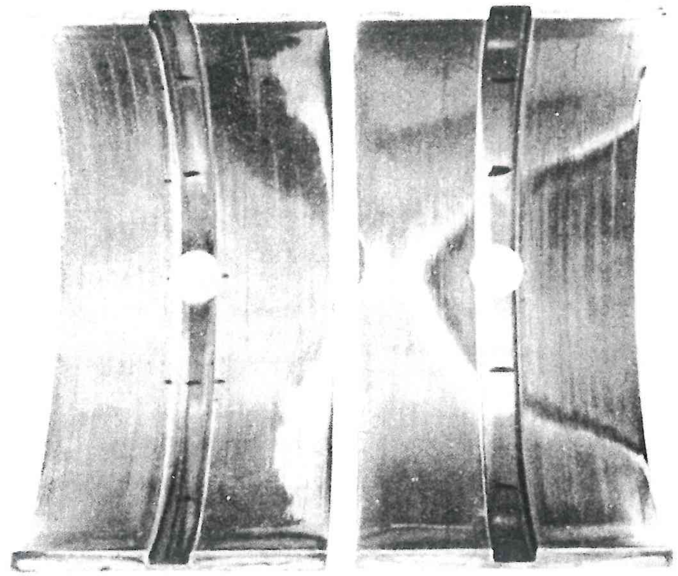

Figure 4.39 Bearing pattern due to misalignment. (Courtesy of Glacier Metal Company Ltd., Wembley, England.)

SEALS

Gearbox shaft seals are usually either elastomeric lip seals or noncontacting labyrinth seals. Elastomeric lip seals are generally found in low-speed applications in conjunction with rolling element bearings. The limiting surface speed of a lip seal is on the order of 3600 fpm. Somewhat higher speeds are possible but require careful design and development. When journal bearings support a shaft, the shaft movement may be excessive for successful lip seal operation.

Labyrinth seals because they are noncontacting are not speed limited and also do not introduce any friction torque into the system. They are used in critical high-speed applications because there is little likelihood of failure. On occasion, carbon face seals are incorporated in gear applications where speeds are high and a more positive seal than a labyrinth is desired.

Labyrinth and lip seals will allow oil leakage if the gearbox internal pressure is higher than the external ambient pressure. This situation may occur for the following reasons:

1. The gearbox may be vented to an area where the pressure is higher than ambient pressure. This causes the unit to be back-pressured, resulting in oil leakage.

2. On occasion the gearbox may serve as the sump for the scavenge oil of another piece of equipment, such as a turbine. This oil may be aerated and at a pressure higher than ambient.
3. The area external to the gearbox may be at a slight vacuum and the gear unit, if vented to atmosphere, will be at a higher internal pressure.
4. If the gear unit is not vented, heat generated in the box can expand the air, creating a pressure differential with the outside.

If any of the conditions noted above result in oil leakage, several solutions are possible:

1. Lip seals with greater interference or higher spring force can be used; however, this may result in excessive wear.
2. Labyrinth seals can be internally pressurized or internally drained to prevent oil leakage.
3. A face seal can be incorporated which can seal against small pressure differentials.

The following paragraphs will describe the seal configurations most commonly used in gearboxes.

Labyrinth Seals

Labyrinth seals are devices that limit leakage between a rotating shaft and a stationary housing by maintaining a close radial clearance between the two. Figure 4.40 illustrates a typical labyrinth seal where several stages of knives deter oil leakage from the gearbox cavity. The labyrinth pictured has two sets of knives with an interstage area which can be used for one of two purposes. A drain can be located between the knives such that any oil leakage past the first stages will be diverted back into the gearbox.

The interstage area can also be used to introduce air pressurization and maintain the air side of the seal at a higher pressure than the oil side. Usually, a pressure differential of 1 or 2 psi is sufficient to effectively eliminate oil leakage. This technique is used when a source of bleed air such as a turbine is readily available. An estimate of the airflow through a labyrinth seal to create a desired pressure differential can be made with the following equation [11]:

$$M = K_1 K_2 K_3 K_4 P_0 A$$

where

M = airflow, lb/sec
K_2 = coefficient dependent on the ratio of radial gap G to knife tip width W (Figure 4.41)

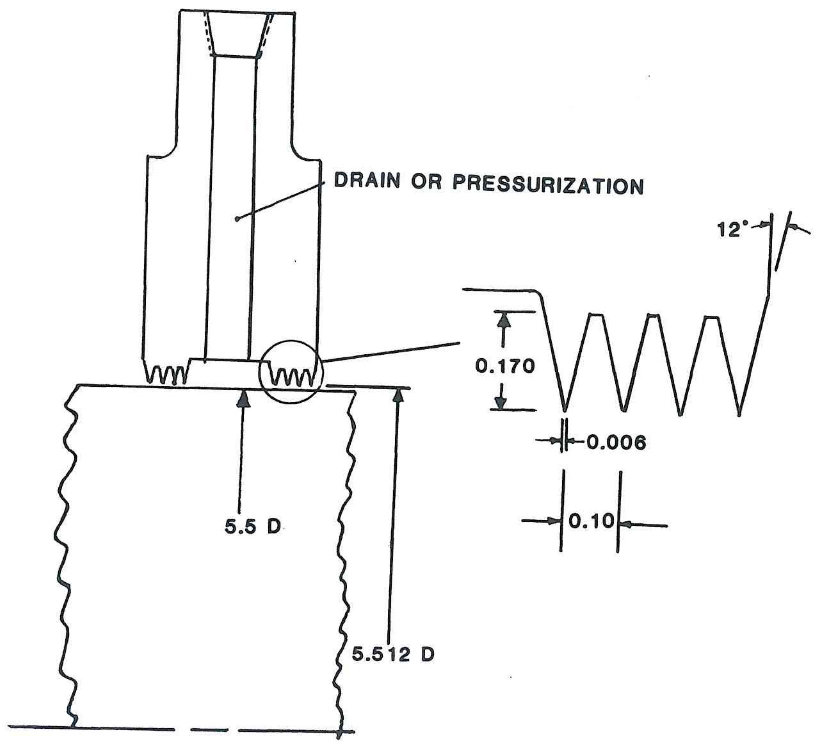

Figure 4.40 Labyrinth seal.

K_3 = coefficient dependent on the ratio of radial gap G to knife pitch P and number of knives (Figure 4.41)
K_1 = coefficient dependent on pressurization air temperature (Figure 4.41)
K_4 = coefficient dependent on the pressure ratio (low pressure/high pressure) across the seal and the number of knives (Figure 4.41)
P_0 = high pressure, psia
A = gap area, in.^{2}

As an illustration of this analysis, assume that a pressure of 2 psig (16.7 psia) is desired in the interstage area of the seal shown in Figure 4.40. The air is at a temperature of 100°F.

$$\text{Pressure ratio} = \frac{14.7}{16.7} = 0.88$$

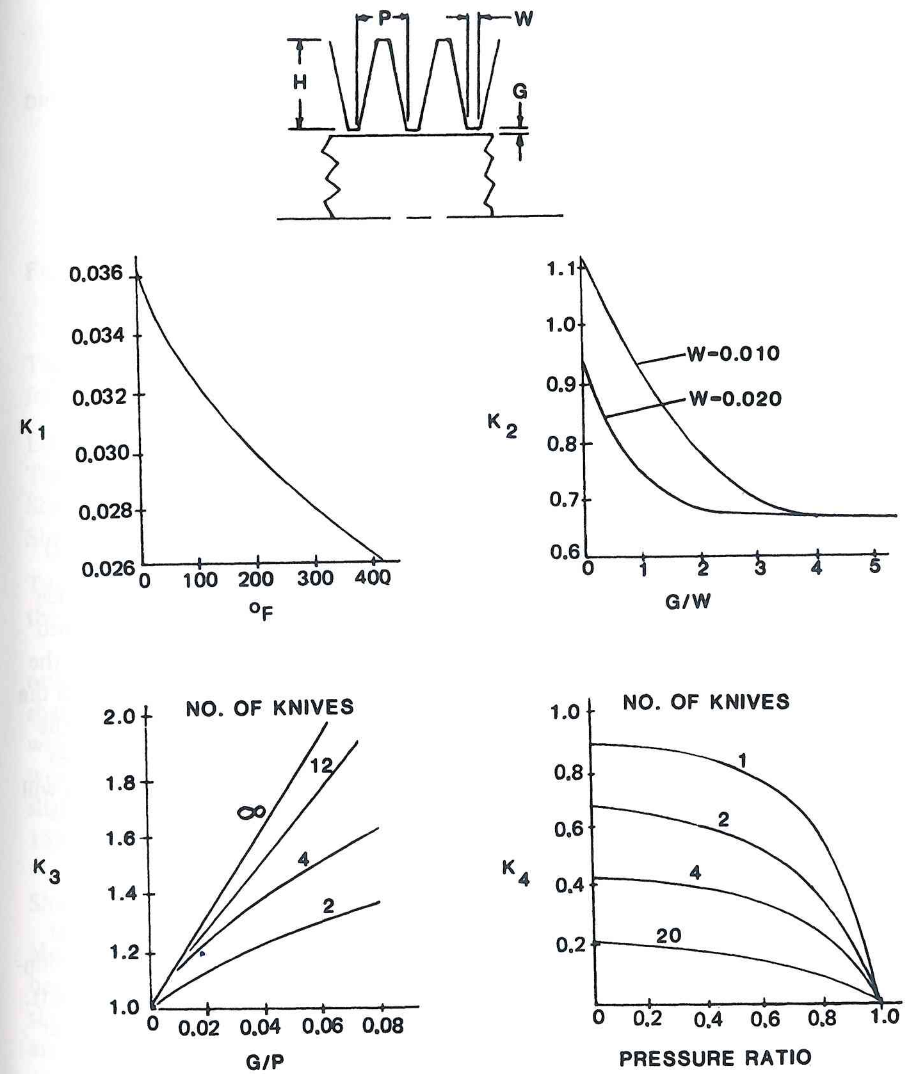

Figure 4.41 Labyrinth seal air leakage. (From Ref. 11.)

$P = 0.10$

$W = 0.006$

Number of knives $= 4$

$K_1 = 0.0325$

$K_2 = 1.0$

$K_3 = 1.5$

$K_4 = 0.2$

$A = 0.104$

$P_0 = 16.7$

$M = 0.017$ lb/sec airflow

If volume flow is desired, the conversion from pounds per second to standard cubic inches per minute (SCIM) is

$$5000 \text{ SCIM} \cong 0.004 \text{ lb/sec}$$

Therefore,

$$M = 21250 \text{ SCIM} = 12.3 \text{ SCFM} \qquad \text{(airflow through one-half the seal)}$$

In the interest of limiting leakage it would be advisable to reduce the clearance to the smallest possible amount. The practical clearance limit must take into account bearing tolerances, dynamic deflections, and thermal distortions. If the shaft motion exceeds the clearance, a rub will occur which will wear or score the knives and the shaft. In journal bearing gear boxes the radial clearance is in the order of 0.010 in. To achieve closer clearances it is possible to coat the knives with an abradable material such as tin babbitt. If the shaft touches, no harm will result and minimum gaps can be maintained.

Elastomeric Lip Seals

Figure 4.42 illustrates a typical elastomeric lip seal. By elastomeric it is meant that the sealing lip is made of a synthetic rubber compound. Oil sealing is accomplished through an interference fit between the flexible sealing lip and the shaft. Because the elastomer can lose tension and elasticity during operation, a garter spring is usually incorporated on the lip to ensure pressure at the sealing interface. In the design of Figure 4.42 a dirt excluder lip on the air side is used to shield the oil sealing lip from contaminants. Because the dirt exclusion lip is unlubricated its design interference will be less than the oil seal lip interference.

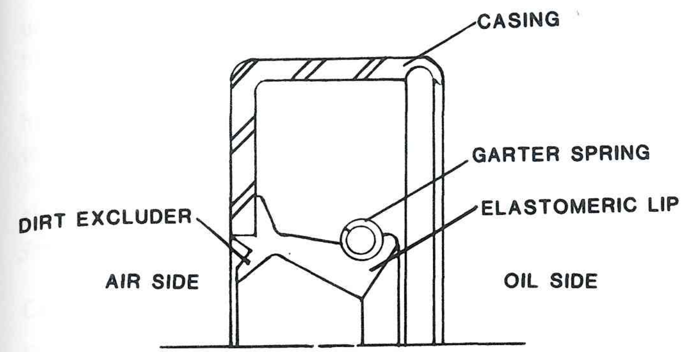

Figure 4.42 Elastomeric lip seal.

The amount of interference and lip contact pressure required depends on the following variables:

Elastomer compression set
Thermal expansion
Shaft eccentricity
Shaft-to-bore misalignment

Typical lip-to-shaft interferences, or pinches as they are sometimes called, are on the order of 0.035 in.

A paradox of lip seals is the fact that although they are sealing devices, in order to operate properly there must be a lubricant film between the lip and the shaft. Successful sealing and long seal life depend on maintenance of this film, which is usually about 0.0001 in. thick. If no film is present, the seal lip will wear out or tear. The majority of lip seal applications, about 80% will leak slightly; on the order of 0.002 g/hr or about 1 drop every 11 hr [12]. About 15% will leak 0.002 to 0.1 g/hr. This level of leakage may be unacceptable.

Shaft Specification

Medium carbon steel such as SAE 1035 or 1045 or stainless steel shafts are the best sealing surfaces, but chrome or nickel plated shafts are also acceptable. The shaft hardness should be Rc 30 minimum. Shaft finish should be 10 to 20 μin. and the shaft should be plunge-ground with no machine lead. Wear sleeves are commercially available which can be pressed over the shaft and used as a sealing surface. One advantage of wear sleeves is that they can be easily replaced in case of surface distress. Seal manufacturers' catalogs list standard shaft diameters for which lip seals are readily available. The diameter tolerances are as follows:

Shaft diameter (in.)	Tolerance
Up to 4.0	±0.003
4.001-6.0	±0.004
6.001-10.0	±0.005

The maximum eccentricity a lip seal can accommodate varies with the shaft rpm. At 1000 rpm the eccentricity can be up to 0.015 in. while at 4000 rpm eccentricity is limited to 0.007 in. Operating eccentricity is a combination of the amount by which the shaft is off center with respect to the bore and the dynamic runout. It can be measured by the total movement of an indicator mounted on the casing and held against the side of the shaft while the shaft is slowly rotated.

Bore Specification

The seal outside diameter-bore interface is a potential oil leakage path. The press fit of the seal in the housing should be a minimum of 0.004 in. with higher press fits for bore diameters above 4 in. These recommendations apply to ferrous housings. If another material such as aluminum is used, the higher coefficient of thermal expansion must be considered. The bore finish should be a maximum of 125 μin. It is possible to procure lip seals with elastomeric coatings on the outside diameter which will seal at the bore interface, but these seals must be assembled with higher press fits than seals with metal casings. It is good practice to brush a synthetic rubber coating on the bore inside diameter prior to assembly to fill minor imperfections on the bore and enhance sealing.

Lip Seal Materials

The most commonly used lip seal materials in gearbox applications are the nitriles, sometimes known as Buna N. The standard nitrile compound is compatible with most mineral oils and can operate continuously at temperatures from -65 to 225°F. Nitrile seals are relatively inexpensive and readily available. This material is not suited for temperatures above 250°F because it tends to harden and it is not recommended for highly compounded lubricants (EP additives).

For high-speed and high-temperature applications fluoroelastomers are recommended. These materials, sometimes known by the trade name Viton, can be used in a temperature range of -40 to 400°F. They have outstanding resistance to a wide variety of fluids, including synthetic lubricants such as Mil-L-7808 and Mil-L-23699. For critical applications the fluoroelastomers are

undoubtedly the best technical choice. The disadvantages of this material are higher cost and limited availability of particular configurations.

Polyacrylate elastomers are used for operation with EP-type fluids. They have good resistance to temperatures up to 300°F but have poor low-temperature properties.

Silicones are applicable to a wide temperature range of -100 to 300°F and also have low friction and wear properties. They have poor resistance to oxidized oil and some EP additives.

Carbon Face Seals

Figure 4.43 illustrates a carbon face seal installation. The seal assembly consists essentially of two flat-surfaced rings, the runner rotating with the shaft and the stationary carbon member carried in the housing. The carbon sealing face must be free to follow the motions of the runner despite axial shaft movement or deviations from pure rotational movement. In order to accommodate shaft axial and radial movement, yet maintain a sealing force, the carbon sealing face is spring loaded against the runner. The spring force is basically constant over some range of axial movement and the seal must be axially positioned at assembly to be within its operating range. This may be accomplished by shimming at assembly. Because the carbon sealing face is free floating a secondary static seal is required to close the leakage path and this is usually an elastomeric O-ring. To achieve long seal life the spring force must be minimized yet adequate to overcome axial friction, inertia, and dynamic forces. It can be seen from Figure 4.43 that at the radial sealing interface fluid tends to be pumped back into the bearing cavity due to centrifugal force; thus the face seal can seal against small pressure differentials.

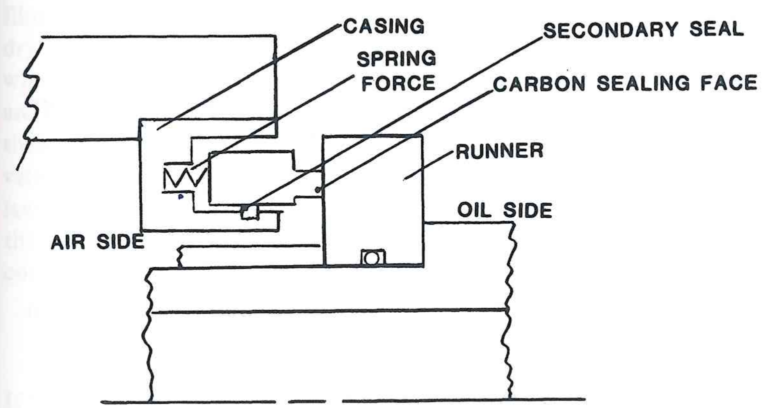

Figure 4.43 Carbon face seal.

The rubbing surfaces must be compatible such that during mechanical contact there is no galling, abrasion, or wear. Carbon is used because of its rubbing friction characteristics, compatibility with many materials, temperature resistance, low mass, low porosity, and ease of fabrication. Runners can be hardened stainless steel or plated steels such as 4140 coated with chrome or carbide. Runner hardness should be Rc 58 minimum and the runners should be ground and lapped to a flatness of three helium light bands.

Normally in a face seal installation the runner is the rotating member; however, there are some designs with stationary runners and rotating seal assemblies. This type of design is speed limited because of the effect of centrifugal force on the internal seal components.

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5
LUBRICATION SYSTEMS

The purpose of a gearbox lubrication system is to provide an oil film at the contacting surfaces of working components to reduce friction. Equally as important, the lubrication system absorbs heat generated in the gearbox and dissipates it so that component temperatures do not become excessive. Prior to discussing specific details of gearbox lubrication, it is appropriate to review lubrication fundamentals.

VISCOSITY

Friction between rubbing surfaces is reduced by separating the surfaces with a film of oil. The lower fluid friction is substituted for the frictional resistance of dry metal surfaces. Viscosity is a measure of fluid friction. Consider Figure 5.1, where plate 1 is moving with a velocity U over a stationary plate 2 and the plates are separated by an oil film of thickness h. The force F on plate 1 causes a shearing stress in the fluid. The fluid in contact with the moving plate is at velocity U and the fluid in contact with the stationary plate is at rest. Newton's law of viscous flow states that the shearing stress in the fluid is proportional to the rate of change of the velocity, and if we assume that the rate of shear is constant,

$$S_s = \frac{\mu U}{h}$$

If U is in inches per second, h is in inches, and S_s is in pounds per square inch, μ is the absolute viscosity in pound seconds per inch squared. A pound second

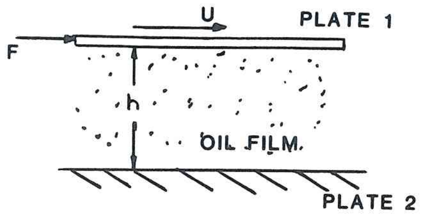


Figure 5.1 Fluid friction.

per inch squared is called a Reyn in honor of Osborne Reynolds. In the metric system the absolute viscosity is usually expressed in centipoise:

$$Z = \mu(6.9 \times 10^6)$$

where

Z = centipoise or 1/100 dyne-sec/cm^2
μ = Reyns or lb-sec/in.^{2}

Quite often, viscosity is expressed in Saybolt seconds universal (SSU), where the viscosity is determined with an instrument that measures the time in seconds for 60 cm^3 of a fluid to pass through a standard capillary tube at a given temperature. This is a kinematic viscosity and is related to absolute viscosity as follows:

$$Z_k = \left(0.22T - \frac{180}{T}\right)$$

where

Z_k = kinematic viscosity, cSt
T = number of Saybolt seconds

The absolute viscosity in centipoise is

$$Z = \rho Z_k = \rho\left(0.22T - \frac{180}{T}\right)$$

where ρ is the specific gravity at the given temperature. Specific gravity is measured by the petroleum industry at 60°F with a glass hydrometer calibrated in degrees as specified by the American Petroleum Institute. This is an arbitrary scale which is converted to actual specific gravity as follows:

$$\rho_{60} = \frac{141.5}{131.5 + \text{degrees API at } 60^\circ\text{F}}$$

Table 5.1 Viscosity and Specific Gravity of SAE Oils

SAE number	Saybolt seconds universal		Degrees API at 60°F	ρ at 60°F	Specific gravity, ρ	
	100°F	210°F			100°F	210°F
10	183	46	30.2	0.875	0.861	0.822
20	348	57	29.4	0.879	0.865	0.827
30	489	65	28.7	0.883	0.869	0.830
40	680	75	28.3	0.885	0.871	0.832
50	986	90	26.6	0.895	0.881	0.842
60	1394	110	26.3	0.897	0.883	0.844
70	1846	130	25.6	0.901	0.887	0.848

where ρ_{60} is the specific gravity at 60°F. The specific gravity at any other temperature is related to ρ_{60} as follows:

$$\rho = \rho_{60} - 0.00035(^\circ\text{F} - 60)$$

Table 5.1 gives SSU and ρ_{60} values for various SAE oils. It can be seen from this table that the viscosity of oil varies widely with the operating temperature. The viscosity-temperature relationship of lubricating oils can be plotted as a straight line if the scales are correctly arranged. To determine the viscosity of an oil at

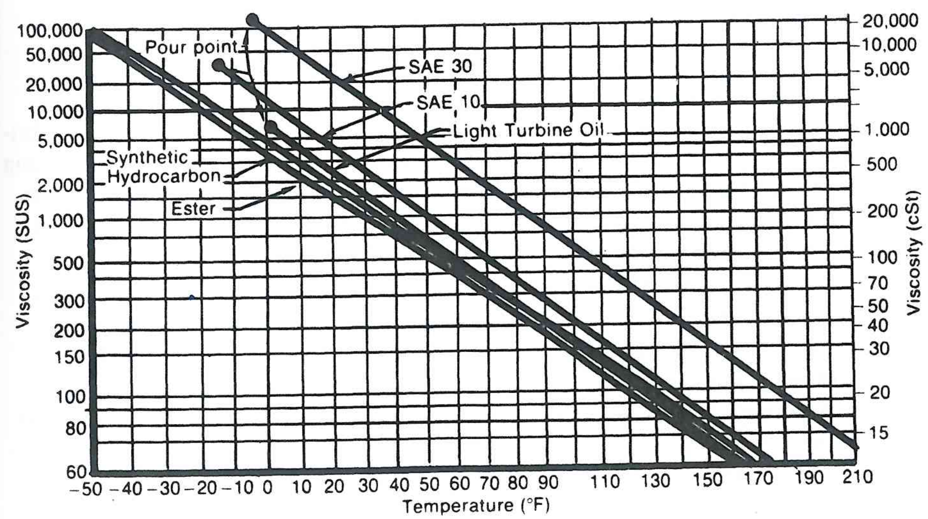


Figure 5.2 Oil viscosity versus temperature. (From Ref. 1.)

any temperature the Saybolt seconds are found at two temperatures, usually 100 and 210°F and a straight line is drawn between these points. Figure 5.2 presents the viscosities of some common fluids as a function of operating temperature.

VISCOSITY INDEX

The viscosity index is a measure of how much the oil viscosity varies with temperature. It would be ideal if the fluid viscosity were the same at low and high temperatures, but this is unattainable. Fluids that have low viscosity variations with temperature have a high viscosity index, whereas a low viscosity index defines a fluid with a widely fluctuating viscosity with respect to temperature. Typical viscosity indexes for petroleum oils range from 90 to 110.

POUR POINT

The pour point of an oil is the lowest temperature at which it will flow. This parameter must be considered when designing a unit for outdoor use; however, in most low-temperature applications the oil will be heated prior to startup. When no provision for oil heating is made, the pour point should be 20°F above the lowest expected ambient temperature.

