GEAR DRIVES FOR TURBOMACHINERY

by Peter Lynwander

INTRODUCTION

Gearboxes for turbomachinery applications are designed to achieve years of trouble-free operation. As operating speeds increase, and reliability requirements become critical due to the high cost of downtime, gear manufacturers are refining their analytical, design and manufacturing techniques to keep pace with new technology. It has become apparent that gear units, when incorporated into a system of rotating machinery, become susceptible to a variety of problems. The gear manufacturer and the user, therefore, must take a systems approach to the specification, installation, operation and maintenance of a gearbox. All characteristics of the drive system from the driver to the driven equipment including the lubrication system and accessories can influence gearbox operation.

This chapter presents material concerning all phases of gearbox application with emphasis on information that will help the user operate and maintain the equipment. A discussion of selection and design procedures is included to familiarize the reader with gearbox principles.

Throughout the chapter standards and practices developed by the American Gear Manufacturers Association are referred to. Successful selection, rating, installation and maintenance of gearboxes can be accomplished by the use of AGMA standards and practices.

TYPES OF GEAR DRIVES

The choice of a gear drive depends on the application, its environment and the physical constraints of the system. The gearbox geometry is defined by four parameters which are determined by the characteristics of the driving and driven machinery:

Horsepower transmitted
Ratio required (reduction or increasing)
Speeds
Arrangement of Shafting (Fig. 1)

When specifying a gear drive, the efficiency requirement, noise generation, and space and weight limitation must also be considered. The physical environment, dust, humidity, corrosive atmosphere, etc. must be addressed in the design stage.

Although the input and output shaft arrangement can be concentric, parallel offset, right

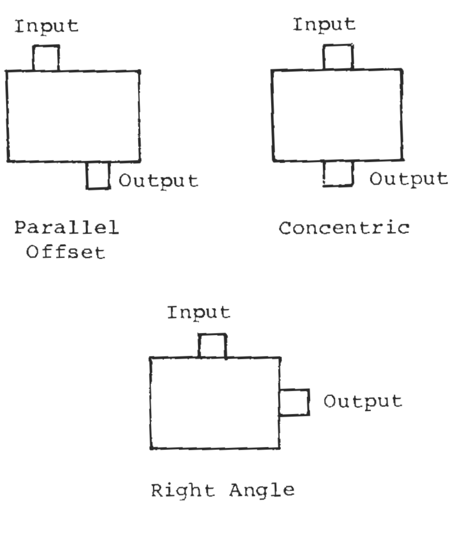


Figure 1. Types of Gearbox Configurations

angle or skewed, the great majority of turbomachinery gearboxes are used in systems that require either parallel offset or concentric shaft configurations. Figure 2 illustrates a typical parallel shaft high speed gear unit.

Parallel offset or concentric drive shaft gearboxes use either spur, single helical or double

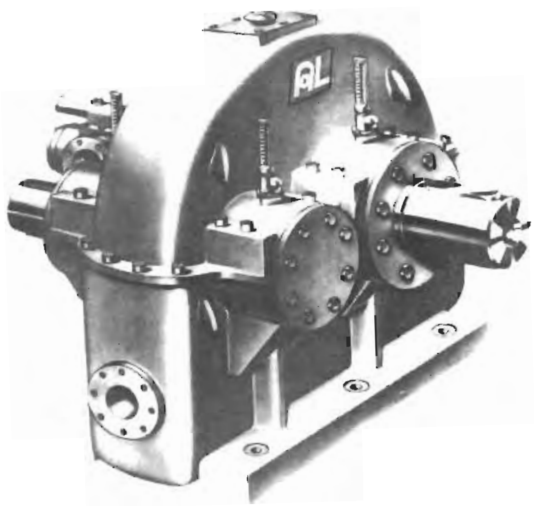


Figure 2. American Lohmann Parallel Shaft High Speed Gear Unit

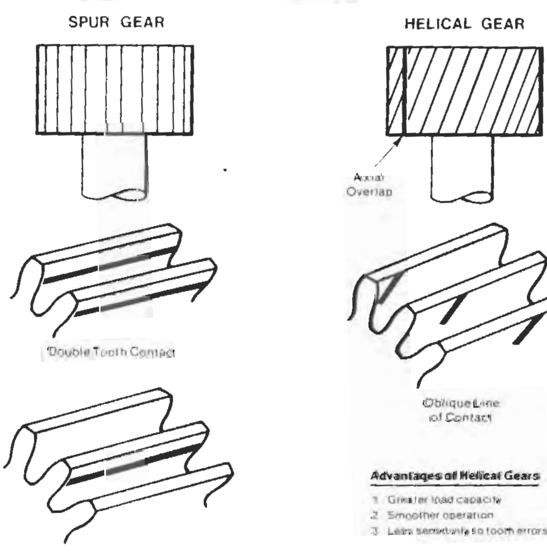


Figure 3. Comparison of Spur and Helical Gear Teeth

helical tooth forms. Figure 3 illustrates the difference between spur and helical gearing. Spur gears have the advantage of not generating axial thrust loads but are limited in capability and they generate more noise and vibration than helical gears. The reason for this is that helical gears have an overlap in both the axial and transverse planes, Fig. 3. With a conventional spur gear, the load is transmitted by either one or two teeth at any time. Thus, the elastic flexibility is continuously changing as load is transferred. With helical gearing, the load is shared between a sufficient number of teeth to allow a smooth transference of load and a constant elastic flexibility. In addition, helical gears have larger load carrying capability.

In order to take advantage of helical gearing, yet not generate axial thrust, double helical gearing can be used. Many industrial and marine applications utilize soft, approximately Rc 35, double helical gearing. Double helical gearing offers low noise and vibration along with zero net axial thrust. Also, the ratio of face width to diameter in each half of the mesh can be held to reasonable limits, therefore, end loading of the tooth face due to tooth errors or deflection is less likely to occur.

The advantage of zero thrust is offset by the fact that double helical gears must adjust themselves axially. One gear of the set is free to move axially and continuously shifts to achieve equilibrium. This can lead to detrimental axial vibrations. Also, if the gear is hung up axially due to external loads or internal friction overloading of one helix will occur.

In order to achieve minimum envelope and maximum reliability the latest technology util-

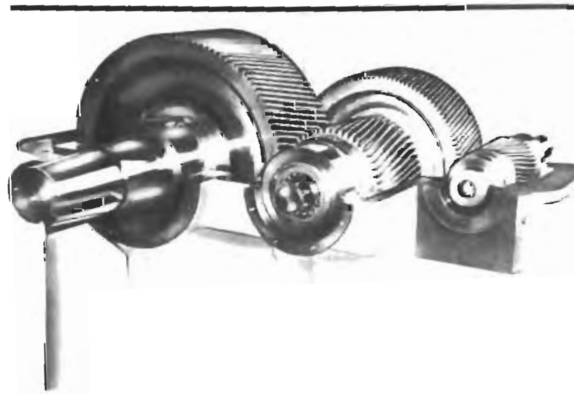


Figure 4. Two Stage, Single Helical Generator Drive Manufactured by American Lohmann. 4500 HP, 14500 RPM In, 1500 or 1800 RPM Out

Table 1. High Speed Gear Unit Service Factors

(Courtesy American Gear Manufacturers Association)

AGMA STANDARD PRACTICE FOR
HIGH SPEED HELICAL AND HERRINGBONE GEAR UNITS
Service Factor Values

APPLICATION	Service Factor — Prime Mover: Motor	Turbine	Internal Combustion Engine (Multi-Cylinder)
BLOWERS			
Centrifugal	1.4	1.6	1.7
Lobe	1.7	1.7	2.0
COMPRESSORS			
Centrifugal-process gas except air conditioning	1.3	1.5	1.6
Centrifugal-air conditioning service	1.2	1.4	1.5
Centrifugal-air or pipe line service	1.4	1.6	1.7
Rotary-axial flow-all types	1.4	1.6	1.7
Rotary-liquid piston (Nash)	1.7	1.7	2.0
Rotary-lobe-radial flow	1.7	1.7	2.0
Reciprocating-3 or more cyl.	1.7	1.7	2.0
Reciprocating-2 cyl.	2.0	2.0	2.3
DYNAMOMETER - test stand	1.1	1.1	1.3
FANS			
Centrifugal	1.4	1.6	1.7
Forced draft	1.4	1.6	1.7
Induced draft	1.7	2.0	2.2
Industrial and mine (large with frequent start cycles)	1.7	2.0	2.2
GENERATORS AND EXCITERS			
Base load or continuous	1.1	1.1	1.3
Peak duty cycle	1.3	1.3	1.7
PUMPS			
Centrifugal (all service except as listed below)	1.3	1.5	1.7
Centrifugal-boiler feed	1.7	2.0	
Centrifugal-descaling (with surge tank)	2.0	2.0	
Centrifugal-hot oil	1.5	1.7	
Centrifugal-pipe line	1.5	1.7	2.0
Centrifugal-water works	1.5	1.7	2.0
Dredge	2.0	2.4	2.5
Rotary-axial flow-all types	1.5	1.5	1.8
Rotary gear	1.5	1.5	1.8
Rotary-liquid piston	1.7	1.7	2.0
Rotary-lobe	1.7	1.7	2.0
Rotary-sliding vane	1.5	1.5	1.8
Reciprocating-3 cyl. or more	1.7	1.7	2.0
Reciprocating-2 cyl.	2.0	2.0	2.3
PAPER INDUSTRY			
Jordan or refiner	1.5	1.5	
Paper machine-line shaft	1.3	1.3	
Paper machine-sectional drive	1.5		
Pulp beater	1.5		
SUGAR INDUSTRY			
Cane knife	1.5	1.5	1.8
Centrifugal	1.5	1.7	2.0
Mill	1.7	1.7	2.0

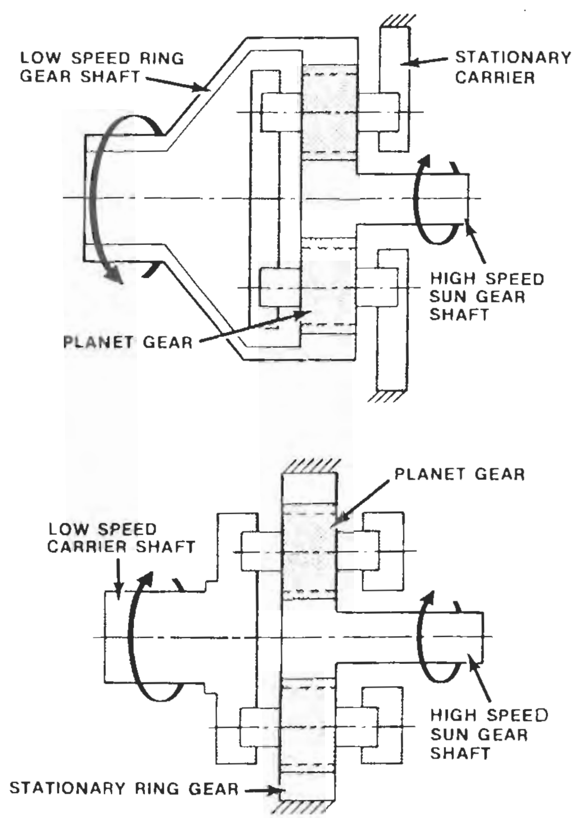


Figure 5. Basic Planetary Gear Configurations

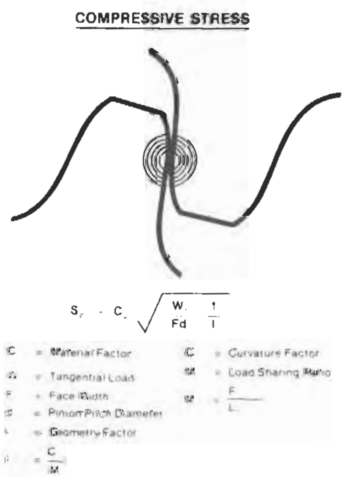


Figure 6. Calculation of Compressive Stress (Pitting Criterion)

izes single helical, hardened and precision ground gearing. With single helical gears the thrust load axially locates the gear shaft against the thrust bearing. Bearing design has progressed to the point where thrust loads are routinely handled either by hydrodynamic tapered land or tilting pad configurations or anti-friction thrust bearings. Because case hardened gears have maximum load carrying capacity, gear size can be minimized, therefore, the ratio of face width to diameter of a single helical gear can be held to reasonable limits. Also, the bearing span with single helical gears is shorter resulting in lower elastic deflection. Figure 4 illustrates a generator drive gearbox with two stages of single helical gearing.

A single helical hardened and ground gear set can reduce by up to one half the envelope and weight of a through hardened double helical gearbox with equivalent capacity. The inherent precision of the grinding process results in accurate tooth geometry leading to minimum noise and vibration. It is possible to harden and grind double helical gear teeth, however, in order to grind a one piece double helical gear a large central gap is required between the two helices to allow runout of the wheel. Gears can be ground in halves and then assembled, but this presents serious alignment and attachment problems.