GEAR LUBRICANTS

The selection of a gear oil depends on several factors, including the unit's operating speed and load, ambient temperatures, and which lubricants will be available at the operating site. The most important parameter in selecting a lubricant is the viscosity. High-speed units require less viscous oil than gears operating at low speed. At high speed an acceptable oil film is generated at the tooth contact even with a low-viscosity oil. Also, the churning that occurs at high speed will be less severe with a low-viscosity oil, resulting in lower power losses. At lower operating speeds a thinner oil film is generated and more viscous oils are required to separate the contacting surfaces. Also, low-speed gears are loaded to higher levels. Often, a gear unit will contain both high- and low-speed gear meshes. In this case a compromise must be struck and some development may be necessary. AGMA Standard 250.04 [2] defines a series of oils by viscosity as shown in Table 5.2. Each grade is given an American Gear Manufacturers Association (AGMA) lubricant number.

Table 5.2 Viscosity Ranges for AGMA Lubricants

Rust and oxidation inhibited gear oils	Viscosity range[a]		Extreme pressure gear lubricants[c]	Viscosities of former AGMA system[d]
AGMA lubricant number	mm^2/s (cSt) at 40°F	Equivalent ISO grade[b]	AGMA lubricant number	SSU at 100°F
1	41.4-50.6	46		193-235
2	61.2-74.8	68	2 EP	284-347
3	90-110	100	3 EP	417-510
4	135-165	150	4 EP	626-765
5	198-242	220	5 EP	918-1122
6	288-352	320	6 EP	1335-1632
7 Comp[e]	414-506	460	7 EP	1919-2346
8 Comp[e]	612-748	680	8 EP	2837-3467
8A Comp[e]	900-1100	1000	8A EP	4171-5098

Note: Viscosity ranges for AGMA lubricant numbers will henceforth be identical to those of ASTM 2422.
[a]Viscosity System for Industrial Fluid Lubricants, ASTM 2422. Also British Standards Institute, B.S. 4231.
[b]Industrial Liquid Lubricants–ISO Viscosity Classification. International Standard, ISO 3448.
[c]Extreme pressure lubricants should be used *only* when recommended by the gear drive manufacturer.
[d]AGMA 250.03, May 1972 and AGMA 251.02, November 1974.
[e]Oils marked Comp are compounded with 3 to 10% fatty or synthetic fatty oils.
Source: Ref. 2.

Table 5.3 Specification for R&O Gear Oils (Including Compounded Gear Lubricants)

Property	Test procedure	Criteria for acceptance		
Viscosity	ASTM D88	Must be as specified		
Viscosity index	ASTM D2270	90 min		
Oxidation stability	ASTM D943	Time to reach a neutralization number of 2.0:[a]		
		AGMA grade	Hours (minimum)	
		1, 2	1500	
		3, 4	750	
		5, 6	500	
Rust protection	ASTM D665	No rust after 24 hr with synthetic seawater		
Corrosion protection	ASTM D130	No. 1 strip after 3 hr at 120°C (250°F)		
Foam suppression	ASTM D892	Must be within these limits:	Max volume of foam (ml) after:	
		Temperature	5-min blow	10-min rest
		Sequence I 24°C (75°F)	75	10
		Sequence II 93.5°C (200°F)	75	10
		Sequence III 24°C (75°F)	75	10
Demulsibility	ASTM D2711	Must be within these limits:[a]		
		Max. percent water in the oil after 5-hr test	0.5%	
		Max. cuff after centrifuging	2.0 ml	
		Min. total free water collected during entire test	30.0 ml	
Cleanliness	None	Must be free from grit and abrasives		

[a]The criteria for acceptance indicated for oxidation stability and demulsibility are not applicable to compounded gear oils.
Source: Ref. 2.

RUST- AND OXIDATION-INHIBITED OILS

AGMA lubricant numbers 1 to 8 are petroleum-based rust- and oxidation-inhibited oils which meet certain American Society of Testing and Materials (ASTM) specifications, as shown in Table 5.3:

Oxidation Stability (ASTM D943) In this laboratory test, pure oxygen is bubbled through a mixture of oil and water in the presence of copper and iron wire catalysts at 95°C. The test life is reported as the time in hours it takes the oil to reach an acidity number of 2.0.

Rust Protection (ASTM D665) This procedure measures the rust-preventing characteristics of an oil in the presence of synthetic seawater. A steel specimen is used.

Corrosion Protection (ASTM D130) This test evaluates the ability of an oil to control the corrosion of copper and copper alloys in the presence of water at elevated temperatures.

Foam Suppression (ASTM D892) When air contaminates an oil circulation system, the efficiency of the system depends on the oil's natural resistance to foaming and its ability to break foam quickly. In this test foam is created in the test oil by blowing air through it for 5 min. The volume of foam is recorded at the end of a 10-min settling period.

Demulsibility (ASTM D2711) The degree of oil demulsibility is determined by the amount of time it takes equal portions of distilled water and oil to separate after the two have been mixed by a 1500-rpm steel paddle at a temperature of 180°F.

The lighter rust- and oxidation-inhibited oils, AGMA lubricant numbers 1 to 4, are sometimes referred to as turbine oils and are widely used in gear units, particularly those operating at high speeds.

EXTREME PRESSURE GEAR LUBRICANTS

Extreme pressure lubricants are petroleum-based oils containing special chemical additives which can increase the load-carrying capacity of gears by forming a film on the metal surfaces which provides separation when the lubrication film becomes thin enough for the asperities to contact. Some boundary films will melt at lower temperatures than others and will then fail to provide protection at the surfaces. For this reason, many extreme pressure lubricants contain more than one chemical for protection over a wide temperature range. Some of the EP additives commonly used in gear oils are those containing one or more compounds of chlorine, phosphorus, sulfur, or lead soaps. EP additives are

Table 5.4 Specification for Extreme Pressure Gear Lubricants

Property	Test procedure	Criteria for acceptance			
Viscosity	ASTM D88	Must be as specified			
Viscosity index	ASTM D2270	90 min			
Oxidation stability	ASTM D2893	Increase in kinematic viscosity of oil sample at 95°C (210°F) should not exceed 10%			
Rust protection	ASTM D665	No rust after 24 hr with distilled water			
Corrosion protection	ASTM D130	No. 1 strip after 3 hr at 100°C (212°F)			
Foam suppression	ASTM D892	Must be within these limits:			
				Max volume of foam (ml) after:	
			Temperature	5-min blow	10-min rest
		Sequence I	24°C (75°F)	75	10
		Sequence II	93.5°C (200°F)	75	10
		Sequence III	24°C (75°F)	75	10
Demulsibility	ASTM D2711 (Modified for 90 ml water)	Must be within these limits:			
			AGMA grades		
			2EP to 6 EP	7EP, 8EP	
		Max. percent water in the oil after 5-hr test	1.0%	1.0%	
		Max. cuff after centrifuging	2.0 ml	4.0 ml	
		Min. total free water collected during entire test (start with 90 ml of water)	60.0 ml	50.0 ml	
Cleanliness	None	Must be free from grit and abrasives			
EP property	ASTM D2782 (Timken Test) DIN 51-354 (FZG Test)	An oil must pass both a 60-lb Timken OK load, and 11 stages on the FZG machine with A/8.3/90°C parameters for acceptance.			
Additive solubility	None	Must be filterable to 25 μm (wet or dry) without loss of EP additive			

Source: Ref. 2.

chemically reactive and care must be taken when they are used that metals such as zinc or copper which may be in the gear unit are not attacked. An existing unit should not be changed from a straight mineral oil to an EP oil without the manufacturer's approval.

Lubricant numbers 2EP to 8EP in AGMA Standard 250.04 [2] are EP oils which meet certain ASTM specifications, as shown on Table 5.4. Most of the tests are as those described for Table 5.3, however, for EP properties a Timken test and a FZG test are specified.

In the Timken test a rectangular steel block is brought into contact with a rotating steel cylinder to determine the maximum load a lubricant will carry before its film strength is exceeded. The load is increased until the block shows surface distress, which indicates lubricant failure. The FZG test uses operating gears to measure wear and surface distress. This is a back-to-back test with two connected gears operating at 1760 rpm and two connected 2640-rpm pinions. Load is applied by loosening the coupling in the pinion shaft and placing weights on a load arm. The preload is locked into the system by tightening the bolts of the coupling, and the load arm and weights are removed before starting the machine. Twelve uniform weights are consecutively applied and after each run the gears are checked for weight loss and visual condition. When a 10-mg weight loss is recorded between two runs the oil is considered to have failed. The test gears are spur and are immersed in 194°F oil during operation.

SYNTHETIC LUBRICANTS

Synthetic lubricants are a broad range of compounds derived from chemical synthesis rather than from the refining of petroleum. They have the following advantages:

High-temperature thermal and oxidative stability
Low-viscosity variation over a broad temperature range
Low-temperature capability
Long service life

Synthetic lubricants used in helicopter transmissions and geared gas turbine engines have the ability to operate at temperatures as low as -65°F and as high as 400°F or more during a duty cycle. These lubricants have been developed for military applications and are often designated by military specifications.

The largest class of synthetic lubricants in use today are the esters, which are materials that contain the ester chemical linkage. Two oils that are commonly used in gear applications are Mil-L-23699 and Mil-L-7808. Esters are characterized by an even balance of properties. They have wide operating

Table 5.5 AGMA Lubricant Number Recommendations for Enclosed Helical, Herringbone, Straight Bevel, Spiral Bevel, and Spur Gear Drives

	Ambient temperature[b-e]	
Type of unit[a] and low-speed center distance	−10 to +10°C (15 to 50°F)	10 to 50°C (50 to 125°F)
Parallel shaft (single reduction)		
Up to 200 mm (to 8 in.)	2–3	3–4
Over 200 mm to 500 mm (8 to 20 in.)	2–3	4–5
Over 500 mm (over 20 in.)	3–4	4–5
Parallel shaft (double reduction)		
Up to 200 mm (to 8 in.)	2–3	3–4
Over 200 mm (over 8 in.)	3–4	4–5
Parallel shaft (triple reduction)		
Up to 200 mm (to 8 in.)	2–3	3–4
Over 200 mm, to 500 mm (8 to 20 in.)	3–4	4–5
Over 500 mm (over 20 in.)	4–5	5–6
Planetary gear units (housing diameter)		
Up to 400 mm (to 16 in.) O.D.	2–3	3–4
Over 400 mm (over 16 in.) O.D.	3–4	4–5
Straight or spiral bevel gear units		
Cone distance to 300 mm (to 12 in.)	2–3	4–5
Cone distance over 300 mm (over 12 in.)	3–4	5–6
Gearmotors and shaft-mounted units	2–3	4–5
High-speed units[f]	1	2

[a]Drives incorporating overrunning clutches as backstopping devices should be referred to the gear drive manufacturer as certain types of lubricants may adversely affect clutch performance.
[b]Ranges are provided to allow for variation in operating conditions such as surface finish, temperature rise, loading, speed, etc.
[c]AGMA viscosity number recommendations listed above refer to R & O gear oils shown in Table 5.3. EP gear lubricants in the corresponding viscosity grades may be substituted where deemed necessary by the gear drive manufacturer.
[d]For ambient temperatures outside the ranges shown, consult the gear manufacturer. Some synthetic oils have been used successfully for high- or low-temperature applications.
[e]Pour point of lubricant selected should be at least 5°C (9°F) lower than the expected minimum ambient starting temperature. If the ambient starting temperature approaches the lubricant pour point, oil sump heaters may be required to facilitate starting and ensure proper lubrication.
[f]High-speed units are those operating at speeds above 3600 rpm or pitch line velocities above 25 m/sec (5000 fpm) or both. Refer to AGMA Standard 421, Practice for High Speed Helical and Herringbone Gear Units, for detailed lubrication recommendations.
Source: Ref. 2.

temperature ranges and high viscosity indexes (125 to 250). Thus esters require low torques for low-temperature operation and provide good lubrication characteristics at high temperatures.

A limitation of ester lubricants is low compatibility with some polymeric materials such as those used in seals. Also, synthetic lubricants are significantly more expensive than conventional petroleum oils.

LUBRICANT VISCOSITY SELECTION

In general, the lowest viscosity oil sufficient to form an adequate oil film at all operating conditions should be chosen. As a guide, Table 5.5 gives recommended lubricant numbers for enclosed drives. The viscosity associated with each number is presented in Table 5.2. In practice, the lubricant selection is usually a compromise between the requirements of the various oil-wetted components such as gears and bearings and the particular application requirements. For instance, a turbine generator set packager may desire a common lubrication system for the turbine, gearbox, and generator. If the turbine requires a particular lubricant such as a synthetic oil, the gear manufacturer will be requested to design for this fluid.

TYPES OF LUBRICATION SYSTEMS

There are two types of lubrication systems in use: splash and forced feed. In the splash system the unit is filled with oil to a predetermined level and operated as a sealed system with no external connections. In the forced feed system, oil is introduced into the unit through jets under pressure. Scavenge oil is pumped through a cooler and filter prior to reentering the gearbox.

The splash system is far simpler and less expensive than the forced feed design but is applicable only to low-speed units. As speeds increase, the heat generated in the gearbox becomes excessive and an external system is required to cool the lubricant. Also, oil must be precisely introduced at the gear and bearing interfaces, and this is accomplished through strategically placed jets.

For every gear drive there is a mechanical rating: the load the transmission can transmit based on stress considerations. In addition, there is a thermal rating, which is the average power that can be transmitted continuously without overheating the unit and without using special cooling. AGMA thermal ratings [3] are based on a maximum oil sump temperature of 200°F. If the thermal rating is less than the mechanical rating, additional cooling or a forced-feed lubrication system is required.

An empirical method for estimating the thermal rating of low speed (maximum rpm of 3600 or pitch line velocity of 5000 fpm) gear units is given in Ref. 3. Basically, the calculation of the thermal rating is a heat transfer problem where assumptions must be made for the heat generated in the unit and the coefficient of heat transfer of the surface area of the casing. If sufficient heat can be conducted and radiated through the casing the gearbox sump temperature will stabilize below the limiting temperature. Quite often gear units are located in areas where conditions are detrimental to good heat transfer. The following environmental conditions must be considered when defining the thermal rating [3]:

Operation in an enclosed space
Operation in a very dusty atmosphere where fine material covers the gear unit
Operation in a high-temperature ambient such as near motors, turbines, or hot processing equipment
Operation in high altitudes
Operation in the presence of solar or radiant energy

Auxiliary cooling can be used in combination with splash lubrication to increase the thermal rating. Air is forced past the radiating surfaces of the gear casing by strategically placed fans. The casing can also be cooled by a water jacket. In this scheme water passages are built into the gear housing, usually on the high-speed side, and heat is carried away by a cooling water flow. To operate a gear unit at maximum efficiency, auxiliary cooling schemes should include thermostatic controls so that the oil is not cooled unnecessarily. Operating with too cool a lubricant increases churning losses. The heat transfer from the gear casing can be increased by adding cooling fins which will increase the surface area.

Splash Lubrication

Successful splash lubrication of a gearbox is more a matter of development than analysis. The problem is to distribute oil to each component sufficient for lubrication and cooling yet minimize churning losses and heat generation. Figure 5.3 shows a two-stage parallel shaft unit using splash lubrication. The simplest scheme is to let the bull gear dip into the oil, as shown in the upper part of Figure 5.3, and lubricate all bearings and the high-speed mesh by the mist that is created by the action of the bull gear churning through the oil. This scheme will be more successful wtih antifriction bearings than with journal bearings, which require far more oil. To provide a positive oil supply to the bearings a feed trough can be incorporated, as shown in Figure 5.3, which catches oil flung off the bull gear. The oil drips down the casing walls, is trapped by the trough, and is distributed to the bearings.

A further degree of sophistication is shown at the bottom of Figure 5.3. An oil baffle surrounds the lower portion of the bull gear, allowing the oil level

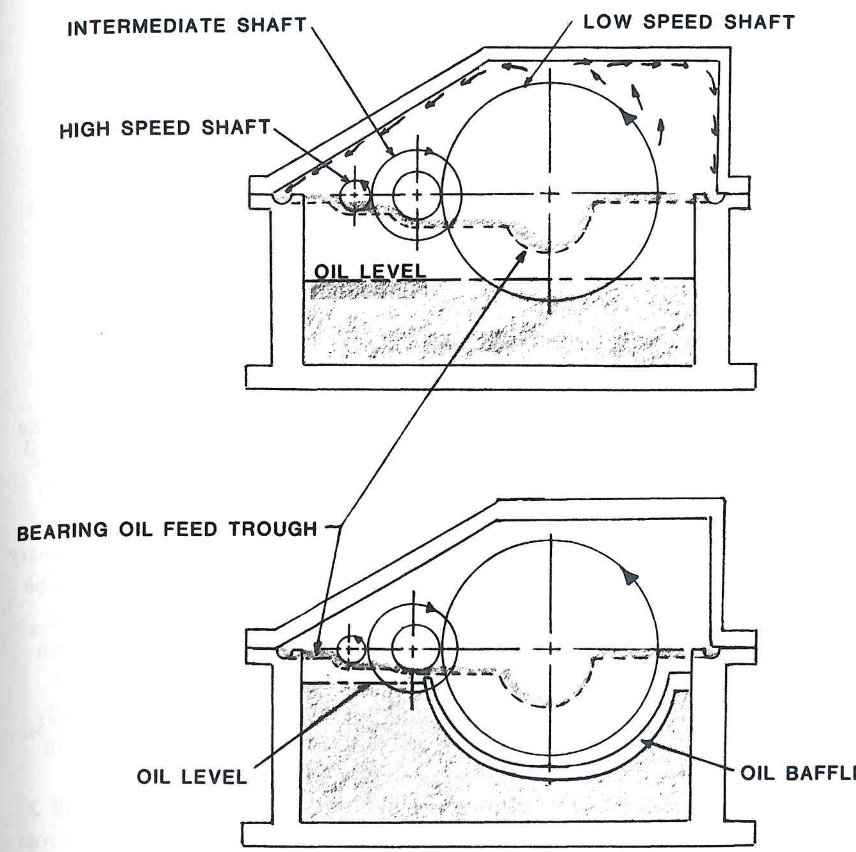


Figure 5.3 Examples of splash lubrication.

to be raised such that the large gear on the intermediate shaft dips into the lubricant and positively lubricates the high-speed mesh. Because the bull gear cannot churn through the sump, heat generation is minimized and efficiency increased. If the bull gear speed is so slow that sufficient oil is not splashed on the casing walls to lubricate the bearings, scrapers can be used to strip the oil ring that rotates with the bull gear rim. The oil is stripped from the rim and directed to the bearing feed trough (Figure 5.4). A splash lubrication system requires an oil drain, oil filler, an oil-level monitor and a breather. In cold ambients an immersion heater should be provided in the sump. An opening to the sump sufficiently large to allow cleaning of sludge after draining is also good practice.

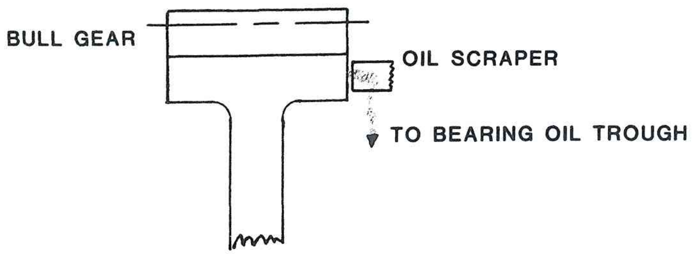


Figure 5.4 Application of oil scraper.

Forced-Feed Lubrication

Figure 5.5 illustrates a typical forced-feed lubrication system for a gearbox. The shaft-driven oil pump (A) sucks oil from the tank (D) through the suction pipe (I). From the pressure side of the oil pump the oil is directed through a cooler (C) and filter (B). A pressure relief valve (Q) is located at the inlet to the gearbox to hold the feed pressure at a constant predetermined level. An auxiliary pump (N) is incorporated to prime the system prior to starting. It could also be used as a backup in case of failure of the main pump. Check valves (G) are located such that the main pump does not pump through the auxiliary system and the auxiliary pump does not pump into the pressure side of the main oil pump. A bypass is provided at the cooler (H) which can be thermostatically controlled so that the oil is not cooled to too low a level. At various points in the system temperature and pressure sensors are located.

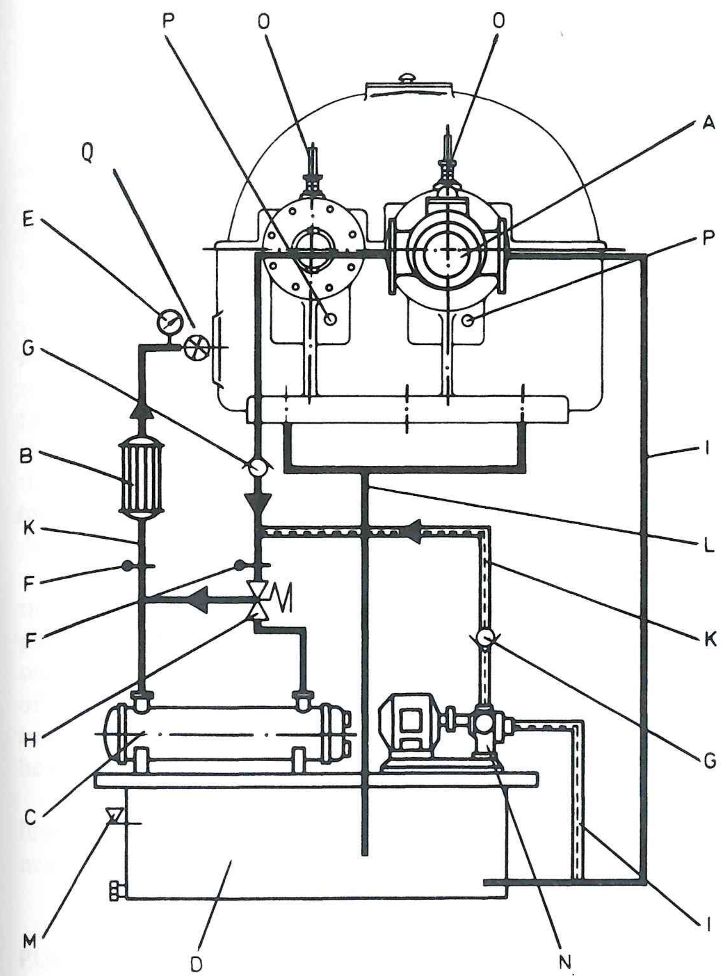
When designing a lubrication system the first step is to estimate the oil flow to the components and the gearbox efficiency. The temperature rise across the gearbox can then be calculated:

$$\Delta T = \frac{Q}{MC_p}$$

where

ΔT = temperature rise, °F
M = oil flow, lb/min (*Note*: 1 gpm $\cong$ 7.5 lb/min)
Q = heat generated, Btu/min [*Note*: Q = hp(42.4)]
C_p = specific heat of oil, $\cong$ 0.5 Btu/lb-°F

For example, a gearbox transmitting 1000 hp with 98% efficiency will reject 20 hp or 848 Btu/min of heat to the oil. If the gearbox flow is 20 GPM or 150 lb/min the temperature rise across the gearbox will be 11°F. In other words, if the oil enters the unit at 130°F it will gain 11°F and be at a 141°F temperature level in the sump. Some working areas of the gearbox may be at significantly higher temperatures than the bulk sump temperature.



Key: A, oil pump; B, oil filter; C, oil cooler; D, oil tank; E, pressure gauge; F, thermometer; G, check valve; H, overflow valve; I, suction pipe; K, pressure pipe; L, return flow pipe; M, oil level gauge; N, primer and spare pump unit; O, temperature and flow gauge; P, connection for remote thermometer; Q, pressure regulator.

Figure 5.5 Forced-feed lubrication system. (Courtesy of American Lohmann Corporation, Hillside, N. J.)

A typical maximum oil in temperature limit for a gearbox with forced-feed lubrication is 130°F. The cooler must be sized to cool the oil to this temperature under the worst operating conditions, such as a hot day. A typical oil temperature rise might be 30°F across the unit. These values are for mineral oils; synthetic oils can operate at higher temperature levels. Some turbine gear sets operate with oil in temperatures of 170°F.

The amount of oil flow supplied to a gear mesh is usually based on experience and experimental data. A rule of thumb might be 0.017 lb/min per horsepower or 0.002 gpm/hp. This is a generous flow and should result in a low-temperature rise. To optimize the efficiency of a gear unit a program can be conducted, gradually reducing oil flow until a predetermined maximum temperature rise is reached. Reducing oil flow will always increase efficiency, particularly in high-speed gearing, where churning is significant, but at the expense of higher scavenge temperatures.

The quantity of oil flow passing through the gearbox is regulated by the oil jet diameters, journal bearing leakages, and oil feed pressure. Usually, gearbox feed pressures are on the order of 20 to 100 psig. To hold a constant feed pressure a regulating valve is incorporated at the gearbox inlet as shown in Figure 5.5 A typical pressure regulating valve will have a spring-loaded bypass loop which opens up when the design feed pressure is exceeded and directs excess oil back to the sump.

The flow of oil through a jet can be calculated as follows:

$$Q = KA\sqrt{\frac{2G\,\Delta P}{W}}\,(W)(60)$$

where

Q = flow, lb/min
K = oil jet discharge coefficient
G = gravity acceleration, 386 in./sec^2
ΔP = pressure drop across the jet, lb/in.^{2}
W = oil specific weight, lb/in.^{3}
A = jet area, in.^{2}

If the discharge coefficient is assumed to be 0.65 and W is taken as 0.032 lb/in.^{3} the equation is

$$Q = 194A\sqrt{\Delta P} = 152(D)^2\sqrt{\Delta P}$$

where D is the oil jet diameter inches. Note that oil flow varies with the square of the jet diameter and as the square root of the feed pressure. The minimum practical jet nozzle diameter is approximately 0.030 in. in diameter. Smaller jets tend to clog too easily due to foreign materials in the oil stream. It is good practice to have at least two jets at each lubricating position in case one clogs up.

Gears with long face widths must have oil evenly applied along the face width. If oil is introduced unevenly, thermal distortions will cause uneven load distribution. A long-face-width gear should have several jets spraying along the axis. There are also special spray nozzles that can be used to fan the oil flow out over a large area.

Relatively little oil is required for lubrication provided that it is properly applied. The bulk of the oil flow is required for cooling the gear tooth and blank. Thus it would appear logical to spray a small amount of oil at the incoming side of a gear mesh and a large amount of oil at the outgoing side, where it can do the most efficient cooling job. In practice, for low- and moderate-speed gear applications lubrication on either the incoming or outgoing side can be satisfactorily developed. For very high speed gears, above 20,000 fpm, it has been shown that lubrication on the outgoing side is more efficient than on the incoming. This is because the bulk of the oil is not being dragged through the mesh, experiencing churning losses. High-speed gear lubrication, however, is very much an art and a particular application may require extensive development to minimize thermal distortions and heat generation.

The extent to which the gear mesh is penetrated by the oil jet is a function of the velocity of the jet. The velocity in turn is a function of the feed pressure. Industrial applications generally operate with feed pressures in the order of 30 psig, whereas high-speed aerospace gearsets introduce oil at pressures of 100 psig. There are no conclusive studies determining optimum feed pressures; however, for high-speed gears it would seem that higher pressures would be beneficial.

Figure 5.5 illustrates the components incorporated in a typical forced-feed lubrication system. The following paragraphs describe in detail the major components of such a system.

PUMPS

The gearbox oil pump must deliver a given quantity of oil over a wide range of oil in temperatures and viscosities. At startup the oil will be cold and viscous. During operation on hot days the lubricant will warm up to the maximum design allowable value. Also, the pump must be capable of priming itself and overcoming the pressure drops in the line between the reservoir and the suction (inlet) port of the pump.

For gearbox lubrication rotary pumps are generally used. Figure 5.6 illustrates some widely used configurations. Rotary pumps are positive-displacement devices which deliver a given quantity of fluid with each revolution. At any particular speed the oil flow is practically constant regardless of the downstream conditions. The pressure developed is dependent on the resistance of the

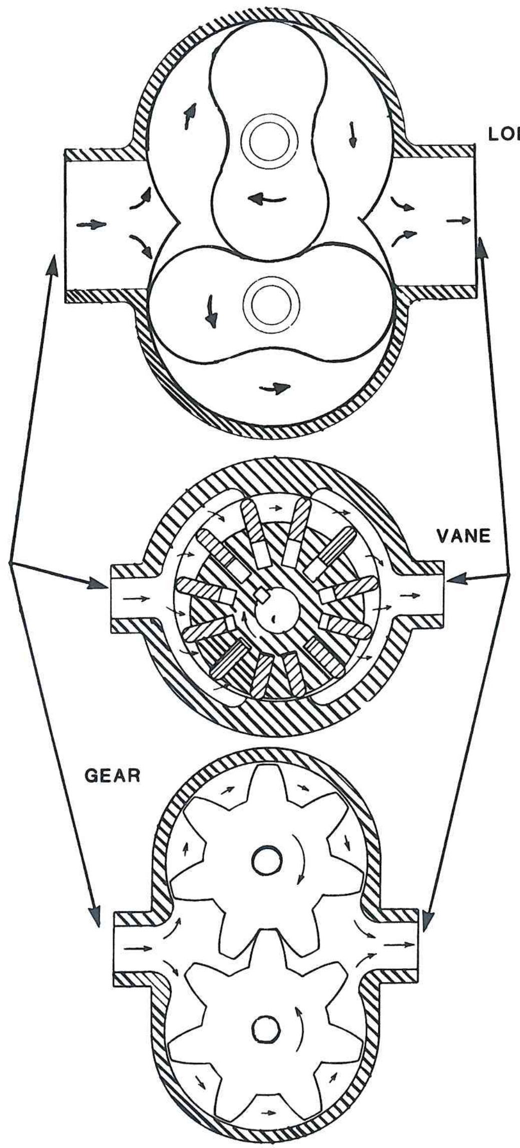


Figure 5.6 Rotary pump configurations.

discharge piping. The resistance includes the effect of oil jet orifices, pipe friction, elbows, and so on. Allowable pump discharge pressure is limited by the strength of the pump and the power of the driving unit.

Other types of pumps, such as centrifugal or piston, are not used on gearbox applications, for the following reasons:

Centrifugal pumps develop pressure as a result of centrifugal force and are used mostly where large flow volumes at relatively low pressures are required. The flow delivered varies considerably with changes in discharge pressure, and centrifugal pumps are not capable of self-priming.

Piston pumps are insensitive to discharge pressure and are self-priming; however, due to their reciprocating motion and the inertia effects of the moving parts, speeds are relatively low; therefore, pump capacities are low compared to rotary configurations. Also, the flow tends to pulsate in a piston pump configuration.

Rotary pumps in gearboxes can be flange mounted to the unit and driven by one of the gearbox shafts or independently mounted with an electric motor or other prime mover driving. When shaft driven, the flow will vary directly with shaft speed. Commercial pump speeds rarely exceed 1800 rpm in gear, vane, or lobe pump configurations, although some small aircraft types achieve much higher speeds. There are screw pump designs which can achieve speeds to approximately 5000 rpm. This is because the flow through the pump is axial and fluid speed is relatively low compared to the peripheral speeds of gear or vane pumps. Figure 5.7 is a schematic of a screw-type pump.

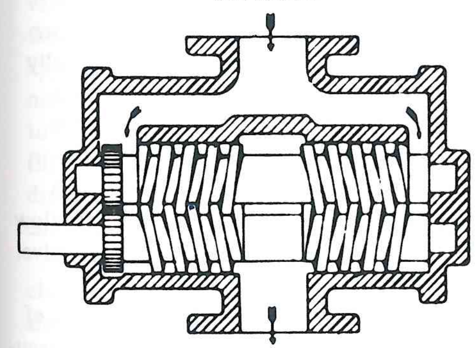


Figure 5.7 Screw pump configuration. (Courtesy of Delaval IMP Pump Division, Trenton, N. J.)

The pump discharge pressure will depend on the required gearbox feed pressure and the pressure drop between the pump outlet and gearbox inlet. This pressure drop is a result of the losses in the filter, cooler, valves, lines, and so on, in the system. The pressure drop can be considerable and care must be taken that the pump design pressure is not exceeded. The great majority of commercial pumps are designed for pressures of 150 psi maximum, although much higher pressure capacity is possible. A pressure relief valve can be incorporated downstream of the pump outlet to protect the pump against overpressure in case of downstream blockage.

The pumping action of all rotary pumps is essentially the same. On the inlet side a void is created as the pumping elements rotate. Fluid, forced by atmospheric pressure, fills this space and is transported to the outlet side. In a gear pump, as the gears rotate, fluid is trapped between the gear teeth and the case, and is carried around to the discharge. In a vane pump, the rotating member with its sliding vanes is set off-center in the casing. The entering fluid is trapped between the vanes and the inside of the case and is carried to the outlet. The term "positive displacement pump" means that with each revolution a specific volume of fluid, depending on the geometry of the elements, is passed through the pump. If there were no clearance between the rotating elements, or between the rotating elements and the casing, the volume of fluid pumped could be easily calculated and predicted. Clearance does exist, however, and depending on the discharge pressure, there is always some internal leakage from the outlet to the inlet side of the pump. This leakage or volumetric pump efficiency must be considered when designing a pump for a specific application. Another consideration is the range of lubricant viscosity the pump will experience. Maximum internal leakage will occur when the fluid is at minimum viscosity; thus the capacity of the pump must be sized for this condition. A gearbox oil pump should be specified to be oversized at least 15% as far as flow requirement is concerned to account for operating variables and pump deterioration over time.

To control the flow into the gearbox a pressure regulating valve is usually incorporated at the unit inlet. The valve is set to maintain a constant inlet pressure, which if the gearbox orifices are correctly sized, will result in the design flow. As the oversized oil pump delivers more flow than required, the pressure will exceed the set point and the regulating valve will bypass enough flow to maintain the design pressure into the gearbox. Figure 5.8 shows the flow versus speed characteristic of a shaft driven oil pump with a pressure relief valve. From startup to point A the flow varies directly with speed. At point A the pressure, which varies with the square of the speed, has reached the set point of the relief valve and the valve cracks open. The flow into the gearbox is then regulated at a constant amount with the valve bypassing sufficient flow to keep the feed pressure constant.

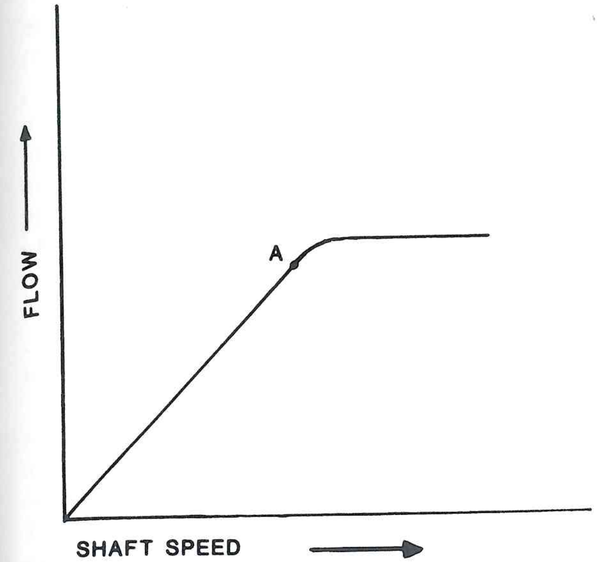


Figure 5.8 Pump flow versus speed characteristic using pressure relief valve.

A significant parameter in designing the lubrication system is the pump suction requirement. Fluid flows into a pump due to the difference in pressure between the pump inlet and fluid source. In most cases the fluid source is at atmospheric pressure, so there is 14.7 psia pressure available to force the lubricant into the pump. Tending to retard the lubricant flow are the static lift (maximum level of fluid below the pump inlet), friction losses due to pipes, valves, elbows, and so on, and entrance losses from the reservoir to the pipe and the pipe into the pump inlet. Commercial rotary pumps generate suction lifts of approximately 5 to 15 in. Hg; 30 in. Hg is equivalent to 14.7 psia; thus commercial pumps can withstand an inlet line pressure drop of approximately 2.45 to 7.35 psia. If the pressure drop in the inlet line is excessive, the pump will not fill with oil and as a result oil flow and discharge pressure will be erratic and drop off. Also, air bubbles will be formed in the fluid and as they implode, cavitation damage may occur, with the potential of pump failure.

To minimize inlet line pressure drop, the reservoir should be located as close to the pump as possible and not too far beneath it. Line and fitting sizes should be as large as possible and bends, valves, fittings, and so on, should be minimized. To help pump priming at startup, the inlet port to the pump should be full of oil. If it is possible for oil to drain down from the inlet port during shutdowns, a foot valve can be placed in the inlet line at a low point. This type of valve has a light spring load which keeps it closed at shutdown and blocks oil drainage from the pump inlet. At startup the pump suction creates a pressure differential across the valve, opening it and allowing flow. In any system when starting up after a long shutdown period it is a good idea to check the pump inlet port to ensure that it is full of oil.

A frequent problem in pump inlet systems is air leaks. Small leaks that are not easily detectable allow air into the line. The air displaces lubricant and the pump discharge pressure and flow will drop off and become erratic. It is difficult to find these leaks but they are quite common and must be considered when troubleshooting pump problems.

The power required to drive an oil pump can be estimated as follows:

$$P = \frac{Q\,\Delta p\,(231\ \text{in.}^3/\text{gal})}{(12\ \text{in./ft})\,(33{,}000\ \text{ft-lb/min/hp})} = 0.0006Q\Delta p$$

where

P = horsepower
Q = flow, gpm
Δp = pressure developed, psi

This is the theoretical power, to which must be added mechanical and viscous losses. The mechanical losses include friction drag of all moving parts, such as bearings, seals, and so on. Viscous losses include the power lost to fluid viscous drag against the pump components as well as shearing of the fluid itself. The mechanical efficiency of a pump is a measure of the magnitude of the horsepower to loss to the theoretical horsepower.

Gearboxes in critical applications often have a shaft-driven main oil pump and an electric motor-driven auxiliary pump. The auxiliary pump is used for prelubrication prior to start up and as a backup in case of failure. Suitable valving must be designed in the system (Figure 5.5) so that the pump flows are properly directed.

FILTRATION

Gearbox lubrication systems are subject to contamination due to a variety of causes. The internal components wear generating particles washed away by the oil stream. Also, at assembly, during maintenance, and even during operation foreign particles find their way into the unit, and these contaminants, if uncontrolled, will cause wear and possibly failure of bearings or gears. Filtration is the mechanism that captures particles in a fluid by passing them through a porous medium. The degree of filtration is generally specified by a micrometer rating:

$$\mu m = 10^{-4}\ \text{cm} = 0.00003937\ \text{in.}$$

As a point of reference a human hair is 70 μm or 0.0028 in. thick.

Typical gearbox filtration specifications call for 40 μm or 0.0016 in. filtration, which can be interpreted to mean that the filter will allow no particle greater than 40 μm to pass. When one considers that journal bearing film thicknesses are on the order of 0.001 in. and less, and gear mesh film thicknesses lower, a case can be made for specifying finer filtration than 40 μm, possibly 10 or even 3 μm. It has been proven that fine filtration is beneficial to component life; however, to obtain full benefits great care must be taken with the entire system or the filter will clog very quickly or be in bypass. For instance, a 55-gal drum of new oil can contain a billion particles 10 μm or larger. Thus new oil should be filtered prior to use in the gear unit. Also, great care must be taken in cleaning all gearbox components and piping prior to assembly. The unit should be flushed with an auxiliary filter prior to use with the standard filter.

The absolute filtration rating of a filter is the diameter of the largest hard spherical particle that will pass through under specified test conditions. To test a filter, a measured quantity of contaminant, typically spherical glass beads in the range 2 to 80 μm diameter, are passed through the test article and captured. The captured fluid is then passed through a very fine membrane filter which is examined under a high-powered microscope. By visual observation, the diameter of the largest glass bead on the membrane is determined.

Filters are sometimes rated by efficiency. One such rating is the nominal efficiency, which is a measure of the retention by weight of a specified artificial contaminant that is passed through a test filter and captured by a membrane filter. Nominal efficiencies are typically in the range 90 to 98%; however, this rating method is losing favor due to lack of reproducibility and uniformity.

Gaining favor in the industry is the beta (β) filtration rating, which is the rating of the number of particles greater than a given size in the influent fluid to the number of particles greater than the same size in the effluent fluid passed through a test filter. For instance, if the number of particles greater than 10 μm per unit volume of fluid entering a filter is 5000 and the number of particles greater than 10 μm per unit volume of fluid leaving the filter is 50, the β_{10} rating is

$$\beta_{10} = \frac{5000}{50} = 100$$

The β rating can be converted to an efficiency as follows:

$$E_x = \frac{\beta_x - 1}{\beta_x}$$

where

E_x = efficiency expressed as percentage of the filter medium's ability to remove particles over a particle size x, by count
β_x = beta filtration ratio (β rating) for contaminants greater than x μm

In the previous example,

$$E_{10} = \frac{100 - 1}{100} = 99\%$$

meaning that the filter will remove a minimum of 99% of all particles greater than 10 μm. The following table gives efficiencies for various beta ratios:

β Ratios versus Efficiency

β Ratio	Efficiency (%)
1	0
2	50
20	95
50	98
100	99
1,000	99.9
10,000	99.99

Figure 5.9 shows how the beta ratio concept can be used. A β_3 filter with a ratio of 1000 will remove 99.9% of all particles greater than 18.5 μm. A β_{10} filter with a ratio of 1000 will remove 99.9% of all particles greater than 26 μm. The β_{10} filter will remove 99.7% of all particles greater than 18.5 μm.

The beta ratio is determined using a continuous flow of contaminated fluid through a test filter. Samples of the fluid from upstream and downstream of the filter are taken and analyzed with an automatic particle counter. This laboratory instrument is capable of determining the particle size and distribution per unit volume of fluid.

Filter media are usually fibrous materials comprised of many fine fibers randomly oriented with diameters ranging from 0.5 to 30 μm. Materials most commonly in use are cellulose, cotton, micro-fiberglass, and synthetics. The smaller the diameter of the fiber used, the closer they can be compacted and the higher the filter efficiency.

When designing a filter, in addition to the filtration rating, the following points must be considered:

Flow rate
Fluid type
Operating temperature
Operating pressure
Dirt-holding capacity
Pressure drop

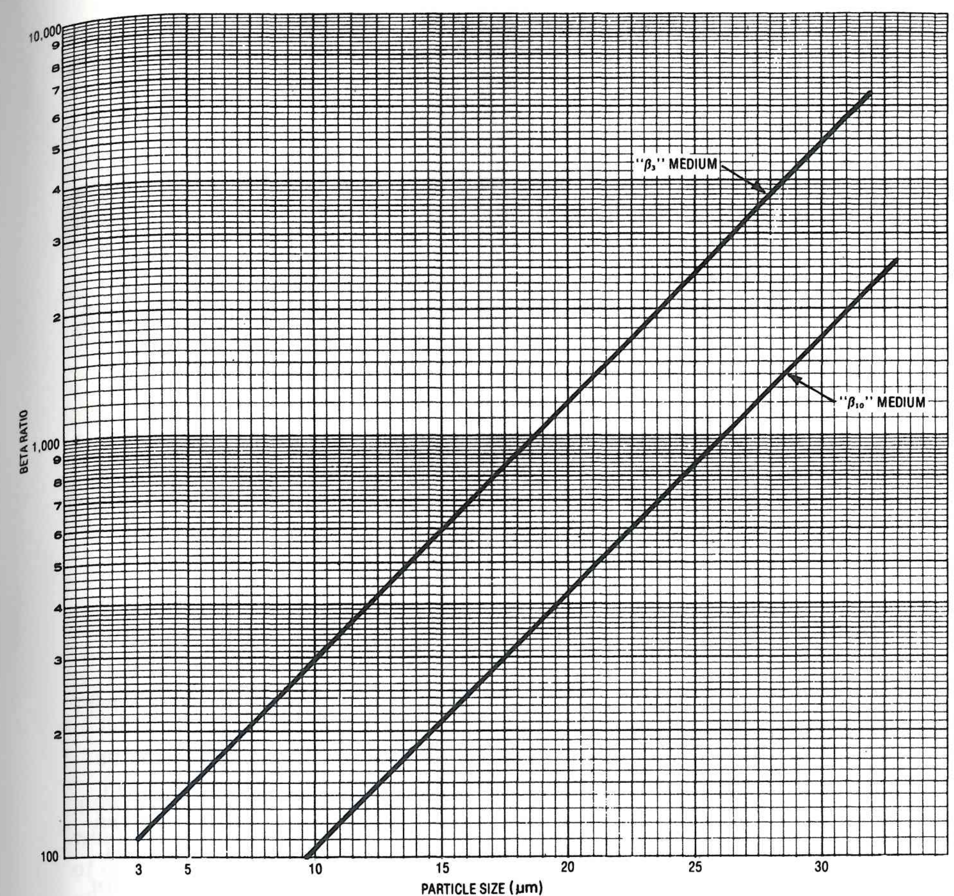


Figure 5.9 Typical beta ratios for β_3 and β_{10} depth-type media versus particle size. (From Ref. 4.)

The flow rate will determine the line size and clean pressure drop for a given size of filter. Fluid type, operating temperature, and pressure will dictate the construction and materials of the filter components. Dirt-holding capacity is an indication of how long an interval may be expected between filter changes.

As the filter collects contaminants during operation, the pressure drop across the filter will increase. Filter elements have collapse pressure ratings and the pressure drop cannot be allowed to exceed this value. Other system considerations such as exceeding the maximum oil pump discharge pressure will also affect the filter pressure drop allowed. In order to monitor the pressure differential across the filter, pressure gauges can be incorporated on the inlet and outlet sides. Filters are available with integral differential pressure indicators that can provide a warning when a set point is reached. To protect the filter and provide a continuous flow of oil, some filters incorporate a bypass valve which will open when a predetermined pressure drop is reached. At bypass, only a portion of the fluid will pass through the filter, allowing unfiltered lubricant to enter the system. For critical applications dual oil filters with changeover valves are specified. With such a design one filter can be serviced while the system is operating using the other filter. Filter elements can either be cleanable and reusable or disposable. Cleanable filter elements are usually made of wire mesh and cleaning is commonly accomplished in an ultrasonic liquid bath. For gearbox applications, in terms of filtration quality, reliability, and ease of maintenance, it is recommended to use disposable elements.

The oil filter should be located on the pressure side of the pump downstream of any component that might produce contaminants, such as the oil cooler. Filters placed in the pump suction line can cause inlet problems as the pressure drop across the filter increases with use.

COOLERS

In a forced-feed lubrication system, the oil in temperature to the gearbox is controlled by passing the hot scavenge oil through a heat exchanger. In order to specify a cooler, the maximum expected oil scavenge temperature must be estimated and the maximum allowed oil temperature into the unit defined. The cooler must be capable of achieving the required oil temperature when exposed to the maximum ambient air temperature anticipated in the application. A generous safety margin should be applied during design to account for deterioration of the cooler. For instance, over time, a water-oil cooler will experience deposits in its tubing reducing efficiency.

The two types of coolers used are liquid to liquid or liquid to air. Figure 5.10 schematically illustrates an oil-water cooler. The hot oil entering through the shell side encounters the cooling water and an equilibrium of temperatures occurs, cooling the oil and heating the water. Where water is not available, radiators are used, blowing cooling air over oil tubes. Air-to-oil coolers require larger envelopes than water-to-oil coolers. Also, on hot days, the air temperature will limit the amount of cooling a radiator can achieve.

OIL RESERVOIR

The oil reservoir or tank may be integral with the gearbox (wet sump) or separately mounted (dry sump) and connected to the gearbox by piping. If

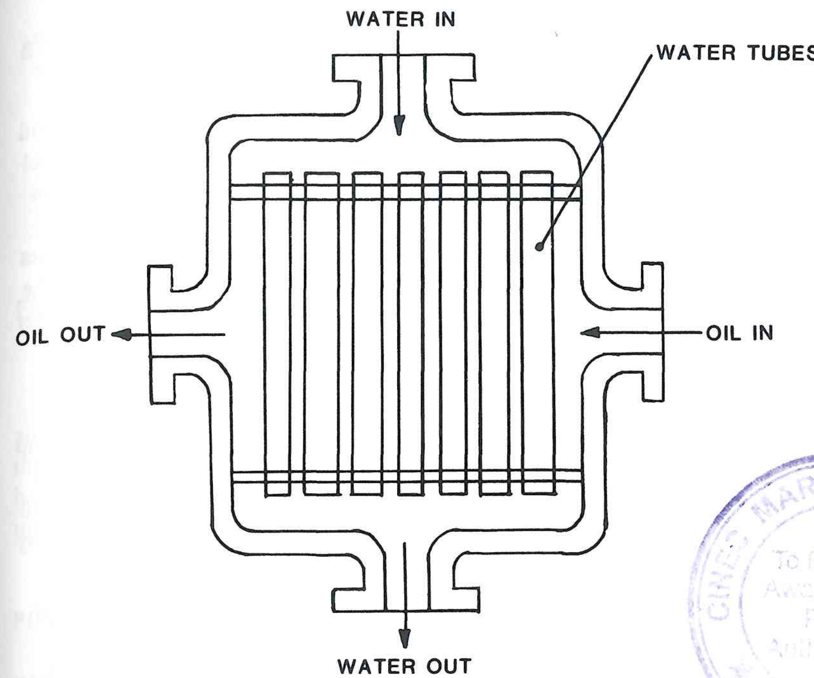


Figure 5.10 Typical oil-water cooler configuration.

integral, on high-speed gearing, the sump should be isolated from the gear windage by some type of baffling so that the oil in the reservoir is still. The level of oil in the reservoir will vary from a maximum when the unit is shutdown and oil has drained from lines and components to the minimum allowed during operation. The minimum operating level is determined by the length of time it is desired to have the oil dwell in the tank to effect deaeration. For instance, if the nominal flow is 20 gpm and a dwell time of 2 min is specified, the minimum oil volume should be 40 gal. The longer the dwell time, the more time air entrained in the oil has to settle out. It is difficult to determine how much dwell time a particular application will require by analytical means. If there is the potential for significant amounts of air to be entrained in the oil, significant dwell time may be required. Air usually enters the lubrication system through pressurized labyrinth seals. Reference 5 specifies an 8-min retention (dwell) time based on nominal flow and total volume below the minimum operating line. This is a conservative value. If there is no likelihood of significant aeration, 2 min of dwell time or less may be sufficient.