Gearbox Rating

Turbomachinery gear units are usually rated by an established practice such as A.G.M.A. Standard 421.06, "Standard Practice for High Speed Helical and Herringbone Gear Units" or API Standard 613, "Special Purpose Gear Units for Refinery Services." The two standards mentioned above are basically for helical and double helical parallel shaft units. Gear horsepower rating is calculated on the basis of strength and durability. In addition, to gear rating, the standards cover other aspects of gearbox design such as bearings, shafting, etc.

Standard parallel shaft helical or double helical gear units are available and described in published catalogs.

Service Factor

Gear catalog horsepower ratings are given with a service factor of one, and before a unit is selected the operating conditions must be defined so that a suitable service factor can be chosen. Service factors are used to take into consideration intangible operating conditions such as misalignments, vibrations, transient loads and shocks. The actual horsepower is multiplied by the service factor to obtain an equivalent horsepower and the gear unit selected must have a rating equal to or greater than the equivalent horsepower. Table 1 presents A.G.M.A. service factors for high speed units. A high speed unit is defined as operating with a pinion speed of 3600 revolutions per minute and higher, or pitch line velocities of 5000 feet per minute and higher (PLV = RPM x Pitch Diameter xπ/12). Standard catalog gear units are listed to approximately 20,000 feet per minute. Applications exceeding this speed must be considered special and exceptional care must be taken in their design and manufacture.

Planetary Gearing

With planetary gearing the transmitted load is shared between several meshes, therefore, gearbox envelope and weight can be significantly reduced compared to parallel shaft designs. Figure 5 illustrates single stage planetary configurations.

In addition to achieving minimum weight and envelope, the small, stiff components used in planetary gearing result in reduced noise and vibration and high efficiency.

When confronted with a choice between parallel shaft and planetary gearing, it would appear that the planetary is more expensive and has more components, however, for high power, high speed applications, the reduced pitch line velocity and smaller components make planetary configurations very attractive and there is a trend toward this type of gearing.

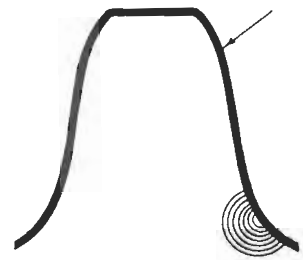
GEAR DESIGN

Modern design methods for turbomachinery drives make use of sophisticated analytical procedures. Computer programs enable the designer to optimize gearbox components with great precision.

Detail gear tooth designs are defined using mathematical models describing the major failure modes of turbomachinery gearing; pitting, breakage and scoring.

- *Pitting*, Fig. 6, is a fatigue phenomenon and occurs as a result of repeated stress cycles which lead to surface and subsurface cracks. Eventually particles detach and pits form. Pitting, although affected by the oil film, can occur even with an adequate film thickness.
- *Breakage*, Fig. 7, of gear teeth is caused by the root bending stress imposed by the transmitted load. Generally, tooth breakage is caused by bending fatigue rather than a transient overload exceeding the gear tooth fracture strength. In some cases pitting or wear may weaken the tooth to the extent that breakage occurs.
- *Scoring*, Fig. 8, is a form of surface damage on the tooth flanks which occurs when the lubricant film fails allowing metal to metal contact. Local welding is initiated and the welded junctions are torn apart by the relative motion of the meshing gear teeth. The flash temperature index [1] appears to be the most reliable method of analysis used at present to predict scoring.

BENDING STRESS



$$S_t = \frac{W_t}{F}\,\frac{DP}{J}$$

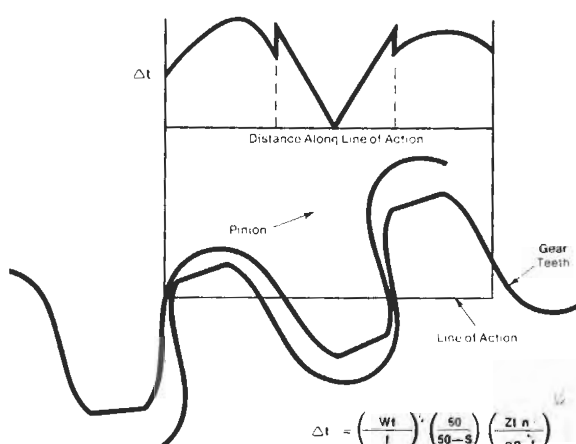
W = Tangential Load
F = Face Width
DP = Diametral Pitch
J = Geometry Factor

$$J = \frac{Y}{(K)\,(M)}$$

Y = Form Factor
K = Stress Concentration
M = Load Sharing

Figure 7. Calculation of Bending Stress (Breakage Criterion)

FLASH TEMPERATURE RISE



1/2

$$\Delta t = \left(\frac{Wt}{F}\right)\left(\frac{50}{50-S}\right)\left(\frac{Zt\,n}{DP}\right)$$

Wt = Tangential Load
F = Face Width
S = Surface Finish
Z = Geometry Factor
n = RPM
DP = Diametral Pitch

Figure 8. Calculation of Flash Temperature Rise (Scoring Criterion)