After setting the minimum operating level of the oil reservoir an estimate is made of the anticipated oil loss due to leakage and any other reasons and the time interval between oil additions in order to determine the maximum operating level to which the reservoir will be filled. At shutdown, when lines and components such as coolers and filters drain back into the reservoir, the oil level will be higher than the maximum operating level; therefore, the tank must have sufficient volume to accommodate the drain backflow and still retain some air space at the top. At initial startup some quantity of oil is required to fill all lines and components. It is good practice to run for a brief period and then check the oil level to determine if additional lubricant is required to come up to the proper operating level. Oil levels are best monitored by a sight glass and dipsticks are also used.

To ensure complete drainage for cleaning and oil changes the bottom of the reservoir should slope toward a low-point drainage connection. The oil pump suction line should connect slightly above the high end of the reservoir bottom so that sediment on the bottom is not pulled into the pump inlet line. Oil return lines should be piped into the reservoir above the maximum operating level away from the area around the pump suction connection so that the oil around this area is undisturbed. To facilitate inspection and cleaning of the reservoir, sufficiently large openings must be provided.

When cold ambient operation is anticipated, the reservoir can be heated either by applying external heat or incorporating a thermostatically controlled immersion heater in the tank. With immersion heaters care must be taken not to overheat the oil in contact with the element, since this will lead to lubricant degradation. The heater watt density should not exceed 15 W/in.^{2} [5].

BREATHER (VENT)

The gearbox breather is used to vent pressure that may be built up in the unit. Pressure may occur as a result of air entering the lubrication system through seals or the natural heating and cooling of the unit. When a cold gearbox starts up, the heat generated during operation causes air within the case to expand. The same effect is noted with units operating outdoors as they heat up during the day and cool at night.

Breather caps usually have some type of baffling built in to keep particulate contamination out of the gearbox, but moisture can enter. It is possible to have a completely sealed system and incorporate an expansion chamber in place of the breather. Figure 5.11 presents a schematic of an expansion chamber which is screwed into the gear case or reservoir. The expansion or contraction of air is accommodated by flexing of the diaphragm. In harsh environments care must be taken in choosing the diaphragm material such that it is compatible with the environment.

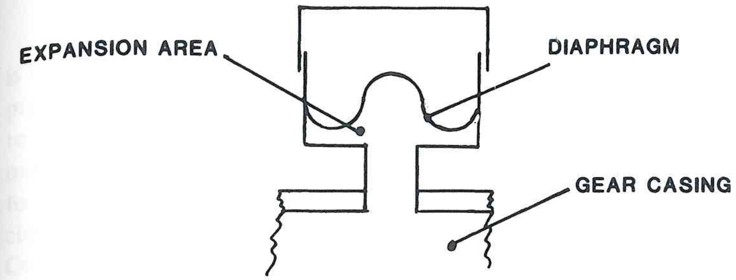


Figure 5.11 Expansion chamber.

In a complicated system the gearbox vent may be inadvertently routed to an area with an unfavorable environment and cause operating difficulties. For instance, if the gearbox is vented to a pressurized area, the gearbox may be back-pressured, resulting in oil leakage through the seals. If vented to an area under a vacuum, oil might be sucked out of the unit through the vent.

PIPING

Although apparently simple, the piping connections for a gearbox can be a source of aggravation at assembly and startup. In many cases the responsibility for supplying piping and lubrication system components is split between the gear manufacturer and the user. The purchase order should be specific as to who supplies what, which specifications apply, and where the interfaces are in order to avoid complications at installation.

As far as gearbox oil feeds are concerned, it is good practice to have only one external connection with all other oil passages placed inside the casing. Then if there is any slight leakage in the piping connections, it will be internal and harmless. Also, there will be less chance of damage to the piping during shipping and assembly. When units are horizontally split, the lubrication piping and components should be mounted on the bottom of the casing such that the top half can be disassembled without disconnecting any lubrication lines.

The piping arrangement must be carefully designed to minimize pressure drops. Care must be taken to position components such as coolers and filters such that they will not drain at shutdown and require refilling at each startup. There are several methods of fabricating lubrication system piping. Reference 5 specifies bending the pipes to suit the required contours and welding them at connections. This is the best way to ensure that there will be no leakage; however, it is more expensive and requires more skill than using threaded joints. The piping terminates in flanged connections which are bolted to the mating part. Seamless carbon steel piping to ASTM A106 or ASTM192 is recommended.

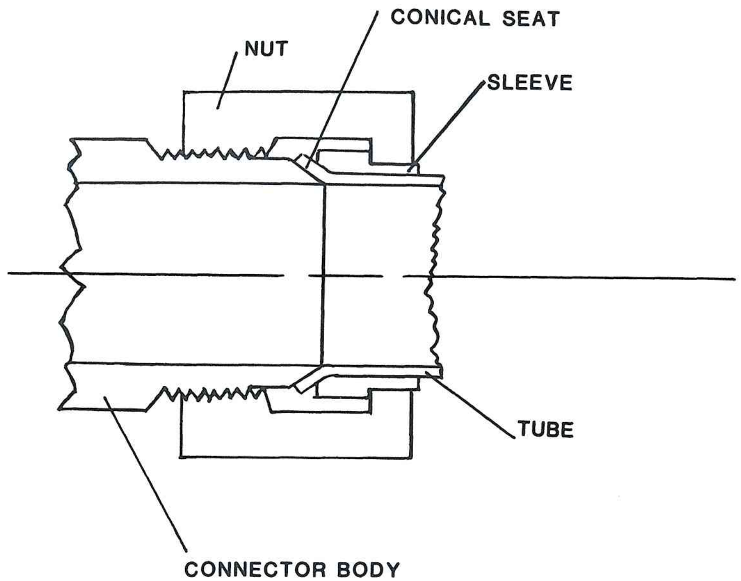


Figure 5.12 Flared fitting.

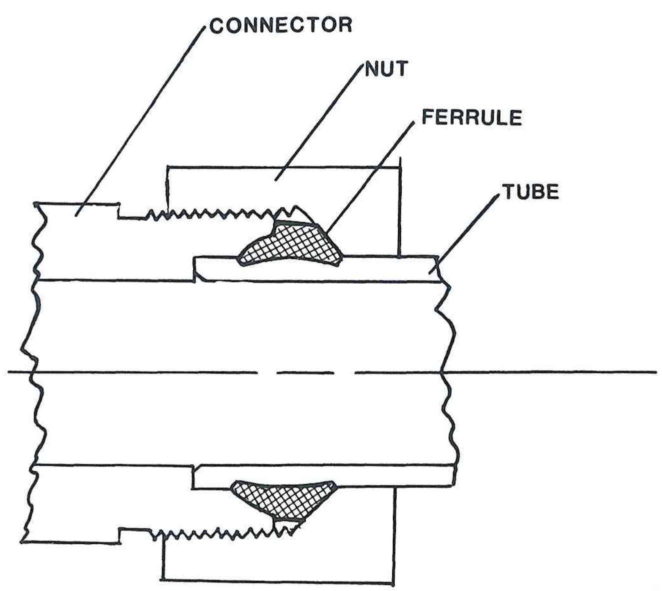


Figure 5.13 Flareless tube fitting.

There are two basic types of threaded joints. One is the pipe thread, which is tapered and produces a metal-to-metal seal by wedging surfaces together as the pipe is screwed in. When connecting this type of joint, sealant should be applied to the male thread. Avoid the first two male threads from the end to keep sealer out of the system. Tetrafluoroethylene-fluorocarbon tape is not recommended to seal pipe threads since pieces may break off and enter the lubrication system, clogging the oil jets. SAE straight thread fittings depend on an O-ring for sealing. Care must be taken in assembly or the O-ring will be damaged, resulting in oil leakage. More positive in stopping leakage than threaded fittings are tube fittings, which fall into three groups: flared, flareless, and the previously mentioned welded or brazed. A typical flared fitting consists of three pieces (Figure 5.12). The seal occurs when the soft tubing is pressed against the hardened conical seat of the connector body. Standard flare angles are 37° and 45° and should never be mixed.

Flareless fittings are usually used where tube thicknesses make use of flared tubes difficult. Figure 5.13 illustrates a bite-style flareless tube fitting where the wedging action of the ferrule provides the seal.

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6
MATERIALS AND HEAT TREATMENTS

The selection of the gear material and its heat treatment is the most basic decision in the design process. The strength of a gear tooth, as shown in Chapter 3 is proportional to its hardness; therefore, the size, configuration, and cost of a gear unit are dependent on the choice of gear material and its processing. Although there are a wide variety of materials and processes to choose from, practical considerations tend to narrow the options. By this is meant that at any gear manufacturer's facility the type of equipment available for tooth finishing and heat treating will limit the designer's choices. In fact, the equipment available dictates the design philosophy and the material and processing specifications.

The types of gearboxes discussed in this book usually use gears made of alloy steels. These materials carry the greatest load in terms of power transmitted per pound of gear and offer high reliability. Steel is defined as iron with carbon percentages in the range of 0.15 to 1.5. A steel is considered to be an alloy when the maximum of the range given for the content of alloying elements exceeds one or more of these limits; 1.65% Mn, 0.60% Si, or 0.60% Cu; or when a definite range or minimum amount of any of the following elements is specified: aluminum, chromium, cobalt, columbium, molybdenum, nickel, titanium, tungsten, vanadium, or zirconium. Table 6.1 lists the chemical compositions of some of the most widely used gear steels. The American Iron and Steel Institute (AISI) or Society of Automotive Engineers (SAE) numbers designate the steel composition and alloy type. The last two digits signify the carbon content: for example, AISI 4140 contains 0.40% carbon. The first two digits designate the approximate alloy content.

Gears are generally either through-hardened in the range Rc 32 to 43 (Bhn 300 to 400) or surface hardened in the range Rc 55 to 70. The hardness

Table 6.1 Chemical Compositions of Gear Steels

AISI number	C	Mn	Ni	Cr	Mo
4140	0.38/0.43	0.75/1.0		0.80/1.0	0.15/0.25
4340	0.38/0.43	0.60/0.80	1.65/2	0.7/0.9	0.20/0.30
4620	0.17/0.22	0.45/0.65	1.65/2		0.20/0.30
4320	0.17/0.22	0.45/0.65	1.65/2	0.4/0.6	0.20/0.30
8620	0.18/0.23	0.70/0.90	0.40/0.70	0.4/0.6	0.15/0.25
3310	0.08/0.13	0.45/0.60	3.25/3.75	1.4/1.75	
9310	0.08/0.13	0.45/0.65	3/3.5	1/1.4	0.08/0.15
2317	0.15/0.20	0.40/0.60	3.25/3.75		

range between Rc 43 and Rc 55 is seldom used since the steel is too hard to cut in this state. Therefore, it may as well be fully hardened to obtain maximum load capacity and finish ground if necessary.

The strongest and most durable gear meshes are made up of two surface-hardened gears using the carburizing process. Utilization of surface-hardened gears will result in the minimum gearbox size, creating savings in materials, machining, and handling costs. On the other hand, there are increased costs associated with carburized gears. Because of heat-treat distortion, griding after hardening will be necessary to achieve high precision of the gear teeth. The grinding and heat treating equipment is very costly and may not be readily available. From a technical point of view, however, carburized, hardened, and ground gears are the best alternative for the high-speed, high-power drives. Of the materials listed in Table 6.1, the low-carbon steels–4620, 4320, 8620, 3310, 9310, and 2310–are carburizing grades.

Another surface-hardening process that is used extensively is nitriding. This process produces a very hard, wear-resistant case which is somewhat more brittle than that of a carburized gear; thus the fatigue properties are not as good. Heat-treat distortion resulting from nitriding is not as great as that from carburizing; thus it is sometimes possible to use gears after nitriding without a grinding operation. Even if grinding is planned, nitriding is sometimes used to limit distortion. It is not uncommon to mesh a carburized and ground pinion with a nitrided gear. When the gear is large, grinding is often considered impractical either because equipment is not available or simply because the cost is too high. AISI 4340 and 4140 are nitriding steels.

Another common material combination is a surface-hardened pinion meshing with a through-hardened gear. Although the gear strength and durability is considerably lower than that of the pinion, the gear will experience fewer stress cycles and a reasonable fatigue life can be achieved. The hardened pinion may tend to wear the softer gear in improving mesh characteristics and also cold working the gear, which increases its surface hardness. The through-hardened gear is considerably cheaper than a surface-hardened gear since the heat treatment is much simpler and no grinding is required.

The least expensive mesh combination is a through-hardened pinion mating with a through-hardened gear. The size of such a unit will be large compared to a gearbox incorporating surface-hardened gears: as much as twice the envelope and weight; however, where there are no restrictions on size or inertia, this type of design is widely used. Most double helical units have through-hardened gears. Of the materials listed in Table 6.1, AISI 4140 and 4340 are through-hardening steels.

There are other methods of surface hardening in addition to those mentioned above, such as induction or flame hardening. Also, other materials are in use, such as AISI 1040, 1045, 1137, and 1340. These are plain carbon steels where the alloying element content is low. These materials and processes are economical and satisfactory for gears requiring only a moderate degree of strength and impact resistance. Where cost is the primary consideration, these options should be considered.

HARDENING FUNDAMENTALS

Steel, as received from the mill, has a structure of nearly pure iron (ferrite) and iron carbide (cementite). To harden a part it has to be heated in the range of 1450 to 1600°F, where it takes the form of austenite, a structure of iron and carbon stable at high temperatures. Depending on how quickly the part is cooled, the austenite transforms to other structures. Figure 6.1 illustrates the formation of the various structures of carbon steel due to quenching. If cooled very rapidly, a hard, strong, and somewhat brittle form called martensite, which is carbon dissolved in iron, is formed.

The maximum hardness that any steel can attain after quenching is a function of its carbon content. Figure 6.2 illustrates this point. The curve reflects 100% transformation of austenite to martensite. From the curve it can be seen that the through-hardening, high-carbon steels 4140 and 4340 can attain a maximum hardness of Rc 58 after quenching. Gears are not used in this state since a fully quenched martensitic structure lacks the touchness to be able to resist shock loading. To improve the gear tooth's ductility and toughness its structure is modified by tempering, a controlled reheating of the part to a temperature below the austenitizing temperature. This process also reduces residual stresses left in the gear during hardening. Tempering temperatures as low as 300°F can produce significant increases in toughness and ductility and reductions in residual stresses with small hardness decreases.

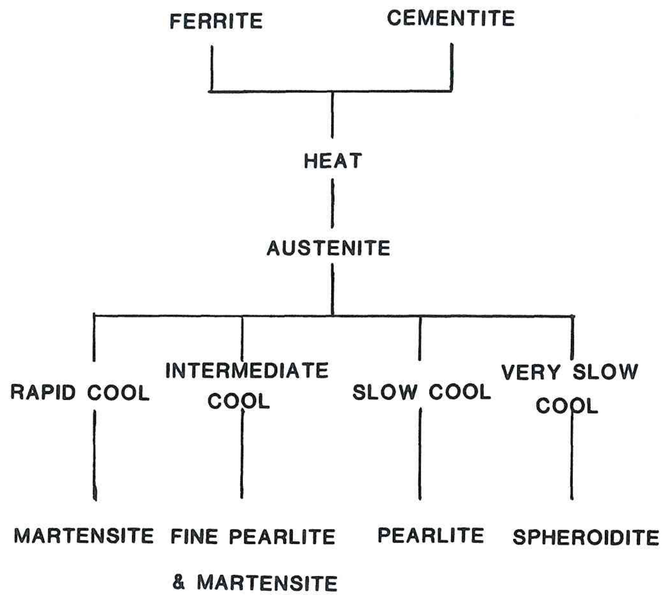


Figure 6.1 Formation of steel structures during quenching.

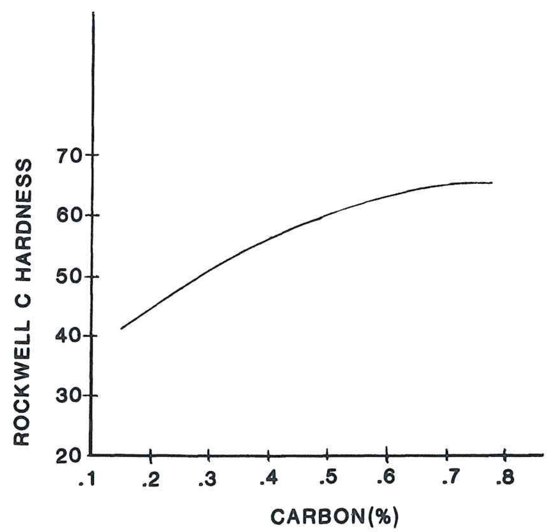


Figure 6.2 Relationship of carbon content to hardness.

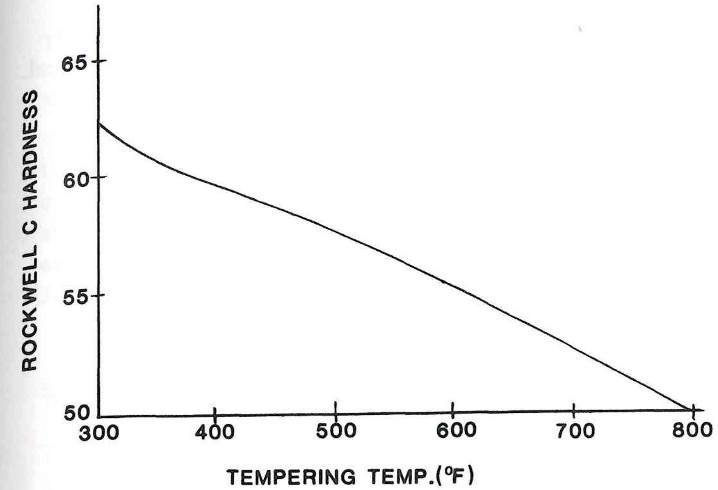


Figure 6.3 Reduction of steel hardness due to tempering.

In the case of carburizing steels which initially have low carbon levels, the tooth surface is enriched with carbon to approximately 0.85% and thus can attain a surface hardness of approximately Rc 65 maximum. The core will remain relatively soft, about Rc 35; thus the carburized tooth has a strong hard case with a tough ductile core. This structure has been found to be optimum for load transmisision. The carburized gear, like the through-hardened gear, must be tempered after hardening. Figure 6.3 shows the surface hardness drop-off of an AISI 9310 carburized gear at various tempering temperatures. The surface hardness was Rc 65 in the as-quenched condition.

HARDENABILITY AND THE USE OF ALLOY STEELS

An alloy steel is chosen on the basis of the mechanical properties required in the part and the heat treatment available to develop these properties. The characteristic of a steel that will determine if it is acceptable is called hardenability. Hardenability is a measure of the severity of cooling conditions necessary to achieve the hardness required. Hardenability of an alloy steel is established by the carbon and alloy content. To obtain a given hardness, higher-hardenability steels require less severe quenching than do low-hardenability steels. A less severe quench will result in lower distortion of the part and less chance of residual stresses, which can manifest themselves in cracks during processing or gear operation.

For the same section thickness a steel with good hardenability can be hardened throughout with a far less severe quench than a low-hardenability steel. For instance, a 1-in.-diameter AISI 1045 steel bar fully hardened at the outside when quenched in water will have a hardness drop-off at its center of 5 points Rc. A 1 in. diameter AISI 4340 bar will have negligible hardness drop-off through its cross section after water quenching.

In order to achieve lower cooling rates and deeper hardening properties, alloys are incorporated in the steel formulation. The following elements are added to alter the properties of steel:

Carbon. The principle hardening element in steel
Manganese. A strengthening and hardening element
Nickel. Improves low-temperature toughness and ductility, reduces distortion and cracking during quenching, and improves corrosion resistance
Chromium. Increases hardenability and forms hard, wear-resistant carbides
Molybdenum. Improves hardenability

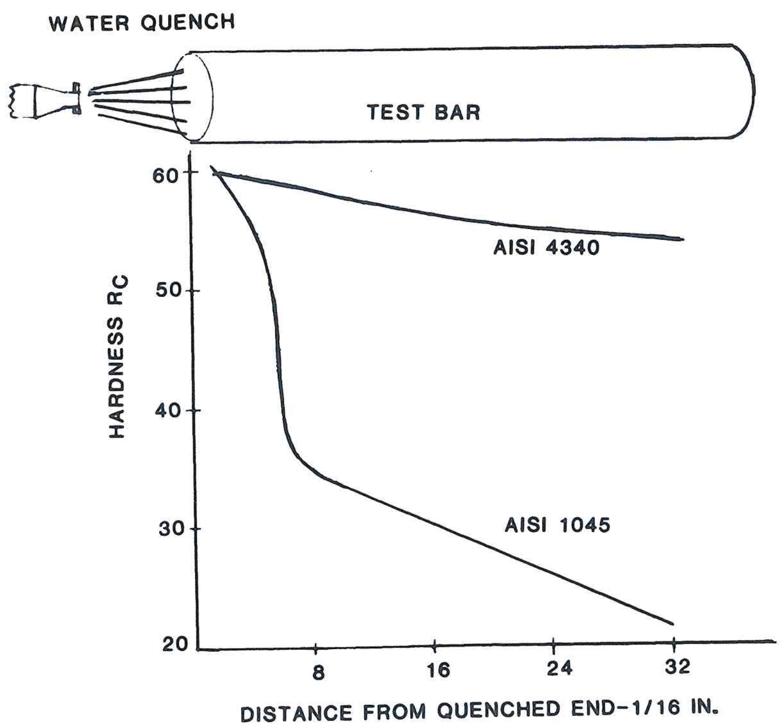


Figure 6.4 Jominy test for hardenability.

The standard method of determining a steel's hardenability is the Jominy bar test. A vertically suspended bar 1 in. in diameter and 4 in. long is heated above its critical temperature into the austenitizing range. It is then water quenched at one end and air cooled at the other end. Hardness is measured along the bar and the gradient between the fully hardened water-quenched end and the softer air-cooled end represents the hardenability of the steel. If the steel has high hardenability, the hardness will be at a higher level farther from the water-quenched end than that of a low-hardenability steel. Figure 6.4 illustrates the results of a Jominy bar test.

STEEL QUALITY

The AISI designation of a steel specifies only the chemical composition of the material. The quality in terms of freedom from seams, cracks, folds, inclusions, or nonhomogenous structure may not be adequately controlled when specifying a material in this manner. When the best quality is required, steels should be produced by the electric furnace process rather than the basic open hearth or oxygen process. Macroscopic, microscopic, and magnetic particle inspection should be required to determine material structure and cleanliness. For critical, high-reliability applications it may be necessary to specify that material be produced by vacuum or double vacuum melting to assure adequate material cleanliness.

The Society of Automotive Engineers issues a series of Aerospace Material Specifications (AMS) that control the quality of steels. For instance, AMS 6260

Table 6.2 AMS Specifications for Gear Steels

AISI (SAE) designation	AMS designation	Remarks
4140	6382H	
4340	6414B	Premium quality, cons. elec. vacuum melted
4340	6414H	
4620	6294D	
4320	6299B	
8620	6276D	Premium quality, cons. elec. vacuum melted
8620	6277B	Premium quality, cons. elec. melted
3310	6250G	
9310	6260H	
9310	6265D	Premium quality, cons. elec. vacuum melted
9310	6267B	Premium quality
2317	—	

covers air-melt AISI 9310 steel and AMS 6265 is the vacuum-melt version. The AMS numbers corresponding to the steels in Table 6.1 are given in Table 6.2.

The steel melting process is critical in controlling material cleanliness. Air melt is the standard procedure. Premium-quality steels undergo vacuum degassing where the billet is heated to a temperature slightly below the melting point in a vacuum environment and hydrogen and oxygen are removed from the steel. This process minimizes nonmetallic inclusions such as aluminum oxide and silicon oxide in the material. A step further in producing clean steel is to melt the material one or more times in a vacuum so that impurities are removed as gases. It has been shown that vacuum melt steels yield significant improvement in gear reliability by minimizing the possibility of fatigue crack initiation due to material impurities. Specifying an AMS steel may add to the cost of a gear initially but in many instances can save the manufacturer the cost of scrapping a semifinished part when material defects are uncovered during machining operations.

PROCESSING OF THROUGH-HARDENED GEARS

Steel, as obtained from the mill, either in an as-forged or rolled condition may not be uniform with high- and low-hardness areas. The first metallurgical operation to be performed is annealing, which involves slow heating, austenization of the steel, and then slow furnace cooling. Annealing will improve machininability by providing a uniform low-hardness structure. This facilitates the initial rough machining.

The part may then be normalized, which involves heating above the austenitizing temperature and then air cooling at an intermediate rate. The resulting structure is somewhat harder, stronger, and less ductile than when in the annealed state. The gear is finish machined at this point except for the teeth and bearing journals. The normalized structure facilitates precise machining and results in a good surface finish.

The part is then hardened and tempered to the design requirement. At this point the gear teeth are cut and the bearing journals finished. Another temper is performed to relieve machining stresses and possibly a finishing operation is required on the teeth such as shaving or lapping. The journals might also require refinishing.

CARBURIZING

In the carburizing process carbon is diffused into the surface of the teeth by controlled exposure at temperatures of 1650°F or above for the length of time

necessary to achieve the desired case depth. After hardening the tooth has a high surface hardness, decreasing to a specified core hardness which depends on the carbon and alloy content of the particular steel used. Carburizing produces the strongest tooth with respect to bending and pitting resistance by generating residual compressive stresses in the case area. The hard surface has excellent wear and scoring resistance.

Before carburizing, the gear teeth are cut and other areas that are to be hardened are machined. Areas of the part that will remain soft must be insulated from the carburizing medium. This may be accomplished by copper plating or other masking procedures or by leaving excess stock to be machined after heat treating. In some cases a carburized area can be machined after hardening if it is to remain soft. After carburizing the part is hardened by quenching. The process will cause distortion in the part; therefore, to attain precise tooth geometry the gear teeth must be ground after hardening. It is important to minimize the distortion so that the stock removal after hardening is small since the grinding process is removing the hard case that is desired. In many cases the gear must be quenched in a die so that distortion can be kept within acceptable limits.

During the carburizing process carbon is introduced in liquid or gaseous form. The most common process now in use is gas carburizing, where a controlled gas surrounds the part in a sealed furnace. The amount of time required to carburize a part is dependent on the temperature; however, carburizing above 1800°F tends to coarsen the grain size to an unacceptable level.

The optimum carbon content to achieve maximum hardness is 0.080 to 0.090%. This is achieved by controlling the richness of the carbon medium. The case depth is controlled by taking sample coupons out of the furnace at intervals during the cycle. When the coupon achieves the required case depth the carburizing cycle is stopped.

Table 6.3 Recommended Case Depth versus Diametral Pitch for Carburized Gears

Diametral pitch	Light case (in.)	Standard case (in.)	Heavy case (in.)
4	0.041	0.053	0.064
6	0.030	0.040	0.049
8	0.028	0.032	0.040
10	0.020	0.027	0.034
20	0.010	0.014	0.018

Source: Ref. 1.

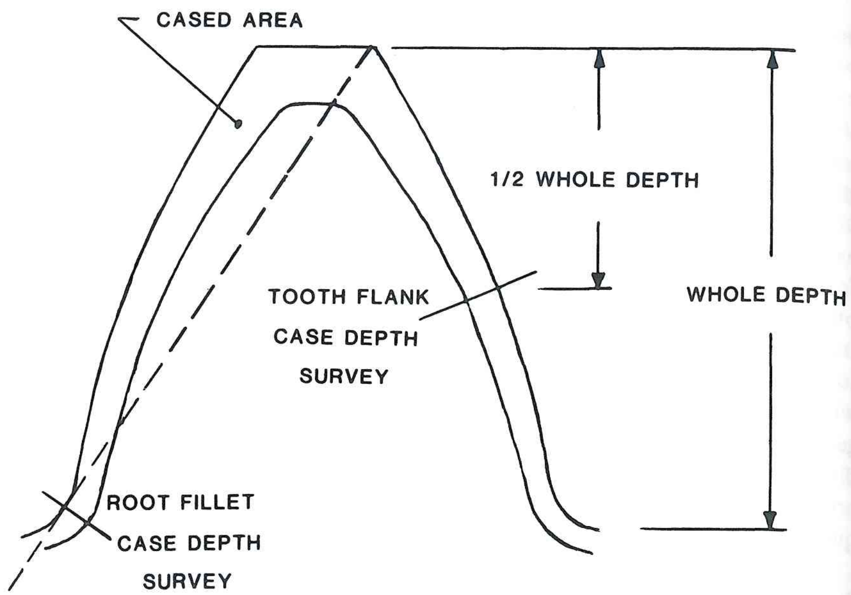


Figure 6.5 Case depth measurement.

The effective case depth is defined as the depth at which the hardness of the case is Rc 50. The case depth must be sufficiently deep to withstand the stresses created by the transmitted load, yet not so deep that the tip of the tooth becomes brittle with a tendency to break off. For this reason it is recommended that tooth tips and end faces be masked off during carburizing and left at core hardness. The case depth required will vary according to the tooth size or diametral pitch. Table 6.3 gives recommended case depths for various diametral pitches.

The case depth is measured by microhardness-testing a cross section of a tooth. This requires sacrificing a production piece or including a test section of gear teeth with the heat-treated lot. The case depth measurement is accomplished in the laboratory using an instrument which measures the hardness from the surface inward every 0.001 in. The hardness survey should be made at the tooth midheight perpendicular to the tooth surface. The root fillet region should also be checked since it is harder for the carbon to penetrate this closed-in area. Figure 6.5 illustrates the areas of the tooth where the survey should be taken. It is not uncommon for the root hardness and case depth to be somewhat lower than the flank, but both should be within design requirements.

In addition to the case depth, the engineering drawing should specify the following:

Minimum case hardness or range. The maximum carburizing surface hardness attainable after grinding is approximately Rc 63; therefore, the most stringent practical drawing requirement is Rc 60 to 63. In many cases Rc 58 minimum is specified and for moderately loaded gears Rc 55 minimum may be acceptable.

Core hardness range. Core hardness should be held within the range Rc 30 to 40. Higher core hardnesses may make the tooth too brittle, and lower values have insufficient strength. It is difficult to control core hardness closely since this parameter is dependent on the base steel carbon content, which varies from one heat value to another. The process controls are directed toward achieving case hardness and the core hardness falls where it will.

There are variations possible in the process used to carburize gears which involve a trade-off between cost and the quality of the metallurgical structure obtained. Table 6.4 shows a carburizing process for a critical 30-tooth, 10-pitch gear with a 1.0-in. face width. This is an elaborate process designed to achieve the best possible metallurgical structure. Note that following carburizing there is a subcritical annealing step the purpose of which is to refine the case structure. This step is sometimes omitted and the part is cooled down to 1500°F and quenched directly from the carburizing furnace. If this is done, care must be taken during the cooling to surround the gear with a carbon atmosphere so that decarburization of the part does not occur. A similar problem arises in the subcritical annealing step. If an air atmosphere is used, the part must be copper plated so that scale formation does not occur during annealing or decarburization during the subsequent hardening.

The hardening process involves heating the gear until the part is completely austenitized and quenching in an oil bath at 80 to 140°F. The objective of the quench is to transform the austenite into martensite; however, in practice, all the austenite can never be transformed. This is undesirable since the retained austenite is not stable and if present at too high a level can spontaneously cause distortion in the part. In order to transform as much of the retained austenite as possible, carburized parts are sometimes subzero cooled in the range -100° to -150°F. This cooling must be accomplished within 20 min of quenching, to avoid austenite stabilization. The minimum volume of retained austenite that can be practically achieved is 10%. For applications that are not extremely critical, 20% can be allowed. Retained austenite is measured definitively by x-ray diffraction or (visually estimated by) metallographic examination of an etched cross section.

Carbides in the case structure are desirable for increased strength and wear resistance, however, if they form continuous or semicontinuous networks at the grain boundaries, stress concentrations result which can initiate crack.

Table 6.4 Carburizing Process for AISI 9310 Steel

Process operation	Temperature (°F)	Time at temperature (hr)	Type atmosphere	Type of Quench	Quench temperature (°F)	Remarks
Normalize	1725	2	ENDO[a]	Chamber	R.T.[b]	
Harden	1500	1	ENDO	Oil	130	5 min in oil
Temper	1050	4	Air	Air	R.T.	
Carburize	1700	As required	Carbon potential	–	–	Furnace cool to 1500°F, air cool to R.T.
Subcritical anneal	1150	2	Air			
Harden	1500	1	ENDO	Oil	130	Agitated oil
Freeze	-120	1		Liquid		Thaw to R.T.
Temper	300	4		Air		

[a]Endothermic atmosphere.
[b]R.T., room temperature.
Source: Ref. 1.

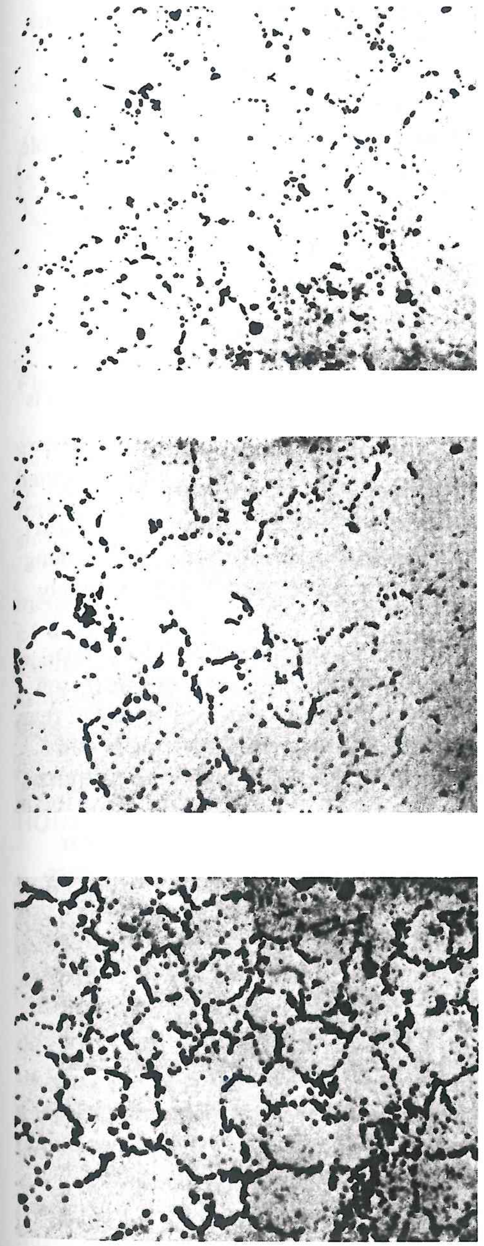


Figure 6.6 Typical metallographic standards for undissolved case carbides. (From Ref. 1.)

Figure 6.6 illustrates the levels of severity of grain boundary carbides. These are photographs of an etched cross section at 500X magnification. Figure 6.6A shows scattered grain boundary carbides and is the maximum condition acceptable for critical applications. Figure 6.6B illustrates a semicontinuous grain boundary carbide network. This is a marginal condition and might be acceptable for noncritical gearing. Figure 6.6C is a continuous grain boundary carbide network and is not acceptable.

NITRIDING

Gas nitriding is a surface-hardening process in which the part is heated to approximately 1000°F in a furnace with an ammonia atmosphere. Quenching is not required, the hardness developing as a result of the formation of hard nitrides near the surface. AISI 4340 and 4140 are nitriding steels, their chromium and molybdenum alloys combining with the nitrogen to form nitrides. The steel must be hardened and tempered before nitriding to a tempering temperature higher than the subsequent nitriding temperature. The nitriding cycle is quite long. For instance, a 0.020-in. case depth requires approximately 40 hr of processing.

Because the part is not quenched and the process is performed at a relatively low temperature, little distortion occurs during nitriding. For this reason many gears are finished by cutting, shaving, or grinding in the soft state and then hardened by nitriding. For extreme accuracy grinding after nitriding is still required; however, for some large gears the process is still preferred to carburizing because of the limited distortion.

During the nitriding process an outer white layer is formed on the gear teeth which is hard and brittle and therefore undesirable. With special processing the white layer can be held to a maximum depth of 0.0005 in. which is usually acceptable. Thicker white layers are unacceptable because of the possibility of spalling or flaking. They must be removed either by grinding, some other mechanical process or chemical action. Nitriding produces shallow case depths compared to carburizing; therefore, if grinding after nitriding is required, stock removal must be closely controlled to preclude grinding away the case.

FORGING OF GEAR BLANKS

Gears up to approximately 6 in. in diameter can be fabricated from bar stock. Larger gears are forged from bars or billets. Forging is defined as the plastic deformation of a metal at an elevated temperature into a predetermined size or shape using compressive forces exerted through some type of die by a

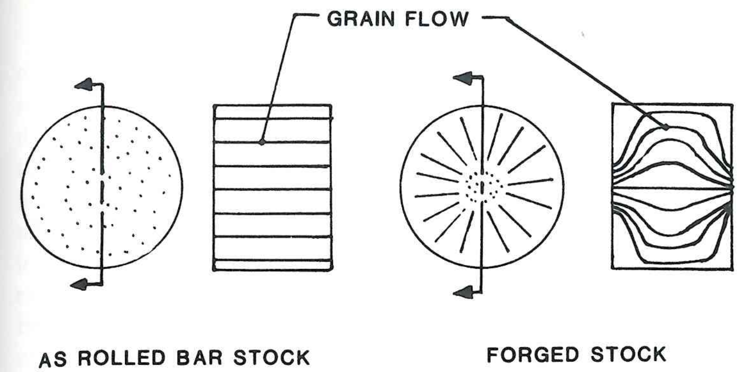


Figure 6.7 Grain flow due to forging.

hammer, press, or an upsetting machine. Many gear blanks are made of upset forgings, where a round bar is struck axially between two flat dies. The bar is shortened and the metal, following the path of least resistance, flows outward.

A forged part will have flow lines which are composed of fibrous nonmetallic inclusions or segregated phases which have been flowed in the direction of working. These fibers, if properly oriented, add to the strength of a part. Figure 6.7 shows the grain glow in a part made from bar stock compared to the same part made by upset forging. The direction of the forged flow lines will improve the bending resistance of the gear that will be fabricated from the bar.

HOUSING MATERIALS

In the design phase of a gearbox the question arises s to whether the housing should be cast or fabricated. A fabrication is comprised of steel plates which are welded or bolted together to form the casing. Usually, a carbon steel such as AISI 1020 is used.

If only one or two units are to be manufactured, the economics favor fabrication. In order to use a casting a pattern must be developed and the cost of fabricating a pattern is significant. Also, if the program is under time pressure, a fabrication can be made in less time than required to develop a pattern and pour a casting.

As the quantity of gearboxes required increases, it becomes more economical to go to a casting. The pattern costs are amortized over the production quantity and the basic material cost of cast iron is lower than steel. Also, less material is used since the casting can be shaped to conform closely to the gear configuration.

There are technical advantages to a casting in that the gearbox weight can be reduced and the casting provides more flexibility to the designer to achieve noise and vibration attenuation.

Cast irons are alloys of iron and carbon containing more than 2% carbon. The types of cast irons most commonly used for gear casings are gray iron and ductile iron. They are specified by American Society of Testing and Materials (ASTM) standards, which define chemical composition and physical properties.

Gray cast iron is graded by minimum tensile strength. For instance, a class 30 gray iron would have a minimum tensile strength of 30,000 psi. Gray iron grades range from 20 to 60. Ductile iron is specified by a three-part system. For instance, an 80-55-06 alloy has a minimum tensile strength of 80,000 psi, a minimum yield strength of 55,000 psi, and 6% elongation in 2 in. Ductile iron classes range in minimum tensile strength from 60,000 to 120,000 psi.

The type of cast iron selected for a given application must take into account not only the physical properties desired but the size and shape of the casting. The solidification rate, which depends on casting geometry and foundry practice, profoundly influences the resulting strength. Actual properties of a cast part will vary with the cooling rate, cross section, microstructure, and the grade of iron used. Like steel parts, cast irons are heat treated for a variety of reasons. These reasons can be summarized under the headings of stress relief, annealing and normalizing, and hardening and tempering.

The stress relief treatment is used because cast irons are susceptible to growth when allowed to stand at room temperature for long periods of time. Stress relief is accomplished at temperatures of 700 to 1300°F. If time permits, castings can be aged for 6 months or more to allow growth to occur and the stress relief can be omitted.

Annealing and normalizing heat treatments are used to modify the castings metallurgical structure to increase machinability. Hardening and tempering, as in steel parts, are applied to improve component strength.

In the casting process discrepancies sometimes occur which leave voids in the material. These can sometimes be repaired by welding. Welding reworks should be inspected by x-ray techniques to ensure a completely crack-free repair.

HARDNESS TESTING

Because so much attention is paid to gear tooth hardness it is appropriate to review the definition of hardness and how this material property is measured. Hardness of a material can be defined as its resistance to permanent deformation. There are several methods to measure this property, the most widely used being Brinell and Rockwell.

In the Brinell test, a load, usually 3000 kg, is applied to the test piece by means of a hardened 10 mm steel or tungsten carbide ball. The resulting impression is measured and the Brinell hardness number (kg/mm^2) is calculated by dividing the load applied by the area of the impression. The Rockwell hardness test uses a diamond cone to indent hard materials. First a minor load of 10 kg is applied to force the penetrator below the surface of the material, following which a major load varying between 60 and 150 kg depending on the scale used is applied. The hardness, which is proportional to the depth of penetration, is read directly. The Rockwell machine is more flexible and easier to use than the Brinell method; therefore, it is widely used for routine testing and the inspection of heat-treated parts. There are two types of Rockwell machines, the Normal tester for relatively thick sections, and the Superficial tester for materials too thin to be tested with the normal instrument. Minor loads on the superficial tester are 3 kg and the major loads vary from 15 to 45 kg. A cone-shaped diamond indenter is used. For work with gear steels the A, C, and D scales on the Normal tester are used. On the Superficial tester the 15-N and 30-N scales are used.

In the laboratory microhardness testing is accomplished using two different types of diamond indenters. One is a pyramid with a square base [diamond pyramid hardness (DPH)] and the other is a rhombic-based pyramid

Table 6.5 Approximate Comparison of Hardness Scales

Brinell 10-mm ball, 3000-kg load	Rockwell 15-N (15-kg load)	Rockwell C (150-kg load)	Rockwell 30-N (30-kg load)	DPH	Knoop
614	90	60	77.5	695	732
587	89.25	58	75.5	655	690
560	88.5	56	74	617	650
534	87.5	54	72	580	612
509	86.5	52	70.5	545	576
484	85.5	50	68.5	513	542
460	84.5	48	66.5	485	510
437	83.5	46	65	458	480
415	82.5	44	63	435	452
393	81.5	42	61.5	413	426
372	80.5	40	59.5	393	402
352	79.5	38	57.5	373	380
332	78.5	36	56	353	360
313	77	34	54	334	342
297	76	32	52	317	326
283	75	30	50.5	301	311

(Knoop). Loads applied are on the order of 5 to 100 kg and the hardness number is proportional to the ratio of the load to a characteristic length of the impression to the second power. The test specimen is an etched, mounted cross section of tooth and the dimensions of the impression left by the indenter are read with a microscope. The DPH numbers correlate well with Knoop results up to a hardness of approximately Rc 58. Beyond this level a Knoop reading of a given specimen referred to the Rockwell C scale will yield a value 1 or 2 points lower than a DPH reading of the same specimen. Table 6.5 gives approximate hardness comparisons for Brinell, Rockwell, DPH, and Knoop. Hardness tests are usually not performed on tooth flanks since the indentation can be a source of stress concentration. On some designs there may be hardened areas which are not functional and can be surfaces for hardness checking. In many cases the gear tooth hardness is verified only by the microhardness survey of the cross-sectioned tooth sample taken from the heat-treated lot. There is nothing wrong with this practice and it is the best way to confirm hardness and case depth in the tooth root.

NONDESTRUCTIVE TESTING

Nondestructive tests are used to detect mechanical defects in material and variations in condition or composition. These tests are performed on production parts and do not impair the component's function.

MAGNETIC PARTICLE INSPECTION

This inspection method is used to identify discontinuities such as cracks, inclusions, or pores which are at or near the surface of a ferromagnetic material. Such discontinuities are areas of stress concentration and can propogate under cyclic loading leading to component failure. Cracks can be a result of poor grinding process or heat treating; therefore, it is good practice to provide for magnetic particle inspection at various stages in the manufacturing cycle. When a defect occurs it is best to identify it as early as possible so that minimum machining time is spent on discrepant material.

Magnetic particle inspection is accomplished by magnetizing a part by passing an electric current through it. Leakage magnetic fields occur at surface or near surface discontinuities. These defects are revealed by their attraction for finely divided magnetic particles which are introduced on the surface of the part being inspected.

The widely used Magnaflux method applies iron oxide particles suspended in kerosene or oil just before or while the part is being magnetized. This is called

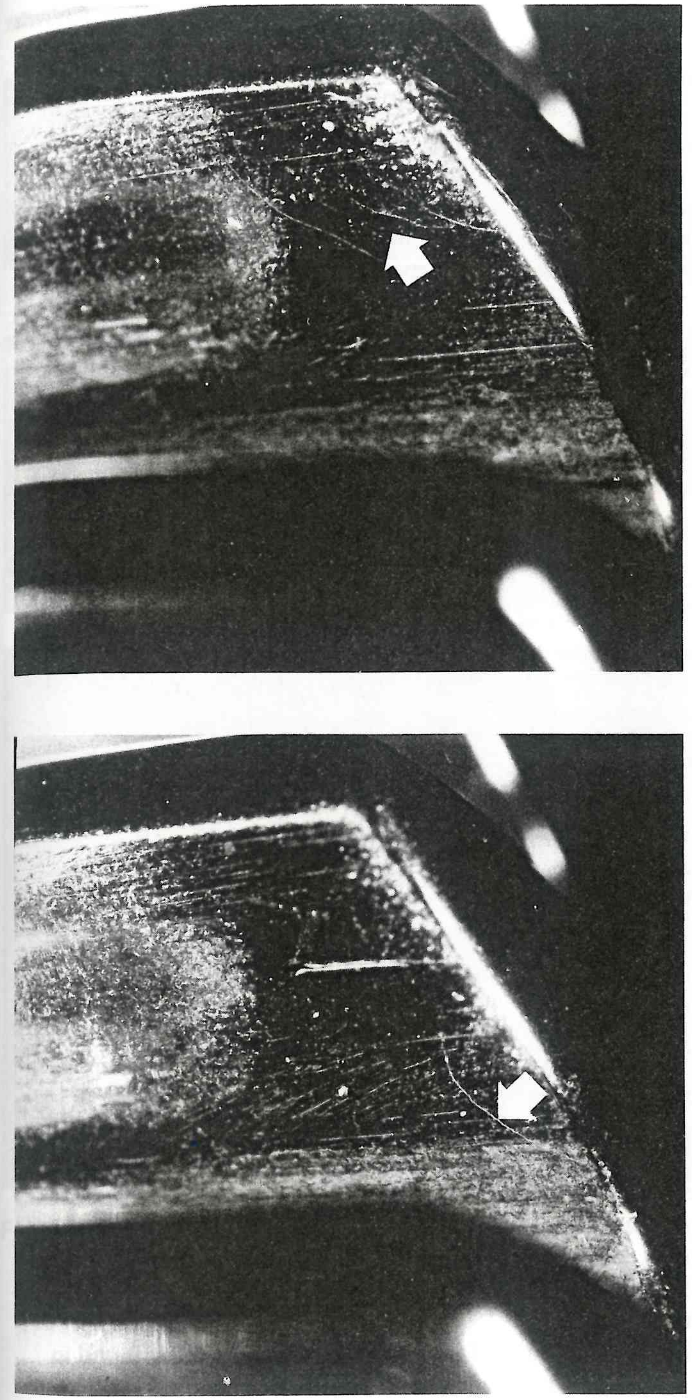


Figure 6.8 Magnetic particle indications (indicated by arrows).

the wet method. There is a dry method where powdered iron oxide is applied while the part is under magnetization. The iron oxide particles cluster about the discontinuity and reveal its approximate location, size, and extent. The black particles are sometimes colored red or gray for better contrast on dark objects. There is a Magnaglo method of inspection where the particles are coated with a fluorescent material which glows when activated by ultraviolet light. This improves the contrast between the iron oxide particles and the part under investigation.

It is important to demagnetize any part that has undergone magnetic particle inspection. If gear teeth or shafts have any residual magnetism, they will tend to attract machining chips or abrasive particles, which can cause difficulty during operation.

To determine whether a part is acceptable following magnetic particle inspection, acceptance and rejection standards must be defined. In general, no indications of discontinuities are allowed on critical areas such as gear or spline teeth. Also, indications that extend over or into an edge, chamfer, corner, radius, fillet, or hole are not acceptable. Often, magnetic particle indications can be removed by localized grinding or remachining. For instance, indications at the edge of a tooth might be removed by regrinding a larger edge radius.

In noncritical areas some indications may be accepted. A typical specification may be as follows:

> A maximum of three indications 3/16 in. long or less are allowed per noncritical area.

Figure 6.8 presents some examples of magnetic particle indications.

LIQUID PENETRANT INSPECTION

Liquid penetrant inspection is a nondestructive method for finding discontinuities that are open to the surface of solid and essentially nonpourous materials. In gearboxes the process is usually used for the inspection of castings. The principle of the liquid penetrant inspection method is to wet the surface of the workpiece with a uniform liquid coating which migrates into cavities that are open to the surface.

A widely used method is the Zyglo inspection, which uses nonmagnetic particles that fluoresce under ultraviolet light. The particles are suspended in oil and applied to the part to be inspected. Excess liquid is removed from the part and the material remaining in any cracks or discontinuities is revealed under inspection by ultraviolet light.