TEST CASE, 4310 HP, 15000 RPM IN, 4.7 RED. RATIO

AMERICAN LOHMANN GEAR ANALYSIS

	DRIVER PINION	DRIVEN EXTERNAL GEAR
NUMBER OF TEETH	36.0000000	169.0000000
HELIX ANGLE (DEG)	9.7429000	9.7429000
PITCH DIAMETER	4.5658537	21.4341463
RELATIVE ROLLING SPEED (RPM)	15000.0000000	3195.2662722
MESH TORQUE (IN-LBS)	17352.3833333	81462.1467593
BENDING GEOMETRY FACTOR	0.5000000	0.5000000
BENDING STRESS (PSI)	36807.3957527	36807.3957527
BENDING LIFE (HOURS)	999999.0000000	999999.0000000
BENDING SAFETY FACTOR	1.3584226	1.3584226
COMPRESSIVE STRESS (PSI)	104597.1429002	104597.1429002
COMPRESSIVE LIFE (HOURS)	999999.0000000	999999.0000000
COMPRESSIVE SAFETY FACTOR	1.4436341	1.4436341
SLIDING VELOCITY AT TIP (FPM)	2603.3564427	-2704.1028425
A.G.M.A. MATERIAL GRADE	2.0000000	2.0000000
ALTERNATING BENDING FACTOR	1.0000000	1.0000000
NUMBER OF MESHES PER REV	1.0000000	1.0000000
BASE HELIX ANGLE (DEG)	9.8223943	9.8223943
OUTSIDE DIAMETER	4.820- 4.825	21.678-21.683
PITCH DIAMETER	4.5658537	21.4341463
FORM DIAMETER	4.3534211	21.2061243
BASE DIAMETER	4.1272171	19.3749915
ROOT DIAMETER	4.196- 4.216	21.054-21.074
ROLL ANGLE-MAX OUTSIDE DIA	34.6968970	28.7871331
ROLL ANGLE-ROUND EDGE DIA	34.2997150	28.6812307
ROLL ANGLE-HIGH SINGLE TOOTH	29.2278037	27.6221242
ROLL ANGLE-PITCH DIA	27.1084259	27.1084259
ROLL ANGLE-LOW SINGLE TOOTH	24.6968981	26.6569603
ROLL ANGLE-FORM DIAMETER	19.2278047	25.4919469
TOP LAND THICKNESS	0.0656685	0.0650168
MAX CASE DEPTH	0.0566829	0.0526829
TRANSVERSE CIR TOOTH THICKNESS	0.1984669	0.1849790
NORMAL CIR TOOTH THICKNESS	0.1946- 0.1966	0.1813- 0.1833
NORMAL DIAMETRAL PITCH	7.9999990	7.9999990
NORMAL PRESSURE ANGLE	24.9999846	24.9999846
LEAD	83.5395087	392.1715827
ROUND EDGE RADIUS MAX	0.0100000	0.0100000
ROOT FILLET RADIUS MIN	0.0548491	0.0497956
WHOLE DEPTH CONSTANT	2.4000000	2.4000000
CLEARANCE AT TIP OF TOOTH	0.0503912	0.0503912
BALL DIAMETER	0.1250000	0.1250000
MEASUREMENT OVER BALLS	4.5515- 4.5565	21.3979-21.4024
BALL CONTACT DIAMETER	4.3833- 4.3881	21.2231-21.2275
DIM OVER TOP LAND	0.1367593	0.1425361

TRANSVERSE DIAMETRAL PITCH	7.8846154
TRANSVERSE PRESSURE ANGLE(DEG)	25.3203000
CENTER DISTANCE	13.0000000
PITCH LINE VELOCITY (FPM)	17930.1067096
MESH RATIO	4.6944444
RELATIVE HORSEPOWER PER MESH	4130.0000000
EFFECTIVE FACE WIDTH	4.5000000
STATIC TANGENTIAL LOAD (LBS)	7601.1561610
DYNAMIC FACTOR	1.0468457
ALIGNMENT FACTOR	1.3200026
MODIFIED TANGENTIAL LOAD (LBS)	10503.5739099
SURFACE FINISH	20.0000000
FLASH TEMPERATURE RISE (DEG F)	90.7282250
PROFILE CONTACT RATIO	1.5470330
FACE CONTACT RATIO	1.9392022
MIN CONTACT LENGTH	6.9803297
MAXIMUM CONTACT LENGTH	7.1231029
BACKLASH	0.0150000
CIRCULAR PITCH	0.3984459
LEAD ERROR (IN/IN)	0.0002000
BASE PITCH	0.3601677
DEPTH TO POINT OF MAX SHEAR	0.0101620

Figure 9. American Lohmann Gear Analysis Computer Output Sheet

The equations shown in Figs. 6 and 7 are in a basic form. In practice, when applied in the A.G.M.A. load rating system, various factors are applied which account for detrimental operating conditions. These factors are listed in Table 2 to illustrate the many parameters that have to be considered. References [1], [2] and [3] illustrate the use of the stress and scoring equations and list allowable design values.

As mentioned earlier, detailed gear tooth geometry is optimized through the use of computer analysis. Figure 9 presents a typical computer output sheet defining gear tooth stresses and geometry. Gear pitch diameters are generally determined by compressive stress considerations, however, the detail tooth design is defined by the flash temperature rise along the line of action. To optimize a given design, the maximum instantaneous flash temperature rise during the arc of approach must equal the maximum instantaneous flash temperature rise during the arc of recess. This is accomplished automatically by the use of iteration techniques built into the computer program. The computer program, starting with standard gear tooth addendums, will automatically vary the pinion and gear addendums in defined increments until the optimum flash temperature condition is obtained. With the resulting "long and short" addendum designs of this nature, standard tooth thicknesses are no longer applicable. If standard tooth thicknesses are utilized, an unbalance of bending stresses between pinion and gear would result. To optimize the bending stresses, the program enters a second iteration procedure which varies tooth thickness until bending stresses are balanced.

Tooth Modifications

If gear teeth were manufactured perfectly and there were no deflections during operation, the tooth form would be a pure involute. Because there are always manufacturing errors and tooth deflections, gear teeth are relieved at the tip and/or flank, Fig. 10. These tooth modifications allow the gears to enter mesh smoothly and reduce dynamic loading, vibration and noise. Typical tooth modifications are in the order of .0004 - .0010 inches.

In order to reduce the effect of misalignment which causes end loading, gears are sometimes crowned in the axial direction (lead modification). Figure 10 illustrates this technique. The amount of crowning at the tooth end is generally in the order of .001 inches.

High speed turbomachinery gearing requires excellent quality control. AGMA Standard 390.03, January 1973, recommends gear specifications for quality. AGMA Quality Classes are numbered from 3 to 15 with 15 being the most precise. The majority of turbomachinery gears will fall in the 10-13 Quality ranges. Table 3, from AGMA 390.03, presents typical quality data.

Following is a definition of each tooth tolerance element shown in Table 3:

- Runout Tolerance - The amount of pitch line runout allowed. Checked by such methods as indicating over pins placed in successive tooth spaces or rolling with a master gear of known accuracy.
- Pitch Tolerance - The allowable amount of variation between corresponding points on adjacent teeth (tooth to tooth).
- Profile Tolerance - The amount of vari-

Table 2. Factors Used in AGMA Rating Equations

Factor	Strength	Durability
Dynamic Load	K_v	C_v
Overload	K_o	C_o
Size	K_s	C_s
Hardness Ratio		C_h
Life	K_l	C_l
Temperature	K_t	C_t
Factor of Safety	K_r	C_r
Load Distribution	K_m	C_m
Surface Condition		C_f

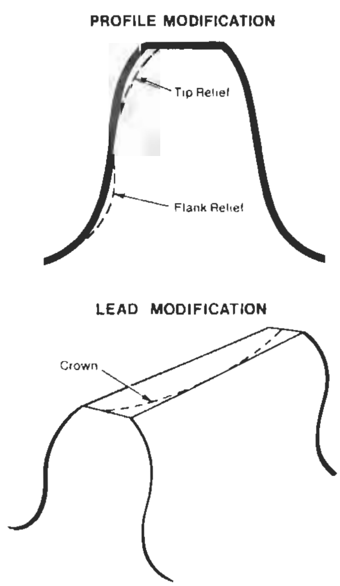


Figure 10. Illustration of Gear Tooth Modifications

Table 3. Gear Tooth Tolerances

(Courtesy American Gear Manufacturers Association)

COARSE–PITCH GEAR TOLERANCES

Tolerances in Ten-Thousandths of an Inch

RUNOUT TOLERANCE — PITCH DIAMETER (INCHES)

AGMA QUALITY NUMBER	NORMAL DIAMETRAL PITCH	3/4	1½	3	6	12	25	50	100	200	400
8	1/2					146.5	174.5	205.8	242.7	286.3	337.6
	1				88.8	104.8	124.8	147.2	173.6	204.7	241.4
	2			53.9	63.5	74.9	89.2	105.2	124.1	146.3	172.6
	4		32.7	38.5	45.4	53.6	63.8	75.2	88.7	104.6	123.4
	8	19.8	23.3	27.5	32.5	38.3	45.6	53.8	63.4	74.8	88.2
	12	16.3	19.2	22.6	26.7	31.5	37.5	44.2	52.1	61.5	72.5
	20	12.7	15.0	17.7	20.8	24.6	29.3	34.5	40.7	48.0	56.6
9	1/2					104.7	124.7	147.0	173.4	204.5	241.2
	1				63.5	74.8	89.1	105.1	124.0	146.2	172.4
	2			38.5	45.4	63.5	63.7	75.2	88.6	104.5	123.3
	4		23.3	27.5	32.4	38.3	45.6	53.7	63.4	74.7	88.1
	8	14.1	16.7	19.7	23.2	27.4	32.6	38.4	45.3	53.4	63.0
	12	11.6	13.7	16.2	19.1	22.5	26.8	31.6	37.2	43.9	51.8
	20	9.1	10.7	12.6	14.9	17.6	20.9	24.7	29.1	34.3	40.4
10	1/2					74.8	89.0	105.0	123.8	146.1	172.3
	1				45.3	53.5	63.7	75.1	88.5	104.4	123.2
	2			27.5	32.4	38.2	45.5	53.7	63.3	74.7	88.1
	4		16.7	19.6	23.2	27.3	32.5	38.4	45.3	53.4	63.0
	8	10.1	11.9	14.0	16.6	19.5	23.3	27.4	32.4	38.2	45.0
	12	8.3	9.8	11.5	13.6	16.1	19.1	22.6	26.6	31.4	37.0
	20	6.5	7.6	9.0	10.6	12.5	14.9	17.6	20.8	24.5	28.9
11	1/2					53.4	63.6	75.0	88.5	104.3	123.0
	1				32.4	38.2	45.5	53.6	63.2	74.6	88.0
	2			19.6	23.1	27.3	32.5	38.3	45.2	53.3	62.9
	4		11.9	14.0	16.6	19.5	23.2	27.4	32.3	38.1	45.0
	8	7.2	8.5	10.0	11.8	14.0	16.6	19.6	23.1	27.3	32.2
	12	5.9	7.0	8.2	9.7	11.5	13.7	16.1	19.0	22.4	26.4
	20	4.6	5.5	6.4	7.6	9.0	10.7	12.6	14.8	17.5	20.6
12	1/2					38.1	45.4	5.36	63.2	74.5	87.9
	1				23.1	27.3	32.5	38.3	45.2	53.3	62.8
	2			14.0	16.5	19.5	23.2	27.4	32.3	38.1	44.9
	4		8.5	10.0	11.8	13.9	16.6	19.6	23.1	27.2	32.1
	8	5.2	6.1	7.2	8.5	10.0	11.9	14.0	16.5	19.5	23.0
	12	4.2	5.0	5.9	6.8	8.2	9.8	11.5	13.6	16.0	18.9
	20	3.3	3.9	4.6	5.4	6.4	7.6	9.0	10.6	12.5	14.7

PITCH TOLERANCE — PITCH DIAMETER (INCHES)

AGMA QUALITY NUMBER	NORMAL DIAMETRAL PITCH	3/4	1½	3	6	12	25	50	100	200	400
8	1/2					19.0	21.7	24.5	27.7	31.3	35.4
	1				14.4	16.3	18.6	21.0	23.7	26.8	30.3
	2			10.9	12.3	14.0	15.9	18.0	20.3	23.0	26.0
	4		8.3	9.3	10.6	11.9	13.6	15.4	17.4	19.7	22.2
	8	6.3	7.1	8.0	9.0	10.2	11.7	13.2	14.9	16.8	19.0
	12	5.7	6.5	7.3	8.3	9.3	10.6	12.0	13.6	15.4	17.4
	20	5.1	5.8	6.5	7.4	8.3	9.5	10.7	12.1	13.7	15.5
9	1/2					13.4	15.3	17.3	19.5	22.1	24.9
	1				10.2	11.5	13.1	14.8	16.7	18.9	21.4
	2			7.7	8.7	9.8	11.2	12.7	14.3	16.2	18.3
	4		5.8	6.6	7.4	8.4	9.6	10.8	12.2	13.8	15.7
	8	4.4	5.0	5.6	6.4	7.2	8.2	9.3	10.5	11.9	13.4
	12	4.0	4.6	5.1	5.8	6.6	7.5	8.5	9.6	10.8	12.2
	20	3.6	4.1	4.6	5.2	5.9	6.7	7.6	8.5	9.7	10.9
10	1/2					9.4	10.8	12.2	13.7	16.5	17.6
	1				7.2	8.1	9.2	10.4	11.8	13.3	15.0
	2			5.4	6.1	6.9	7.9	8.9	10.1	11.4	12.9
	4		4.1	4.6	5.2	5.9	6.7	7.6	8.6	9.8	11.0
	8	3.1	3.5	4.0	4.5	5.1	5.8	6.5	7.4	8.3	9.4
	12	2.8	3.2	3.6	4.1	4.6	5.3	6.0	6.7	7.6	8.6
	20	2.5	2.9	3.2	3.7	4.1	4.7	5.3	6.0	6.8	7.7
11	1/2					6.6	7.6	8.6	9.7	10.9	12.4
	1				5.0	5.7	6.5	7.3	8.3	9.4	10.6
	2			3.8	4.3	4.9	5.6	6.3	7.1	8.0	9.1
	4		2.9	3.3	3.7	4.2	4.8	5.4	6.1	6.9	7.8
	8	2.2	2.5	2.8	3.2	3.6	4.1	4.6	5.2	5.9	6.6
	12	2.0	2.3	2.6	2.9	3.3	3.7	4.2	4.7	5.4	6.1
	20	1.8	2.0	2.3	2.6	2.9	3.3	3.7	4.2	4.8	5.4
12	1/2					4.7	5.3	6.0	6.8	7.7	8.7
	1				3.5	4.0	4.6	5.2	5.8	6.6	7.5
	2			2.7	3.0	3.4	3.9	4.4	5.0	5.6	6.4
	4		2.0	2.3	2.6	2.9	3.3	3.8	4.3	4.8	5.5
	8	1.5	1.7	2.0	2.2	2.5	2.9	3.2	3.7	4.1	4.7
	12	1.4	1.6	1.8	2.0	2.3	2.6	3.0	3.3	3.8	4.3
	20	1.3	1.4	1.6	1.8	2.0	2.3	2.6	3.0	3.4	3.8