It is difficult to set standards of acceptance for discontinuities found in castings since the shapes are usually unique to the particular application. The location of the defect must be taken into account, and whether it may be

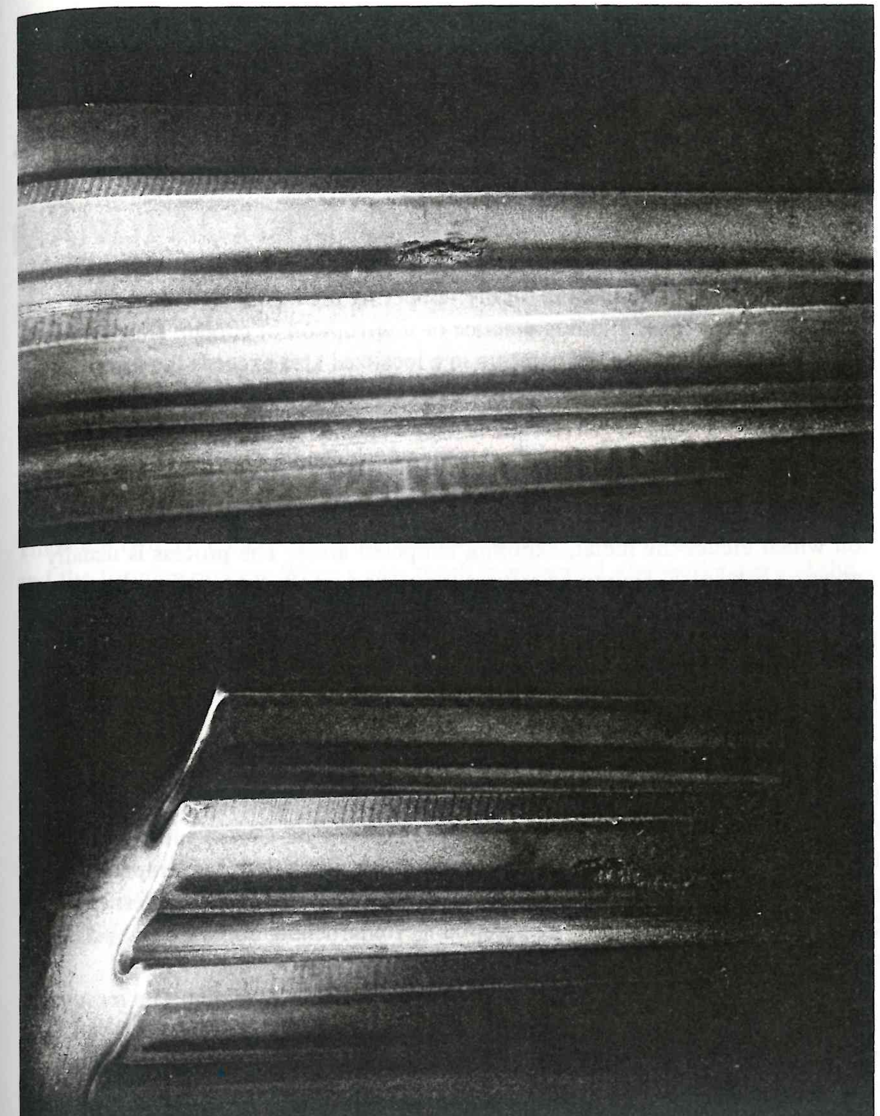


Figure 6.9 Surface temper indications.

detrimental to component strength or serviceability. Minor discontinuities in noncritical areas can be accepted without rework. When there is some question as to whether the discontinuity might propagate and cause distress, it can be removed by localized machining.

SURFACE TEMPER INSPECTION

When hardened components are ground there is the possibility of overheating the surface being machined and locally tempering the overheated areas. This may occur due to improper grinding practice or interruption of coolant flow during machining. If the surface temperature in a localized area exceeds the tempering temperature of the material, the area will soften. In the worst case the local temperature reaches a level sufficiently high to reharden the material, resulting in an area of brittle untempered martensite which can be a source of pitting during operation.

To detect surface temper burns the component is dipped in an acid solution which etches the metal, exposing tempered areas. The process is usually known as Nital etch. Unburned areas of an etched part will be uniformly gray with a dull nonreflective surface. Tempering burns will appear as dark gray or black areas on the etched part. The darkness of the color reflects the severity of the burn. Rehardening burns will appear as white or light areas surrounded by a black tempered area.

In order to set acceptance criteria for tempering, critical areas of the part being inspected must be defined. Also, examples of light temper distinguished by coloration must be available to use as comparisons. Critical gear areas are the loaded flanks of the gear teeth and the root fillet area. Clutch surfaces and bearing surfaces in rolling contact are also considered critical. No temper burns of any degree are allowed in the critical areas. Light temper covering less than 20% of a noncritical area may be acceptable. Rehardening burns are not acceptable in any area.

Because temper is a surface discrepancy, burns can sometimes be removed by remachining the tempered area. In addition to exposing temper, the Nital etch will show up areas that are deficient in carbon or have had excessive stock removal during grinding. These areas will appear light in color when compared to a normal etched surface. Figure 6.9 illustrates tempered gear teeth; the darker color showing the tempered area, which in this case exhibited spalling.

REFERENCE

1. AGMA Standard 246.01A, Recommended Procedure for Carburized Aerospace Gearing, American Gear Manufacturers Association, Arlington, Va., November 1971.

7 MANUFACTURING METHODS

There are two basic methods of manufacturing gear teeth: the generating process and the forming process. When a gear tooth is generated, the workpiece and the cutting or grinding tool are in continuous mesh and the tooth form is generated by the tool. In other words, the work and the tool are conjugate to each other. Hobbing machines, shaper cutters, shaving machines, and many grinders use this principle.

When a gear tooth is formed, the tool is in the shape of the space that is being machined out. Some grinding machines use this principle with an indexing mechanism which allows the gear teeth to be formed tooth by tooth. Broaches are examples of form tools that machine all the gear teeth simultaneously. Gears of the type discussed in this book are initially cut on a hobbing machine or shaper cutter. The tooth forms are further refined by lapping, shaving, grinding, or honing. Following are descriptions of these various processes.

HOBBING

Figure 7.1 illustrates a hobbing machine and Figure 7.2 presents the hob itself. In this process the gear teeth are generated with the hob and workpiece rotating in a constant relationship while the hob is being fed into the work. Hobbing is a versatile and economical method of cutting gears. A hob of any given normal pitch and pressure angle will cut the teeth of all spur and helical gears having the same normal pitch and pressure angle. Hobs producing involute gears are basically straight sided and generate the involute form on the gear tooth by the meshing action.

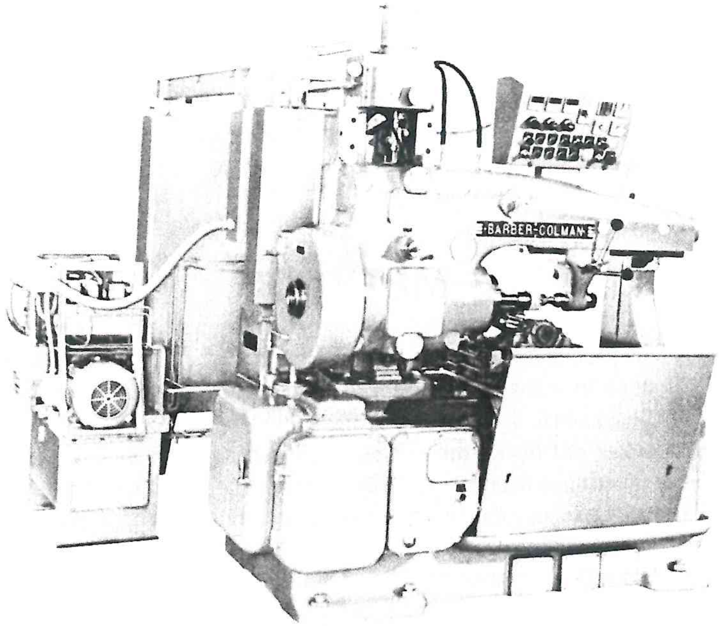


Figure 7.1 Hobbing machine. (Courtesy of Barber Colman Corporation, Rockford, Ill.)

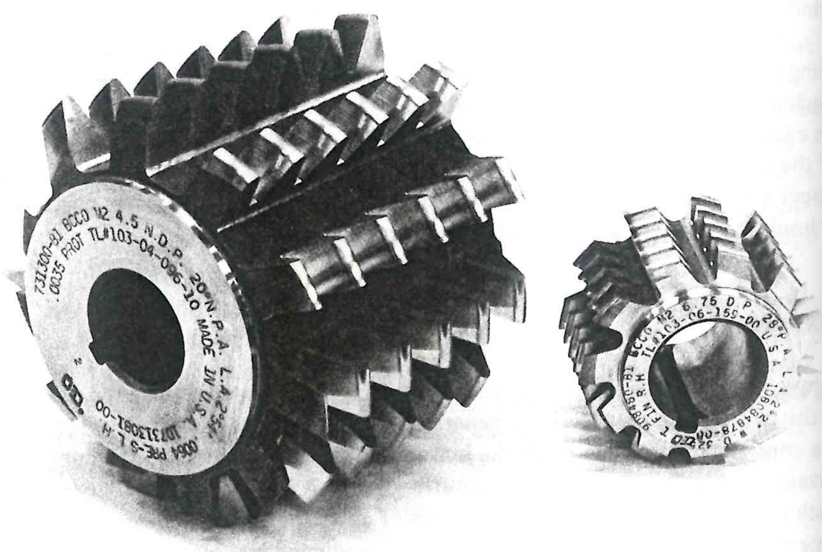


Figure 7.2 Gear hobs. (Courtesy of Barber Colman Corporation, Rockford, Ill.)

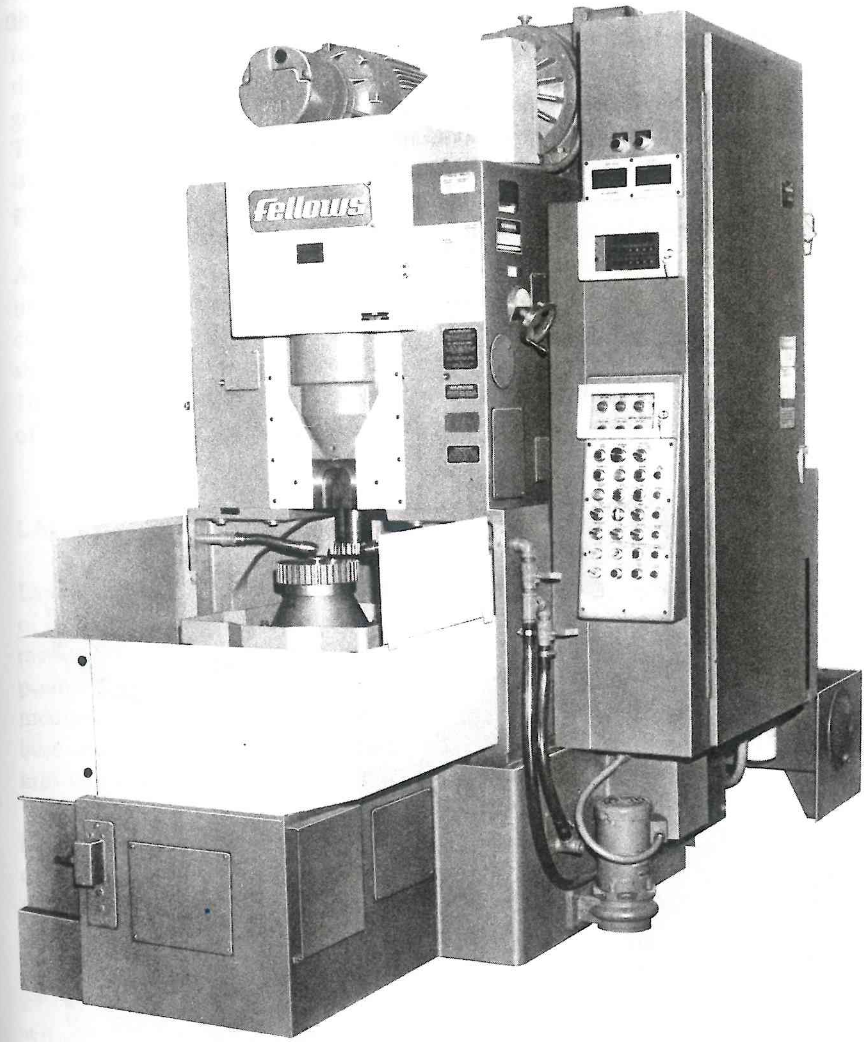


Figure 7.3 Shaper cutting machine. (Courtesy of Fellows Corporation, Springfield, Vt.)

Hobbing produces gears of qualities up to American Gear Manufacturers Association (AGMA) Quality Class 12. To achieve high accuracy the hobbing machine must be in excellent condition, the tooling holding the gear rigid and accurate, the gear blank accurate, and the hob of high quality. All types of materials can be hobbed. The great majority of hobbed gears are in a hardness range Rc 30 to 38; however, with special material cutters hardnesses up to Rc 60 can be machined. Hobbing will produce surface finishes in the range 16 to 64 μin. rms.

The only limitations of the hobbing process are the inability to machine internal gears and a requirement for axial space behind the gear teeth to allow the hob to run out. Gears that are adjacent to a shoulder or cluster gears cannot be hobbed.

Figure 7.4 Shaper cutters. (Courtesy of Fellows Corporation, Springfield, Vt.)

SHAPING

Figure 7.3 illustrates a shaper cutter machine and Figure 7.4 shaper cutters. Shaping is a generating process where the tool is in the form of a shape conjugate to the tooth being cut. When cutting involute gear teeth the shaper cutter is in the form of an involute gear which is hardened and has cutting clearance on the tooth sides. The gear blank and the cutter are rotated in the proper ratio while the cutter reciprocates axially through the gear blank. If a spur gear is being generated, the cutter will reciprocate through the workpiece in a straight path. To generate a helical gear the cutter must reciprocate in a helical motion which is imparted by a helical guide. This additional tooling required to cut helical gears is a disadvantage of shaper cutting compared to hobbing.

Shaping can be applied to internal as well as external spur or helical gears. Also, herringbone gears are cut by the shaping process. The shaper cutter does not require a large runout beyond the gear; therefore, it is a good method for cutting cluster gears or gears close to a shoulder. As with hobbing, the best shaper cutting can produce gears up to AGMA Quality Class 12 with finishes to 16 μin. rms. Hardnesses to Rc 43 can be shaper cut; however, the great majority of gears are cut in the range Rc 30 to 38.

LAPPING

Lapping is a method of correcting small errors in profile, lead, spacing, or runout of gear teeth. A gear can be lapped either by running it with its mating gear or meshing it with a lapping tool in the form of a gear. An abrasive lapping compound is introduced into the mesh to promote removal of metal. The abrasive medium must be uniformly spread across the teeth in an oil- or water-soluble base and should contain a rust inhibitor. Gear laps are usually made of cast iron into which the abrasive will embed.

For most lapped gears the process is short, typically minutes. Only small beneficial changes can be attained by lapping and too lengthy a cycle will usually destroy the profile. Because there is no sliding at the pitch line, lapping will tend to remove material above and below the pitch line only. An auxiliary axial sliding motion is incorporated in lapping machines to overcome this problem.

Wide face helical and double helical gears are often lapped for long periods of time such as several hours to improve their tooth contact. It has been found with wide face gears that have a helical overlap greater than 2 that no axial reciprocating motion is necessary while lapping.

Figure 7.5 Shaving cutter. (Courtesy of National Broach and Machine Division of Lear Siegler, Detroit.)

SHAVING

Shaving is a finishing operation which uses a high-speed-steel, hardened and ground, precision cutter which is in the form of a helical gear (Figure 7.5). The cutter teeth have gashes which act as cutting edges. Shaving will improve the tooth spacing, profile, lead, runout, and surface finish which was generated in the hobbing or shaping process. Shaving is often used to refine the tooth surface prior to hardening, thereby minimizing heat-treat distortion.

The shaving cutter is meshed with the work gear in a crossed axis relationship (Figure 7.6), and rotated while the center distance between the two is reduced in small increments. Simultaneously, the work is traversing back and forth in relation to the cutter.

Shaving can achieve gear qualities of up to AGMA Class 13. The quality of a shaved gear is strongly dependent on the quality of the preceding hobbing or shaping operation. Shaving can remove approximately 70% of the errors in the as-cut gear. A surface finish of 25 μin. rms is normally achieved, with much finer finishes possible. The optimum gear tooth hardness for the shaving process is Rc 30.

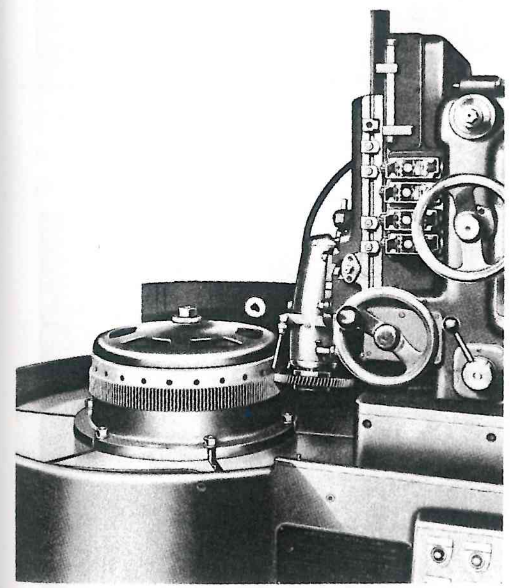


Figure 7.6 Shaving cutter machine. (Courtesy of National Broach and Machine Division of Lear Siegler, Detroit.)

Shaving may be applied to spur and helical gears, both external and internal. Tooth profile modifications and crowning can be achieved. Profile modifications are accomplished by grinding the correct form into the cutting tool teeth. Crowning is accomplished by sinking the cutter more deeply into the tooth ends than into the middle.

GRINDING

The most accurate gears are produced by the grinding process. It is used primarily to finish hardened gears in the range Rc 55 and up. Grinding can produce AGMA Qualities of Class 14 and higher and surface finishes as fine as 10 μin. rms. Gear tooth profile modification and crowning of the gear tooth face can be accurately produced by the grinding process. The two basic techniques used to grind gear teeth are the form grinding method and the generating method. Either method, when closely controlled, can produce the highest-quality gears.

Figure 7.7 illustrates a form grinding machine which utilizes a disk-type grinding wheel. The wheel form is contoured by a diamond dressing tool into the

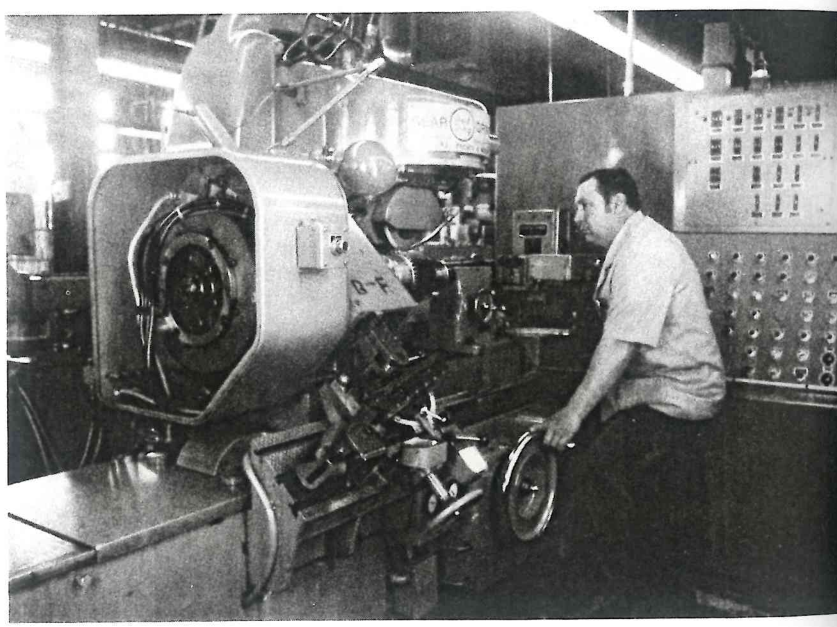


Figure 7.7 Form grinding machine. (Courtesy of National Broach and Machine Division of Lear Siegler, Detroit.)

form of the tooth space being machined out. Both the root and the tooth flank area can be ground, or the flank area alone, by cropping the tip of the grinding wheel. The wheel grinds one tooth at a time and an index plate having the same number of spaces as the number of teeth in the gear is used to index around the part. Tooth spacing is a function of the accuracy of the index plate and spacing error can be minimized by using an index plate larger in diameter than the part being ground.

Generating grinders use disk wheels of various types, such as conical or saucer shaped or threaded worm types of wheels. The work is rolled with respect to the grinding wheel and the wheel reciprocates axially with respect to the work. Generating grinder wheels act as though they were straight-sided racks in mesh with the gear being ground. The disk is therefore dressed in a straight-sided form with the proper pressure angle. The straight-sided form is modified to achieve whatever tooth modifications are required on the gear to be ground.

When grinding case-hardened gears it is important to control all phases of the process in order to minimize the amount of stock removal during grinding. Too much stock removal will result in loss of the hardened surface area and the benefits of case hardening. This is particularly critical in the root fillet area. In order to have sufficient case depth in this critical bending stress region, the tooth must be cut prior to hardening with a generous root fillet radius. Figure 7.8A shows the shallow case depth that results in the root fillet area when the radius is too small. The surface area available for carbon to penetrate in this region is insufficient. After hardening, when the tooth is ground, little or no case will be left and bending failure resistance will be significantly reduced. In order to attain the largest possible root fillet radius the hob or shaper cutter tips must be designed to produce an undercut below the form diameter of the gear tooth (Figure 7.8B). This configuration is referred to as protuberance cut

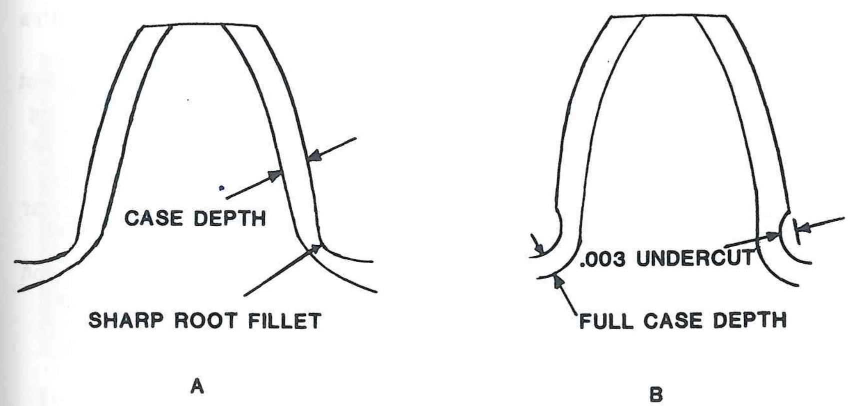


Figure 7.8 Hardened gear teeth in the as-cut condition.

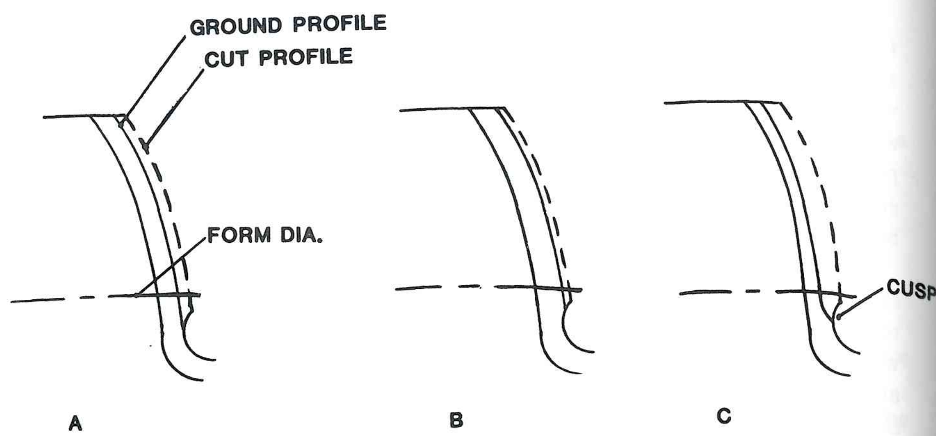
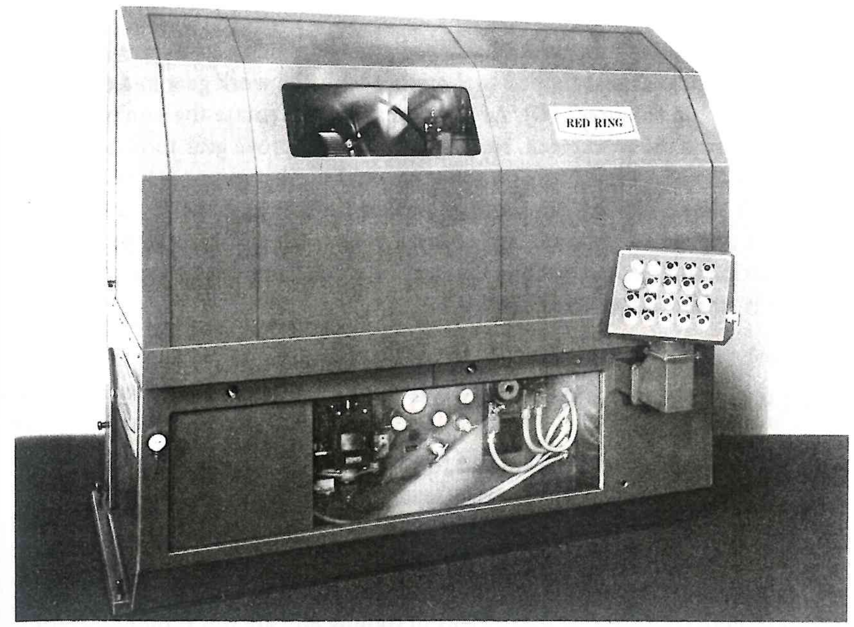


Figure 7.9 Hardened gear teeth after grinding.

since in order to generate the undercut the cutter has a protuberance added at its tip. The case depth in the root fillet is greater with protuberance cut teeth because there is a greater surface area available for carbon to penetrate during the carburizing cycle. Another advantage of protuberance cutting is that only the tooth flank need be ground, leaving the carburized case in the root fillet intact. Figure 7.9 shows the types of tooth profiles that result from this grinding technique. Assume that the as-cut undercut is 0.003 in., as shown in Figure 7.8B. If the stock removal during grinding of the tooth flank is 0.003 in., the profile will be tangent to the root fillet radius (Figure 7.9A). This is the ideal case. Stock removal of less than 0.003 in. will leave some undercut in the finished tooth (Figure 7.9B) and stock removal greater than 0.003 in. will result in a cusp (Figure 7.9C).

There is some question whether a tooth with unground root or one with a fully ground root is stronger. The fully ground root may have a better form, however, some of the hardened case has been removed. With a fully ground root there is always the danger that stock removal has been excessive. Both grinding methods are commonly used and, if correctly done, produce gears that achieve successful operation.

The grinding operation is strongly influenced by the condition of the gear after heat treatment. In carburized parts excessive case carbon can result in temper or grinding cracks during grinding. Oxidation resulting in decarbuziation at the surface will cause overheating during grinding and surface temper. Heat-treat distortion must be minimized so that excessive stock removal is not required to clean up the part. Often, development programs are required to determine what type of distortion will occur during heat treating. The parts can then be cut to take advantage of the movement during heat treating. Use of quenching dies is another method to limit heat-treat distortion.



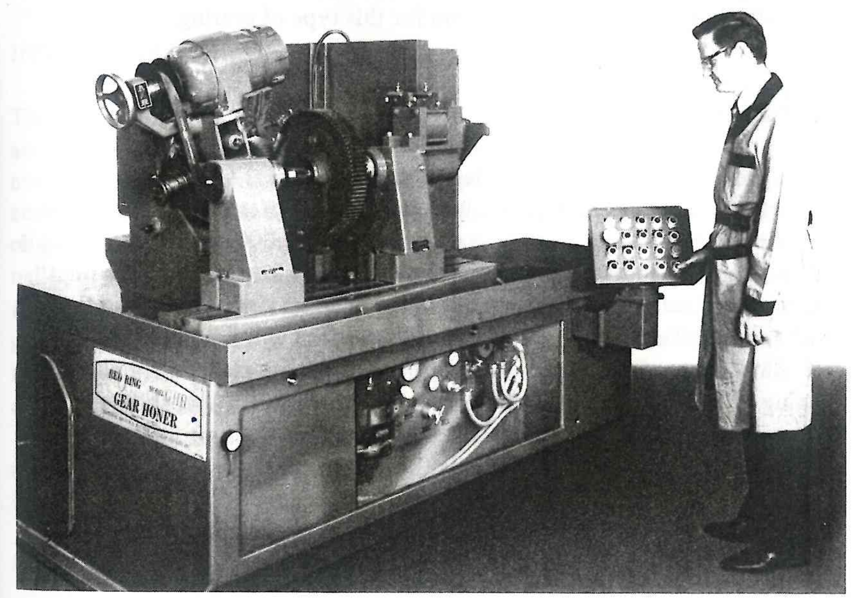


Figure 7.10 Honing machine. (Courtesy of National Broach and Machine Division of Lear Siegler, Detroit.)

HONING

Honing is a finishing process for hardened gears in which an abrasive impregnated plastic helical gear-shaped tool meshes with the work gear in a crossed axis relationship (Figure 7.10). As the tool and work rotate the honing tool traverses across the workpiece. Honing is used to improve gear tooth surface finish, remove nicks and burrs on the teeth, and correct minor errors in the gear tooth shape. Honing can produce surface finishes as fine as 6 μin. rms. It can be applied to external and internal gear teeth and achieve profile modification and crowning. The amount of stock removed in the honing process is small: 0.0003 to 0.001 in.

OTHER METHODS OF GEAR TOOTH MACHINING

The techniques used to generate and form gear teeth discussed above are not the only machining processes in use. Gears are also produced by milling, shear cutting, broaching, stamping, molding, die casting, sintering, and rolling. These processes either do not produce gears of sufficient strength or accuracy for the applications covered in this book or are high-production techniques which require considerable investment in tooling and are not economical for the relatively small production lots run for this type of gearing.

SHOT PEENING

In this process high-velocity particles are shot at the gear tooth, deforming the metal surface, in order to produce residual compressive stresses at the surface. The depressions are very shallow: less than 0.0001 in. Shot peening is used primarily to induce or increase the compressive stress in the root fillet area to a level sufficiently high such that the bending stress applied by the tooth load is offset and fatigue cracks at the surface will not initiate or propagate. Similar beneficial results in pitting resistance are claimed for shot peening of the tooth flanks; however, some users maintain that shot peening increases tooth surface roughness and they shot peen the root fillet area only. This may be accomplished by masking the gear tooth flank during peening or by shaving or grinding the tooth flanks after peening.

Effectiveness of shot peening depends on the following factors:

Type of shot
Hardness of shot
Uniformity of shot

Velocity of the shot stream
Duration of the treatment
Distance from the nozzle to the workpiece
Angle of impact

In order to develop the right combination of these factors for a given application a technique named the Almen strip test is used. The test strip is a standard piece of spring steel which is shot peened on one side. The resulting residual surface compressive stresses make the strip bow upward and the height of the bowed arc is an index of the intensity of the peening. The arc height can be related to the depth of the residual stress layer and the magnitude of residual stress.

Steel shot with a hardness of Rc 45 to 55 is commonly used for through-hardened gears and a shot hardness of Rc 55 to 65 is used for case-hardened parts. In some cases glass shot is used. In order to penetrate the root fillet area the shot diameter should be no larger than half of the smallest fillet radius. The shot quality in terms of spherical shape and uniform size is important. Sharp-edged particles can damage the tooth surface. The shot stream should be approximately perpendicular to the surface being blasted. Because the gear tooth root is contoured, it is usually necessary to blast from more than one position.

INSPECTION EQUIPMENT

The gear tooth involute profile, lead, spacing, runout, and thickness can be accurately measured on the bench using standard inspection equipment; however, specialized machines are available to perform these inspections quickly, economically, and with the capability of recording the results on a permanent chart. When all the foregoing elements of a gear tooth are measured individually, the process is called an analytical inspection. Another technique used is the functional inspection, where the gear in question is rolled with a master gear or its mating gear to determine acceptability. When the functional inspection indicates a problem, an analytical inspection is performed to determine the cause.

The following paragraphs will describe some of the specialized instruments available for performing analytical and functional inspections of gear teeth.

Involute Profile Measuring Machine

Figure 7.11 illustrates one type of involute checker. The gear is mounted in centers and rotates in a timed relationship with a stylus that traverses the

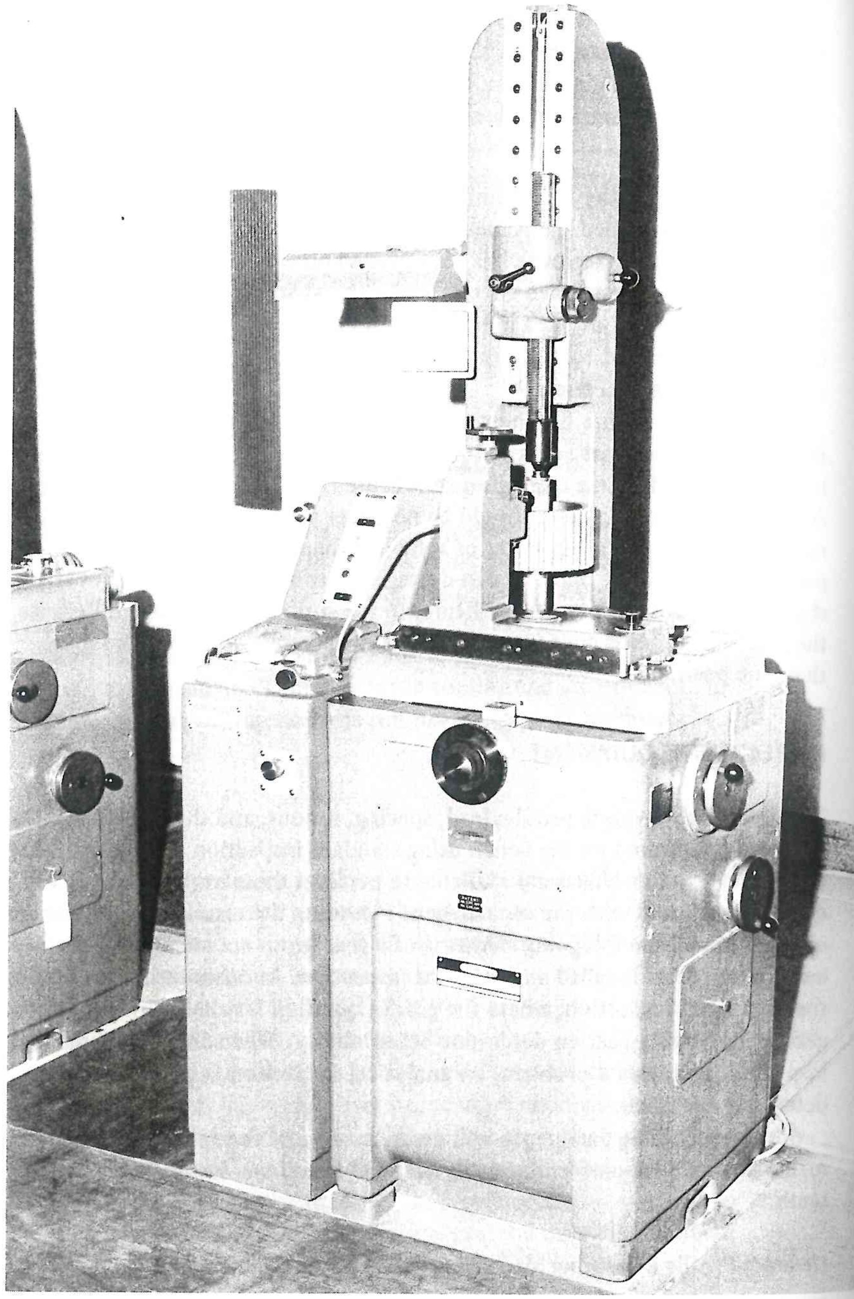


Figure 7.11 Involute profile measuring machine. (Courtesy of Fellows Corporation, Springfield, Vt.)

profile in the transverse plane. The deviation of the profile from a true involute form is measured and displayed on a dial indicator graduated in 0.0001 in. increments and/or a recording pickup which may have a sensitivity as fine as 0.000020 in. The profile is recorded on a strip chart. To calibrate the machine an accurate master with known form is inspected.

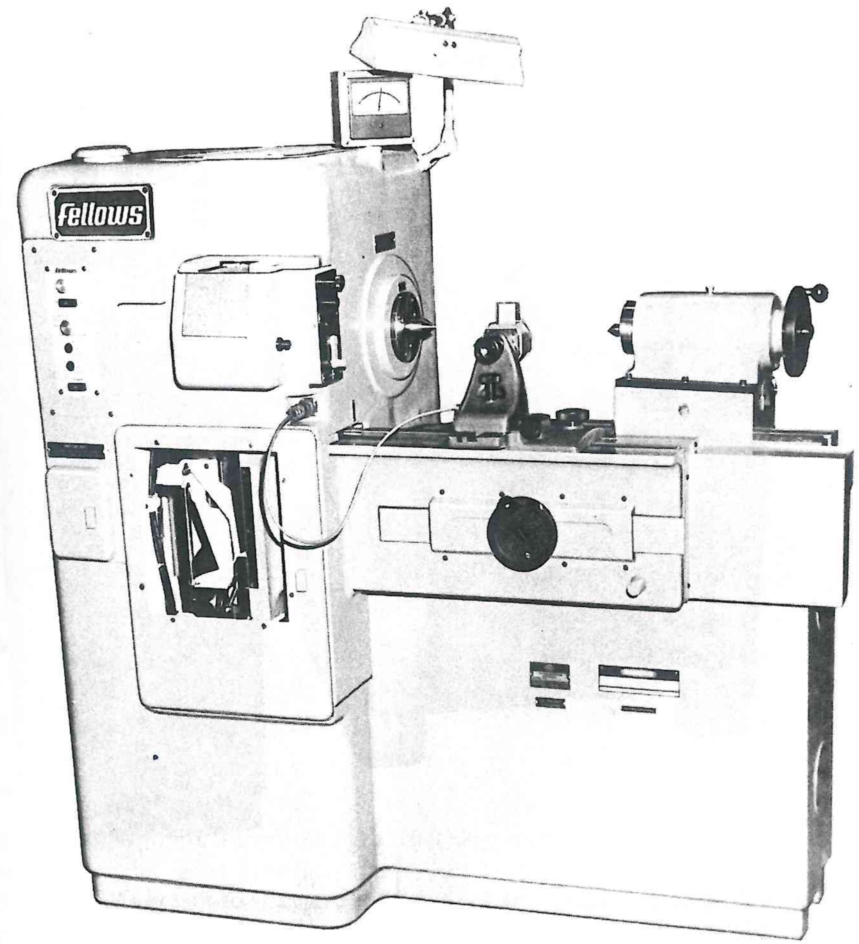


Figure 7.12 Lead checking machine. (Courtesy of Fellows Corporation, Springfield, Vt.)

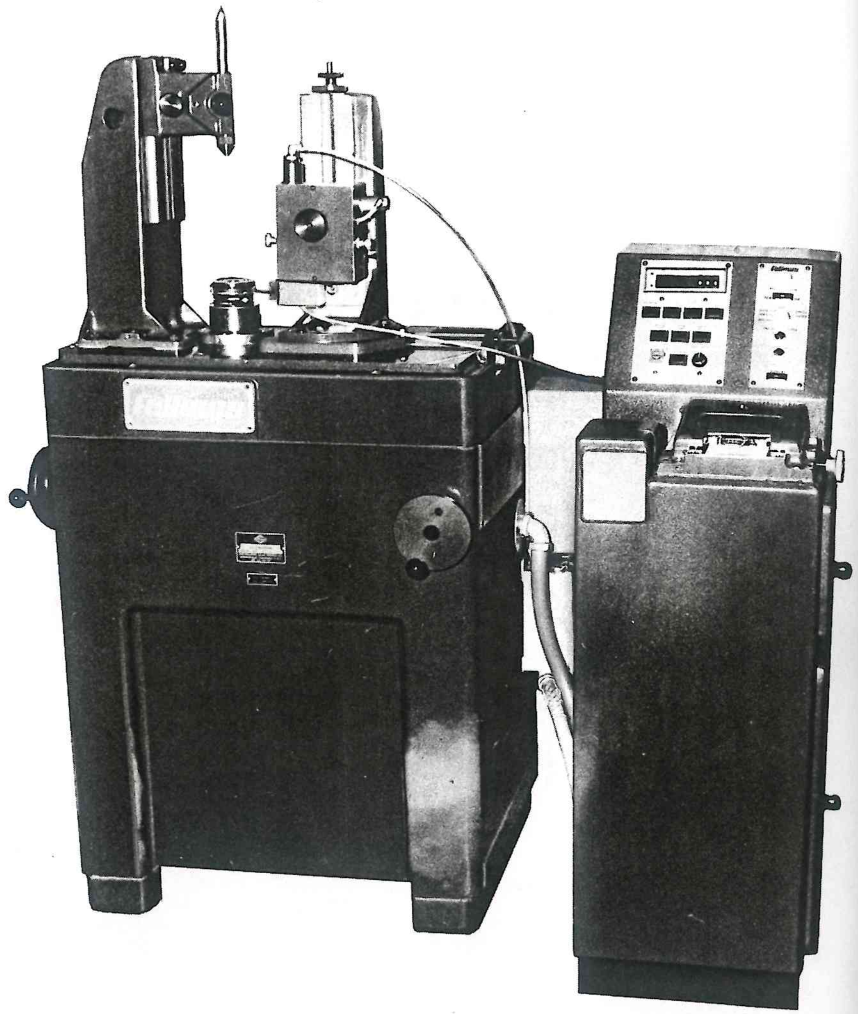


Figure 7.13 True-position spacing checker. (Courtesy of Fellows Corporation, Springfield, Vt.)

Lead Measuring Machine

A gear tooth lead inspection is an indication of the alignment of the face with the axis of rotation. In the case of a spur gear the face width should be parallel to the reference axis. For helical gear teeth the lead check determines if the tooth has the required helix angle. Figure 7.12 illustrates a lead checking machine. The gear is mounted between centers and a stylus traverses the face in an axial direction. For helical gears the gear rotates as the stylus is advanced at the proper rate to satisfy the lead relationship. If the lead is correct the stylus will describe a straight line. The deviation from a straight line is displayed on a dial indicator and/or a strip chart using an electrical pickup. The lead measurement can be made to an accuracy of 0.0001 in. or better.

Tooth Spacing

Two types of tooth spacing checkers are in general use. One measures pitch variation, the difference between one circular pitch, and the circular pitches immediately before and after. This type of machine has a fixed finger that acts as a stop on the pitch line of a tooth. A second finger or stylus senses the position of the adjacent tooth and actuates a dial indicator or recording pen.

The other type of tooth spacing checker measures index variation or the true position of a tooth. The gear is mounted on a rotating disk which turns in increments corresponding to the number of teeth in the gear. A stylus comes in and contacts each tooth near the pitch line. The first tooth is set at 0.0 and the deviation of each subsequent tooth from its theoretical position is measured and recorded. Figure 7.13 illustrates a true-position spacing checker. The two types of spacing checkers will not give identical readings on the same gear because they operate on different principles; however, analysis of the findings of either machine will lead to the same conclusions concerning gear acceptability. On either machine tooth spacing can be checked to accuracies of 0.0001 in. or better.

FUNCTIONAL INSPECTIONS

Composite Inspection

Figure 7.14 illustrates a composite gear checking machine. This type of inspection is sometimes called a red line, in reference to the strip chart that is produced. The gear to be checked is mated with a master gear which is mounted on springs such that the center distance between the two gears can vary as the teeth mesh. The variation in center distance is a function of tooth error and is measured and recorded. The variation in center distance from tooth to tooth is a composite measurement of errors in tooth spacing, profile, lead, and surface

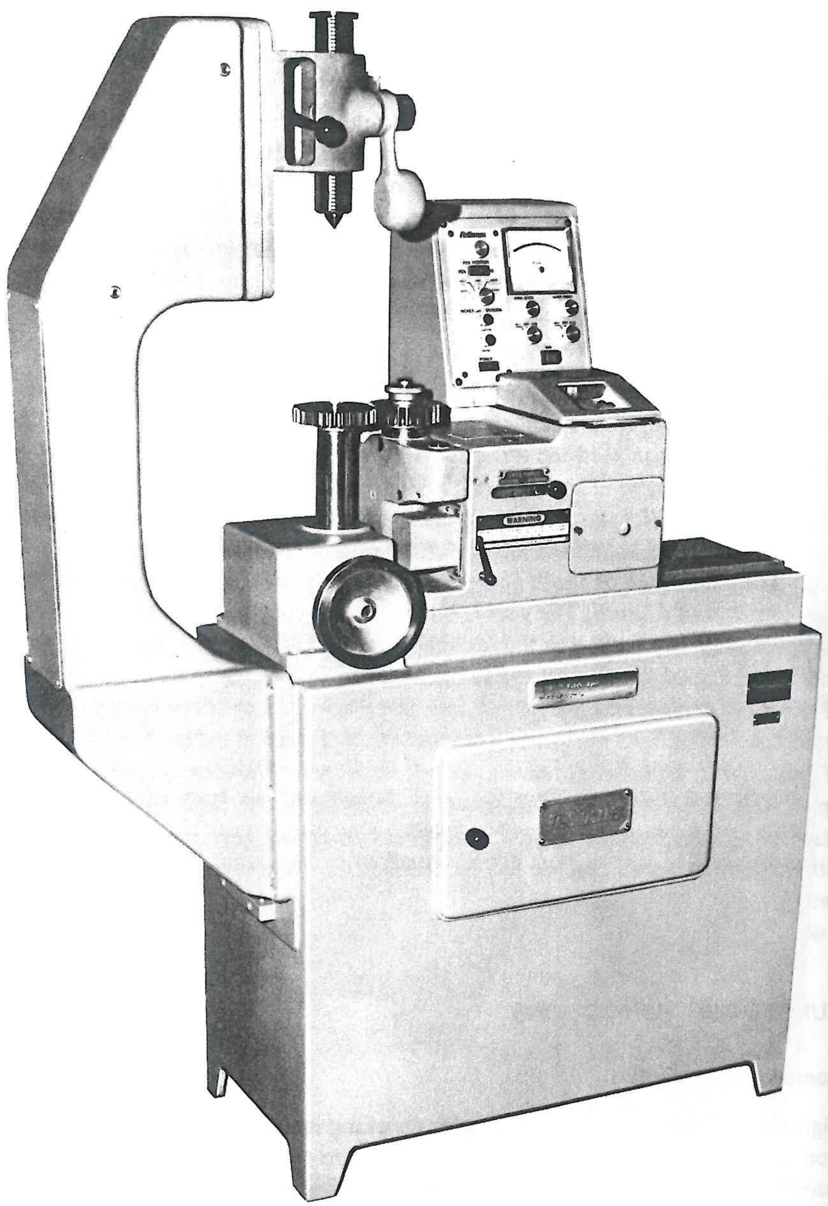


Figure 7.14 Composite gear checking machine. (Courtesy of Fellows Corporation, Springfield, Vt.)

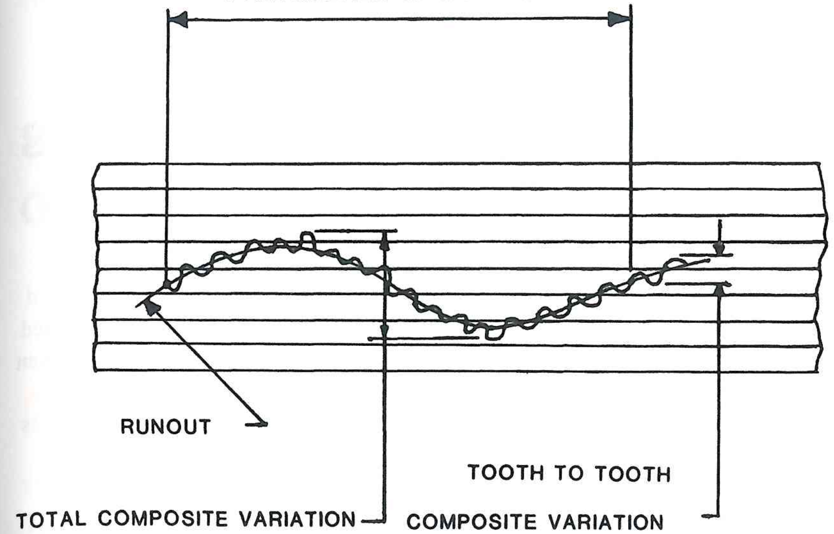


Figure 7.15 Typical red line chart.

finish. The total composite variation is a measure of the gear runout plus the local composite variations. Figure 7.15 illustrates a red line chart.

The composite error reading is influenced by errors in the master gear as well as errors in the part being inspected; therefore, the master must be as perfect as possible. The composite inspection method is a quick, inexpensive way of checking production gears. Analysis of the red line chart can differentiate between the various types of errors; however, the composite check is not sufficiently accurate for measurement of critical gears above AGMA Quality Class 12 or gears that require accurate profile or lead modifications.

Error in Action

Gear tooth inaccuracies will result in variations of velocity ratio in a pair of mating gears as the driven gear accelerates and decelerates. Machines are becoming available which measure this variation. One type uses friction disks with diameters machined to the exact pitch diameters of the gears to be inspected. One disk is attached to the driving gear and the other disk is mounted on the same axis but independent of the driven gear. The driven disk operates at a constant speed and the driven gear angular velocity is compared to it electrically and the variation recorded. This type of checker measures the action of the loaded faces of the gear teeth only, unlike the composite checker, where the mating gears are always in tight mesh with both faces touching. For this reason it is sometimes referred to as a single flank tooth tester.

Tooth Pattern Inspection

Often gear teeth are rolled together after a marking compound is applied to the teeth in order to check the contact pattern. This can be done in an inspection fixture or in the actual gear housing. In an inspection fixture the tooth pattern will show the amount of contact across the face width and indicate how well the teeth line up. In the gear housing errors other than tooth alignment come into play, such as bearing bore parallelism. Sometimes the pattern check is used to determine a further modification of the gear tooth, which is then remachined to achieve good contact. When performing this type of check care must be taken to spread the marking compound very lightly. If the thickness of marking compound is greater than approximately 0.0001 in., the errors that the inspection is attempting to reveal will be masked by excessive smearing of the compound.

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8
GEARBOX ECONOMICS

There are a multitude of variables affecting gearbox costs which make it difficult to determine how expensive a unit must be for a given application. The trade-off between design conservatism and cost is always present, the goal being to achieve satisfactory performance at minimum price. In each application different criteria are applied as to what constitutes satisfactory performance. For a relatively inexpensive gearbox some period of trouble-free performance after which the unit is easily replaced may be perfectly satisfactory. In some process industry applications the cost of downtime incurred if a gearbox must be modified or replaced is so great that the initial gearbox price in comparison is insignificant. In such a case satisfactory performance is equated with extremely high reliability over long periods of time and in order to achieve this reliability, unit costs increase significantly.