PROFILE TOLERANCE — PITCH DIAMETER (INCHES)

AGMA QUALITY NUMBER	NORMAL DIAMETRAL PITCH	3/4	1½	3	6	12	25	50	100	200	400
8	1/2					42.6	47.7	53.1	59.1	65.7	73.1
	1				28.3	31.5	35.3	39.3	43.7	48.6	54.1
	2			18.8	21.0	23.3	26.1	29.0	32.3	36.0	40.0
	4		12.5	13.9	15.5	17.2	19.3	21.5	23.9	26.6	29.6
	8	8.3	9.3	10.3	11.5	12.8	14.3	15.9	17.7	19.7	21.9
	12	7.0	7.8	8.6	9.6	10.7	12.0	13.3	14.8	16.5	18.4
	20	5.6	6.2	6.9	7.7	8.6	9.6	10.7	11.9	13.2	14.7
9	1/2					30.4	34.1	37.9	42.2	46.9	52.2
	1				20.2	22.5	25.2	28.1	31.2	34.7	38.6
	2			13.5	15.0	16.7	18.6	20.7	23.1	25.7	28.6
	4		8.9	10.0	11.1	12.3	13.8	15.3	17.1	19.0	21.1
	8	5.9	6.6	7.4	8.2	9.1	10.2	11.4	12.6	14.1	15.6
	12	5.0	5.5	6.2	6.9	7.6	8.6	9.5	10.6	11.8	13.1
	20	4.0	4.4	4.9	5.5	6.1	6.8	7.6	8.5	9.4	10.5
10	1/2					21.7	24.3	27.1	30.1	33.5	37.3
	1				14.5	16.1	18.0	20.0	22.3	24.8	27.6
	2			9.6	10.7	11.9	13.3	14.8	16.5	18.3	20.4
	4		6.4	7.1	7.9	8.8	9.9	11.0	12.2	13.6	15.1
	8	4.2	4.7	5.3	5.9	6.5	7.3	8.1	9.0	10.0	11.2
	12	3.6	4.0	4.4	4.9	5.5	6.1	6.8	7.6	8.4	9.4
	20	2.9	3.2	3.5	3.9	4.4	4.9	5.4	6.1	6.7	7.5
11	1/2					15.5	17.4	19.3	21.5	24.0	26.7
	1				10.3	11.5	12.9	14.3	15.9	17.7	19.7
	2			6.9	7.6	8.5	9.5	10.6	11.8	13.1	14.6
	4		4.6	5.1	5.6	6.3	7.0	7.8	8.7	9.7	10.8
	8	3.0	3.4	3.8	4.2	4.6	5.2	5.8	6.4	7.2	8.0
	12	2.5	2.8	3.1	3.5	3.9	4.4	4.9	5.4	6.0	6.7
	20	2.0	2.3	2.5	2.8	3.1	3.5	3.9	4.3	4.8	5.4
12	1/2					11.1	12.4	13.8	15.4	17.1	19.0
	1				7.4	8.2	9.2	10.2	11.4	12.7	14.1
	2			4.9	5.5	6.1	6.8	7.6	8.4	9.4	10.4
	4		3.3	3.6	4.0	4.5	5.0	5.6	6.2	6.9	7.7
	8	2.2	2.4	2.7	3.0	3.3	3.7	4.1	4.6	5.1	5.7
	12	1.8	2.0	2.2	2.5	2.8	3.1	3.5	3.9	4.3	4.8
	20	1.5	1.6	1.8	2.0	2.2	2.5	2.8	3.1	3.4	3.8

LEAD TOLERANCE — FACE WIDTH (INCHES)

AGMA QUALITY NUMBER	1 and Less	2	3	4	5
8	5	8	11	13	16
9	4	7	9	11	13
10	3	5	7	9	10
11	3	4	6	7	8
12	2	3	5	6	7

ation allowed in the functional profile (involute).
- Lead Tolerance - Allowable variation of the tooth surface along the face width.

Gear Materials and Heat Treatment
Proper choice of gear material and heat treatment is probably the most important factor in the successful operation of a gear set. In choosing a gear material the tooth hardness and type of heat treatment to achieve that hardness must be considered.

Gears, case hardened by carburizing, provide the best metallurgical characteristics combining good case structure with reasonable ductility. Carburizing produces the "strongest" gear providing bending and pitting fatigue resistance and an excellent wear surface. A disadvantage of carburizing is that gear teeth distort during the heat treatment and, in order to obtain high precision, grinding after hardening is required. Surface hardnesses attained by carburizing are in the order of Rc 55-62. For critical applications Rc 60 minimum should be specified.

The best carburizing steel to achieve high load carrying capacity is SAE 9310 (AMS 6260 or AMS 6265). Other carburizing steels used include SAE 8620, 4620, 4320 and 3310.

The nitriding process is used to case harden gears when distortion must be held to a minimum. Often the gears are finish cut and then nitrided, eliminating the grinding requirement. Nitrided gears do not have the bending and pitting fatigue resistance of carburized gears but do provide a hard, wear resistant case. With nitriding steel such as AMS 6475 (nitralloy N) or AMS 6470 (nitralloy 135) case hardnesses of Rc 65-70 can be achieved. Steels such as SAE 4140 and 4340 are nitrided to hardnesses of 320 to 380 BHN.

Through hardened gears in turbomachinery units are generally in the 300-400 BHN hardness range. Typical through hardening steels are SAE 4140 and 4340.

Often, combinations of pinion and gear materials and heat treatments are used. For instance, if the gear is very large it may not be practical to harden, and the gear set might consist of a carburized and ground pinion driving a through hardened gear.

Bearings
High speed turbomachinery gearboxes generally incorporate journal bearings. Journal bearings offer the possibility of very long life whereas anti-friction bearing fatigue life calculations generally result in finite lives which may not be acceptable. There is also a feeling that journal bearings have far fewer components and therefore are more reliable. Also, journal bearing failure modes are less catastrophic than those of anti-friction bearings and can be detected in time for replacement to be effected.

Journal bearing materials are usually steel lined babbitt or a tri-metal constructon where a high strength, high temperature bearing material is sandwiched between a thin babbitt layer and the steel backing.

Radial journal bearings may have round bores or various other shapes in cases where dynamic instabilities are possible.

Thrust bearings are either micheled for light loads or have tapered lands or tilting pads.

Seals
The most common method of sealing high speed turbomachinery gearboxes is by the use of non-contacting labyrinth seals. Two sets of labyrinths are used with a drain in between which leads back to the oil sump. If air pressure is available the labyrinth can be internally pressurized with 1 or 2 psig.

GEARBOX INSTALLATION

Installation of the transmission is critical for proper performance. The gearbox must be rigidly connected to the foundation which must also be rigid and have a flat mounting surface. If the unit is to be mounted on a surface other than horizontal the manufacturer should be consulted to determine if any lubrication system problems or other difficulties will arise due to the gearbox attitude.

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

If the transmission is to be mounted on a pedestal or base plate the structure must be carefully analyzed to determine if it will withstand operating loads without excessive deflection. When mounting the unit on steel beams a base plate should be used. The base plate should extend under the entire gearbox and be at least as thick as the gearbox base. Both the unit and the base plate should be rigidly bolted to the steel supports with proper shimming to achieve a level surface. If the gearbox is to be mounted on a concrete foundation, grout steel mounting pads into the concrete base rather than grouting the transmission directly into the concrete. This will facilitate any shimming and alignment required. The concrete should be set before bolting down the gearbox and cured before loading is applied.

The gearbox can be brought into alignment by placing broad flat shims under all mounting pads. A feeler gage is used to determine that the thickness of shims is correct under all mounting pads.

The gearbox input and output shafts must be correctly aligned with the driving and driven machine shafts. Even if the total system is delivered on a permanent mounting already aligned at the factory, couplings should be disconnected and alignment rechecked. The alignment may have been disturbed during shipment.

Alignment of the couplings can be performed in two phases. First the components are roughly aligned by eye and the final adjustment is accomplished with the use of measuring instruments. When accomplishing the rough alignment, the unit being moved into alignment is adjusted until the coupling half rims are close horizontally and vertically. Also, the coupling halves axial distance should be as specified. When the rough alignment is completed, the bolts around the base of the unit should be secured but not tightened.

The angular alignment is then checked by mounting a dial indicator on the hub of one coupling with the indicator pin on the face of the mating coupling. The shaft on which the indicator is mounted is rotated and the indicator reading recorded. The unit is adjusted until the specified indicator reading is achieved. A reasonable value might be .003 inches TIR. The dial indicator is then attached to the opposing coupling hub and the procedure done again.

The parallel alignment is then checked by mounting a dial indicator on the hub of one coupling with the indicator pin on the outside diameter of the mating coupling. Again, the shaft is rotated and the unit adjusted until the specified reading is achieved. The indicator is then attached to the opposing coupling half and the procedure done again.

There may be several iterations until both angular and parallel misalignment requirements are satisfied. The mounting bolts should then be torqued and the unit operated until it attains normal temperatures. At this point, the alignment should be rechecked and adjusted if necessary.

If couplings, sprockets, pulleys, or gears are installed in the field do not hammer them into position. This might damage bearings or gears. Instead heat the component and slide it on.

Special attention must be given to back-stops, which should be checked before start-up. The shaft should be turned by hand to determine if the backstop is assembled correctly for the direction of rotation.

LUBRICATION SYSTEM

The purpose of the gearbox lubrication system is to provide an oil film at the contacting surfaces of working components and absorb heat generated in the gearbox so that component temperatures are not excessive. The majority of the oil flow is required for the cooling function. For every gear drive there is a mechanical rating; the load the transmission can transmit based on stress and wear considerations. In addition, there is a thermal rating which is the average power that can be transmitted continuously without overheating and without using special cooling. AGMA thermal ratings are based on a maximum oil sump temperature of 200°F[4]. In turbomachinery applications, the thermal rating is usually less than the mechanical rating and an external cooling system is required. Figure 11 presents a typical gearbox lubrication system in schematic form including pump, cooler, filter and pressure relief valve.

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

$$\Delta t = Q / M\,C_p$$

Where: Δt = temperature rise - °F
M = Flow - Lb./min.
(Note - 1 GPM ≅ 7.5 Lb/min.)
Q = Heat - BTU/min.
(Note - Q = HP x 42.4)
C_p = Specific Heat ≅ .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. Typical operating temperatures for turbomachinery gearboxes are oil "in" of 130°F with a rise of 30°F. These values are for mineral oils; synthetic oils may operate at higher temperature levels.

The amount of oil flow supplied to a gear mesh is generally based on experience and experimental data. A rule of thumb sometimes used is .02 Lbs./min./HP. Oil may be jetted going into or out of mesh or both. For high speed gearing, it is generally agreed the majority of oil should be jetted to the outgoing side to achieve better cooling and reduce losses.

The quantity of flow passing through a gearbox is regulated by the oil jet diameters, journal bearing clearances and feed pressure. Gearbox feed pressures are generally in the order of 20-100 psig. To hold a constant feed pressure, a pressure regulating valve is incorporated at the gearbox inlet as shown in Fig. 11.

Oil pumps for gearboxes are positive displacement types such as gear, vane or screw pumps. The pump capacity will be somewhat oversized to account for variations in gearbox flow and pump deterioration. For instance, a 23 GPM pump might be chosen for a 20 GPM application. The output of positive displacement pumps is directly proportional to pump speed and at any given speed is constant regardless of pressure. The pump must develop sufficient pressure to supply the gearbox feed pressure and make up the pressure drops in the cooler, filter and oil lines and fittings. Therefore, a gearbox with a 50 psig design feed pressure may have an oil pump operating at 100 psig. Sometimes a high pressure relief valve is incorporated at the pump to protect the pump in case a downstream line or valve is inadvertently closed.

The oil cooler is sized on the basis of the magnitude of cooling required and the pressure drop for the rated oil flow. Both air/oil and water/oil coolers are used. If cold starts are anticipated a thermal bypass can be incorporated in the cooler. Dual oil coolers with a changeover valve are sometimes specified for critical applications.

The oil filter is sized on the basis of dirt capacity and pressure drop for the rated oil flow. The degree of filtration is indicated by the micron rating of the filter element. Typical turbomachinery gearbox filters are in the order of 40 microns. Bearing in mind that a micron is .000040 inches a case can be made for using fine filtration of 10 microns or better. For instance, typical gear tooth and bearing oil film thicknesses are in the order of .0005 inches, therefore, a 10 micron maximum filter is required if protection is desired against particles larger than the oil film thickness. As the oil filter collects debris, the pressure drop across the filter increases. The filter, therefore, should incorporate a bypass valve and an impending bypass warning. Dual oil filters with changeover valves are sometimes specified for critical applications.