The cost of a gearbox can be divided into three elements:

Material costs
Manufacturing costs
Purchased items such as bearings, seals, lubrication components, and so on.

These three factors are dependent on the gearbox design and configuration chosen. To illustrate this, let us look at an example of a 500-hp gear unit driven by a 5000-rpm steam turbine and driving an 1800-rpm generator. This application might be satisfied by any of the following designs:

1. Through-hardened cut double helical gearing
2. Through-hardened cut and shaved double helical gearing
3. Single helical, case hardened and ground gearing

4. Single helical, case hardened and ground pinion meshing with a through-hardened cut gear

Design 1 will be the largest unit since the tooth stresses must be low for this relatively inaccurate method of fabrication. Design 3 will be the smallest and therefore will have significantly lower material costs. The single helical gearing, however, requires more machining time since it is cut and ground and also requires more sophisticated heat treatment. Bearings for design 3 will probably be more expensive since there are gear thrust loads to deal with. The larger gear sets of designs 1 and 2 will generate more heat in churning, requiring more cooling oil flow and therefore a larger pump, cooler, filter, reservoir. These are only a few examples of many tradeoffs that are made in selecting a gearbox design.

If one were to solicit competitive quotes from companies making each of these types of designs it would not be possible to predict which would be least expensive. If one company's design were inherently significantly more costly than the others, the company would not long remain in business. Quite often the low bid is not a reflection of economical design or manufacture but an indication of how badly the company wants the particular program.

A cost consideration beyond the gearbox itself is the impact of the unit on the total system. A smaller gearbox will save costs associated with the base plate. Also, shipping costs will be less. The gearbox configuration, parallel shaft with an offset between input and output, or planetary with concentric shafts, will also affect the system design and influence costs. Planetary gears lend themselves to close coupling of the gearbox to the driver or driven equipment, with the potential of saving coupling and shafting costs.

From the discussion above it can be seen that if one wanted to determine the most cost effective gearbox for a new application and were free to choose any manufacturing and heat treating methods, a detailed design study and cost analysis of the numerous potential design solutions would be required to arrive at a sensible conclusion. More often companies buy gearboxes complete from a manufacturer with existing facilities or have their own machine tools and facilities; therefore, the design chosen reflects the type of gearbox the company has made in the past. The cost question then focuses on what materials to use and what degree of quality to require in the gearbox components.

GEAR QUALITY

The types of gears being discussed in this book range in American Gear Manufacturers Association (AGMA) Quality Class from about 8 for the less accurate end to 14 for the most accurate. A Quality Class 8 gear might be used in low-speed mining equipment, while a Quality Class 14 gear might be used in a sophisticated

very high speed compressor drive. Automotive gears are typically Quality Class 10 to 12. The quality achieved is to some extent determined by the finishing process; however, the various processes overlap in terms of quality produced. An approximate comparison follows:

AGMA quality range	Process
8–12	Hobbing and shaping
9–13	Shaving
10–14 and up	Grinding

Figure 8.1 presents an approximate relationship between AGMA quality class and cost. The numbers are relative, with a Quality 14 gear costing approximately 8 times as much as a Quality Class 8 gear. The figure shows that at the high-quality end of the spectrum a small increase in quality requires a large expenditure. Conversely, at the low end decreased quality offers little savings. AGMA standards give recommendations for quality classes suitable for various applications. As experience is gained in a particular service it is possible to ascertain more accurately which tooth tolerances are important and how closely they must be held. It would then be appropriate to buy gears to a specification reflecting that experience rather than to a general quality specification. To be

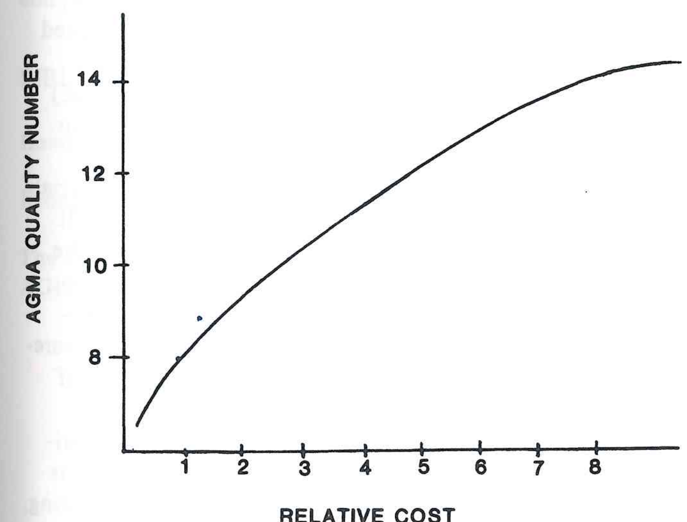

Figure 8.1 Approximate relationship between AGMA quality class and cost.

certain that the quality specified is reflected in the gear tooth geometry, a user should request inspection documentation. These might take the form of red line, involute, lead, or tooth spacing charts. Measurements over balls, bore parallelism, center distance measurements, and so on, might also be requested to be documented. There may be additional costs for this documentation. The need for and extent of inspection documentation is a matter of judgment, but if AGMA quality class requirements of 12 or above are specified, it is recommended that 100% documentation be requested. For Quality Class 14 the user and manufacturer should agree beforehand on how the inspections will be made and the results interpreted. For quality classes lower than 12 it is good practice to document prototype units so that if field problems occur in subsequent production gearboxes, a baseline of quality has been established.

So far, our discussion of quality has been confined to gear tooth geometry. If it is determined that high quality is required in the gear teeth, undoubtedly other components in the unit must also be specified to equivalent quality levels. In fact, failures associated with bearings, the dynamic system, and the lubrication system are more common than gear tooth problems; therefore, complete specification of all components in the unit in addition to the gears is necessary.

MATERIAL COSTS

Gear steels are bought on a dollars per pound basis either as bar stock or in a forged state. Up to a diameter of approximately 6 in. it is cheaper to buy bar stock than forgings. Above 6 in. bar stock becomes harder to procure and forgings have an economic advantage, particularly since forgings can be worked closer to the final shape than bar stock and therefore save machining costs.

Taking a low-alloy steel such as Society of Automotive Engineers (SAE) 1040 as a base, the cost of high-alloy steels used in gears such as SAE 4340 or 9310 may be three or four times as high. Of course, gears made of high-alloy steel will usually be smaller than those of low-alloy steel, so the cost of material is somewhat offset by the smaller quantity. If a requirement for vacuum-melt steel rather than air-melt steel is specified, the price may be doubled. Over the years the quality of gear steels has deteriorated and it is not uncommon to encounter impurities or voids in air-melt steels. These discrepancies are sometimes found only after several machining operations have been completed; therefore, the higher cost of vacuum-melt steels may be offset by a reduction of scrapped work in process.

The question often arises: Should the gear casing be a welded steel fabrication or a casting? From a cost point of view, for a single unit a fabrication is cheaper. This is because a pattern must be machined prior to pouring the casting, and developing the pattern is a costly and time-consuming procedure. As the

quantity of gearboxes required increases, pattern costs can be amortized over the production run and castings become economically attractive. The casting can be poured close to the final shape, and extensive machining and welding can be eliminated. For a typical gear unit, the break-even point may be as low as three gearboxes. Castings may be made of cast iron or steel. A steel casting costs approximately 30% more than a cast iron part. On occasion, gear housings are made of aluminum to save weight. Aluminum is twice as expensive as steel on a dollar per pound basis, but because the housing weight will be much lower, it is not clear which is more expensive. Aluminum housings, however, usually have to incorporate steel bearing liners and inserts for threads, which increase cost.

EFFECT OF QUANTITY ON COSTS

When manufacturing gearbox components, the time to set up the equipment far exceeds the time required to machine the parts. Therefore, it is far more expensive to perform a machining operation on one part than on many parts. For instance, it may take 8 hr to set up a gear grinder and 2 hr to grind a part. Assuming a cost of \$X/hr, one part would cost \$10(X), whereas four parts would cost \$4(X) each [(8/4 + 2)(X)].

If a single special unit is required for an application, design and tooling costs must be considered. These costs will probably exceed the manufacturing expenses. It is therefore worthwhile to attempt to use a standard gear unit or one that has been manufactured for a different application, even if it requires compromising the system design to some extent.

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9
PLANETARY GEAR TRAINS

A planetary gear train is one in which the power is transmitted through two or more load paths rather than the single load path of a simple gear mesh. Figure 9.1 illustrates the components of the simplest type planetary, a sun gear, planet gears, a ring gear, and the planet carrier. When the carrier rotates about the center of the system a point on a planet gear not only rotates about the axis of the planet gear but also about the center of the system and this type of drive is called epicyclic.

In the United States, planetary gears are widely utilized in automotive automatic transmissions and in aerospace drives such as turbine engine reduction gears or helicopter transmissions but are not used extensively in industrial applications. There is a trend toward increasing utilization of planetary gears because they offer the following advantages:

1. Because they share load between several meshes, planetaries are more compact than parallel shaft drives and offer significant envelope and weight savings. An example illustrates this point:

> Let us take a 1000-hp electric motor at 1800 rpm driving a compressor at 7200 rpm. The required American Gear Manufacturers Association (AGMA) service factor is 1.6.
>
> A hardened and ground, helical, parallel shaft gear set would have the following dimensions:
>
> Pinion pitch diameter: 4.0 in.
> Gear pitch diameter: 16.0 in.
> Face width: 4.0
> Gearbox envelope: 24 X 12 X 12 in.
> Gearbox weight: 650 lb

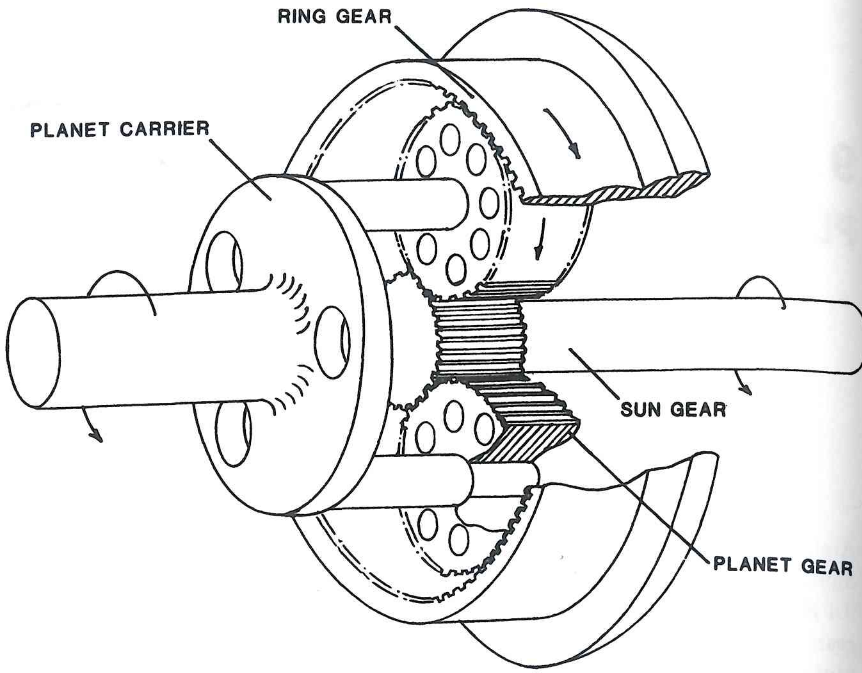

Figure 9.1 Simple planetary gearset components.

The equivalent planetary gear set with a stationary ring gear and rotating carrier would have the following dimensions:

Sun pitch diameter: 3.5 in.
Ring pitch diameter: 10.5 in.
Planet pitch diameter: 3.5 in.
Face width: 3.0 in.
Gearbox envelope: 15 in. diameter × 10 in. length
Gearbox weight: 250 lb.

Figure 9.2 illustrates the gearbox size comparison.

2. In addition to achieving minimum weight and envelope, the relatively smaller and stiffer components which result from the use of planetary gearing lead to reduced noise and vibration and increased efficiency. Noise and efficiency are strongly dependent on the speed of the components. In the example above, the parallel shaft pitch line velocity is 7540 fpm compared to the planetary gear pitch line velocity of 4948 fpm, a significant difference. The lower planetary pitch line velocity is a result of the smaller sun gear and the epicyclic

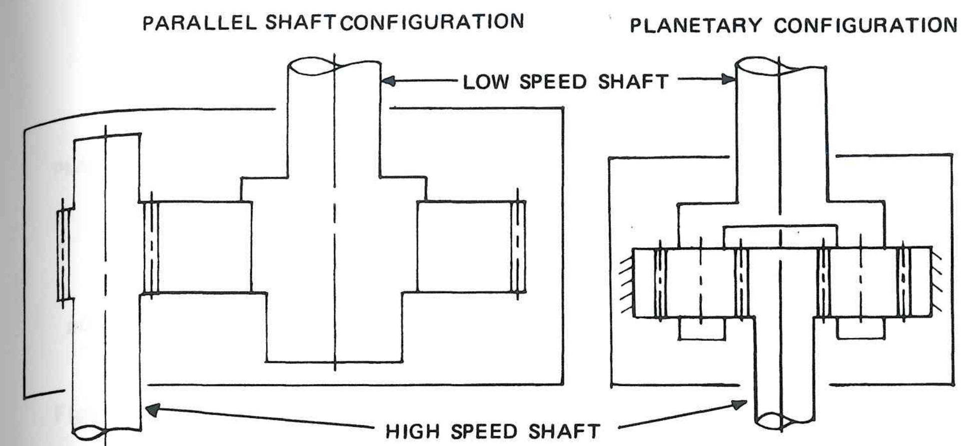

Figure 9.2 Gearbox size comparison.

action of the gear mesh. Later in this chapter the method of calculating epicyclic speeds is derived.

3. The input and output shaft axes in planetary gearing are concentric. This can lead to space savings in some installations by allowing the driving and driven equipment to be in-line. The coaxial feature is very important in automatic transmissions since it makes rapid speed changes possible without the necessity of taking gears out of mesh. The rotating components are controlled by the use of clutches and brakes to achieve speed changes.

4. The resultant radial forces on the input and output shafts of planetary gearboxes are zero since the arrangement cancels out all radial forces. In effect, the planetary gearbox transmits only torque. This simplifies bearing design and in some applications allows close coupling of the gearbox to the driving or driven equipment.

ANALYSIS OF PLANETARY GEAR ARRANGEMENTS

Planetary gears can be arranged in a multitude of configurations to achieve specific ratios and power splits. This section presents a method of calculating planetary speed ratios and power flows. The method is applied to several examples and by following the logic the reader can generate equations describing the mechanics of any planetary configuration. The following nomenclature will be used throughout.

Symbols

R = gear pitch radius, in.
W = gear angular velocity, rad/sec

W_T = tangential load, lb
n = gear rpm
Subscripts
s = sun gear
p = planet gear
r = ring gear
c = carrier
1 = primary stage (high-speed stage)
2 = secondary stage (low-speed stage)

For example, the symbol R_{s_1} would be the pitch radius of the primary-stage sun gear.

Simple Single-Stage Planetary

Figure 9.1 shows the general case where the sun gear, ring gear, and planet carrier are all free to rotate. Directions of rotation are assumed to be as shown in the figure. With all members rotating this is termed a differential system and there are potentially six different ways to connect prime movers and driven equipment:

Inputs	Outputs
Sun	Carrier, ring
Carrier	Sun, ring
Ring	Carrier, sun
Sun, ring	Carrier
Carrier, sun	Ring
Ring, carrier	Sun

The rotating elements in each of the arrangements above have a distinct speed and torque relationship. In order to define the angular velocity of all three elements in the planetary gear train, the angular velocity of two elements must be specified. Let us derive the generalized speed relationships of the gear train in Figure 9.1.

1. A point on the pitch diameter of the sun gear has a tangential velocity of W_sR_s.
2. The point on the sun gear pitch diameter is meshing with a point on the planet gear pitch diameter which has a tangential velocity made up of two components, $W_pR_p + W_cR_s$.

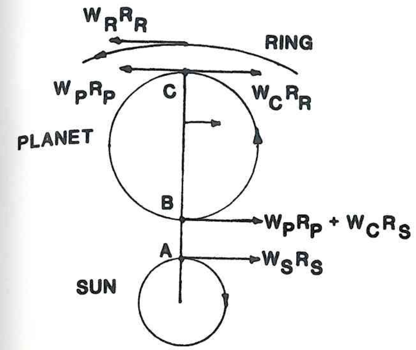

Figure 9.3 Idealized planetary system.

Figure 9.3 shows the planetary system idealized as friction disks with a point A on the sun disk meshing with a point B on the planet disk and also shows their respective tangential velocities. Because the points are meshing they must have the same tangential velocities; therefore,

$$W_sR_s = W_pR_p + W_cR_s \tag{9.1}$$

3. At the ring gear mesh a similar equation may be written:

$$W_rR_r = W_pR_p - W_cR_r \tag{9.2}$$

Note that at point C on the planet gear (Figure 9.3) the components of the tangential velocity are in opposite directions and subtractive, whereas at point B they are in the same direction and additive. The tangential velocity of the planet carrier is zero at the center of the system and equals W_cR at any radius R. Combining Eqs. (9.1) and (9.2), the general speed ratio equation is

$$W_sR_s = W_rR_r + W_c(R_s + R_r) \tag{9.3}$$

Let us apply this equation to the example shown earlier in this chapter:

$$R_s = \frac{3.5}{2}$$

$$R_p = \frac{3.5}{2}$$

$$R_r = \frac{10.5}{2}$$

In this case the ring gear was stationary; therefore,

$$W_r = 0$$

$$\text{Ratio} = \frac{W_s}{W_c} = \frac{R_s + R_r}{R_s} = 4$$

In a simple gear train the rpm of the planet gear would be rpm sun $\cdot \left(\frac{R_s}{R_p}\right)$; however, because of the epicyclic action the planet rpm is reduced and can be calculated from either Eq. (9.1) or (9.2) with

$$\text{rpm sun} = 7200$$

and

$$\text{rpm carrier} = 1800$$

the planet rpm = 5400 and the pitch line velocity of the sun planet mesh is

$$\text{PLV} = \frac{5400(3.5)\pi}{12} = 4948 \text{ fpm}$$

The rolling velocity of the sun with the planet is 5400(3.5/3.5) = 5400 rpm and its absolute velocity is 7200 rpm.

If, in the system shown in Figure 9.1, the carrier is held stationary, $W_c = 0$ and Eq. (9.3) becomes

$$\frac{W_s}{W_r} = \frac{R_r}{R_s}$$

In this case, where there is no epicyclic action, the speed ratio is simply the ratio of the sun and ring gear radii. This configuration is sometimes called a star system.

If the sun gear is held stationary, $W_s = 0$ and Eq. (9.3) becomes

$$\frac{W_r}{W_c} = -\frac{R_r + R_s}{R_r}$$

The negative sign indicates that the sense of rotation of one of the components shown in Figure 9.1 was chosen opposite to the direction in which it actually rotates. In this case, if the ring gear is driving, the carrier would rotate in the same direction as the ring gear rather than the direction shown in Figure 9.1 (viewed from the sun gear end).

In the equations above rpm can be used in place of angular velocity and number of teeth in place of pitch radius. Quite often, planetary gear equations are presented in this manner.

If all components of a planetary gear set are free to rotate, the speed ratios are dependent on the power split between the components. Figure 9.4 shows the tangential tooth loads at the sun and ring gear mesh points. The torque and horsepower of each component are as follows:

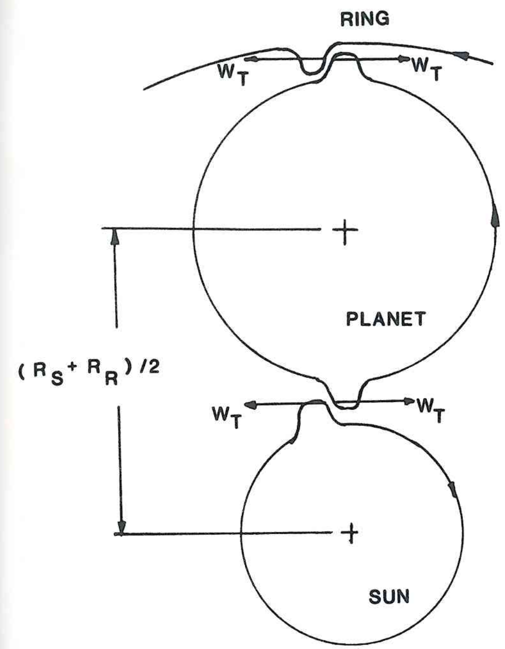

Figure 9.4 Planetary tangential tooth loads.

Component	Torque (in.-lb)	Horsepower
Sun	$W_T R_s$	$\frac{W_T R_s n_s}{63{,}025}$
Carrier	$2W_T \frac{R_s + R_r}{2}$	$\frac{W_T(R_s + R_r)n_c}{63{,}025}$
Ring	$W_T R_r$	$\frac{W_T R_r n_r}{63{,}025}$

It should be noted that the tangential load W_T used in the table is the summation of the tangential loads at each planet gear mesh.

Let us assume that in the previous example the 1800 rpm carrier with a 1000 hp input drives a compressor which is attached to the sun gear and also drives an oil pump which is attached to the ring gear. The oil pump absorbs

50 hp. The three horsepower equations in the previous table can be solved for the three unknowns W_T, N_s, and R_r. Using the carrier horsepower equation with $R_s = 1.75$, $R_r = 5.25$, and $n_c = 1800$ the tangential load $W_T = 5002$ lb. Knowing W_T, the sun and ring horsepower equations yield

$$n_s = 6840 \text{ rpm}$$
$$n_r = 120 \text{ rpm}$$

It should be noted that in this discussion no attempt was made to account for the power loss due to friction in the train.

The arrangement shown in Figure 9.1 is capable of producing gear ratios in a range of approximately 2:1 to 10:1. The speed ratio derived earlier in this section is R_r/R_s for the stationary carrier, rotating ring gear case. In this situation the input and output directions of rotation are reversed. With a stationary ring gear and rotating carrier, the speed ratio is $1 + (R_r/R_s)$ and the input and output rotation directions are the same. At low ratios the sun gear diameter approaches the ring gear diameter and therefore planet gear diameters become small and impractical. As one strives for the higher ratios in a single-stage planetary, the sun gear becomes smaller and is stress limited. Also, at ratios above approximately 10 the ring gear becomes large and more economical arrangements are available.

Compound Planetary

For a ratio of approximately 10:1 to 16:1, the compound planetary gear set (Figure 9.5) provides a practical and economic configuration. The speed reduction equations for this design follow:

$$W_s R_s = W_{p1} R_{p1} + W_c R_s$$
$$W_r R_r = W_{p2} R_{p2} - W_c R_r$$
$$W_{p1} = W_{p2}$$

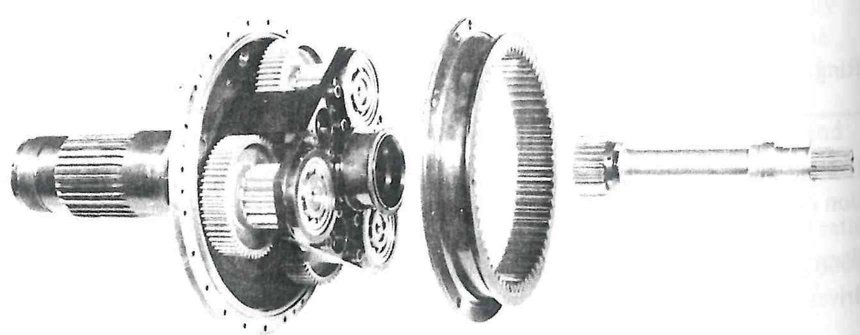

Figure 9.5 Compound planetary gear.

Combining these equations, we have

$$W_s = W_r \frac{R_r R_{p1}}{R_s R_{p2}} + W_c \left(1 + \frac{R_r R_{p1}}{R_s R_{p2}}\right)$$

In the case of a stationary ring gear and rotating carrier the input and output direction of rotation are the same and the speed ratio is

$$R = \frac{W_s}{W_c} = 1 + \frac{R_r R_{p1}}{R_s R_{p2}}$$

In the case of a stationary carrier and rotating ring gear the input and output directions of rotation are opposite and the speed ratio is

$$R = \frac{W_s}{W_r} = \frac{R_r R_{p1}}{R_s R_{p2}}$$

For low ratios in the range 3:1 a reverted type of compound planetary is used as shown in Figure 9.6. The speed ratio for this train is simply

$$R = \frac{W_{s1}}{W_{s2}} = \frac{R_{s2} R_{p1}}{R_{s1} R_{p2}}$$

In this case the input and output directions of rotation are reversed.

Multistage Planetaries

By combining stages of planetary gearing in various arrangements, large ratios can be achieved. For instance, let us take two stages of simple planetary gearing as shown in Figure 9.1 and combine them in two ways as shown in Figure 9.7A and 9.7B. Figure 9.7A presents a drive where the primary-stage carrier drives the secondary-stage sun gear. The reduction ratio of the primary stage is

$$R_1 = \frac{W_{s1}}{W_{c1}} = \frac{R_{s1} + R_{r1}}{R_{s1}}$$

The reduction ratio of the second stage with a carrier output is

$$R_2 = \frac{W_{s2}}{W_{c2}} = \frac{R_{s2} + R_{r2}}{R_{s2}}$$

and the total reduction is

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