Lubrication system components can be supplied either by the gear manufacturer or 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.

A.G.M.A. Standard 250.03, "Lubrication of Industrial Enclosed Drives", May 1972 gives recommendations for the grades of oils to be used in gearboxes. Lubricant viscosity recommendations are specified by AGMA Lubricant Numbers as shown in Table 4. The former AGMA viscosity system (AGMA 250.02, December 1955 and AGMA 252.01, May 1959) has been abandoned in favor of a newer, more universal system as shown in Table 4A. The AGMA Lubricant Numbers have been retained but the corresponding viscosity ranges will now conform to the ASTM-ASLE system (ASTMD-2422 "Viscosity System for Industrial Fluid Lubricants") which is widely accepted as the standard of the petroleum industry.

AGMA Lubricant number recommendations for drives using all types of gearing except worm gearing are given in Table 5. It can be seen from Table 5 that most turbomachinery gearboxes will utilize an AGMA type 1 or 2 lubricant.

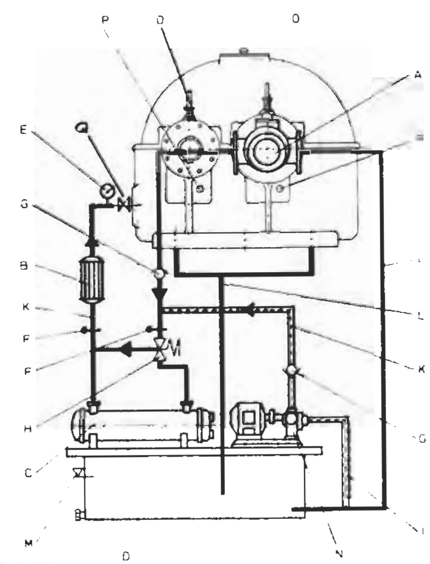
Lubricant maintenance, oil change intervals, flushing, etc., is discussed in the maintenance section.

GEAR TESTING AND OPERATION

Turbomachinery gear units are generally tested by the manufacturer prior to delivery. The type of test should be agreed to in advance. A typical light load, full speed acceptance test is as follows [5]:

1. Operate the gearbox at maximum continuous speed until bearing and lubrication oil temperature has stabilized.
2. Increase the speed to 110 percent of maximum continuous speed and run for a minimum of 15 minutes.
3. Reduce speed to maximum continuous and run for four hours at a minimum.

The following measurements should be made



Legend:

A Oil pump
B Oil filter
C Oil cooler
D Oil tank
E Pressure gauge
F Thermometer
G Check valve
H Bypass
I Suction pipe
K Pressure pipe
L Return flow pipe
M Oil level gauge
N Primer and spare pump unit
O Temperature gauge
P Connection for remote thermometer
Q Pressure regulator

Figure 11. Typical Gearbox Lubrication System

Table 4. AGMA Lubricants
(Courtesy American Gear Manufacturers Association)

AGMA STANDARD SPECIFICATION
LUBRICATION OF INDUSTRIAL ENCLOSED GEAR DRIVES
Viscosity Ranges for AGMA Lubricants

Rust and Oxidation Inhibited Gear Oils	Viscosity Range ASTM System (2)	Extreme Pressure Gear Lubricants (5)
AGMA Lubricant No.	SSU at 100°F	AGMA Lubricant No.
1 2 3	193 to 235 284 to 347 417 to 510	2 EP 3 EP
4 5 6	626 to 765 918 to 1122 1335 to 1632	4EP 5EP 6EP
7 comp. (1) 8 comp. (1) 8A comp. (1)	1919 to 2346 2837 to 3467 4171 to 5098	7EP 8EP

NOTE: Viscosity ranges for AGMA lubricant numbers will henceforth be identical to those of the ASTM System (2).

Table 4A. AGMA Lubricants
(Courtesy American Gear Manufacturers Association)

Equivalent Viscosities of Other Systems (For Reference Only)

AGMA Lubricant No.	Equivalent ASTM-ASLE Grade No. (2)	Metric Equivalent Viscosity Ranges cST at 37.8°C (100°F)	Viscosity Ranges from Former AGMA System (3), (4)	
			SSU at 100°F	SSU at 210°F
1 2, 2EP 3, 3EP	S215 S315 S465	41.4 to 50.6 61.2 to 74.8 90 to 110	180 to 240 280 to 360 490 to 700	
4, 4EP 5, 5EP 6, 6EP	S700 S1000 S1500	135 to 165 198 to 242 288 to 352	700 to 1000	80 to 105 105 to 125
7 comp, 7EP 8 comp, 8EP 8A comp	S2150 S3150 S4650	414 to 506 612 to 748 900 to 1100		125 to 150 150 to 190 190 to 250

(1) Oils marked comp. are compounded with 3% to 10% fatty or synthetic fatty oils.
(2) "Viscosity System for Industrial Fluid Lubricants", ASTM 2422. Also British Standards Institute, B.S. 4231.
(3) AGMA 250.02 (December 1955).
(4) AGMA 252.01 (May 1959).
(5) Extreme pressure lubricants should be used *only* when recommended by the gear drive manufacturer.

Table 5. AGMA Lubricant Recommendations
(Courtesy American Gear Manufacturers Association)

AGMA STANDARD SPECIFICATION
LUBRICATION OF INDUSTRIAL ENCLOSED GEAR DRIVES

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

Type of Unit / Size of Unit	Other Lubricants		AGMA Lubricant Number	
	Ambient Temperature F			
Main Gear Low Speed Centers	-40 to 0	-20 to +25	15 to 60	50 to 125
Parallel shaft, (single reduction), up to 8 in.	Automatic Transmission Fluid (or similar product—see note 5)	SAE 10W/30 or 10W/40 Motor Oil (or similar product—see note 5)	2–3	3–4
Over 8 in. and up to 20 in.			2–3	4–5
Over 20 in.			3–4	4–5
Parallel shaft, (double reduction), up to 8 in.			2–3	3–4
Over 8 in. and up to 20 in.			3–4	4–5
Over 20 in.			3–4	4–5
Parallel shaft, (triple reduction), up to 8 in.			2–3	3–4
Over 8 in. and up to 20 in.			3–4	4–5
Over 20 in.			4–5	5–6
Planetary gear units				
O.D. Housing up to 16 in.			2–3	3–4
O.D. Housing over 16 in.			3–4	4–5
Spiral or Straight bevel gear units				
Cone distance up to 12 in.			2–3	4–5
Cone distance over 12 in.			3–4	5–6
Gearmotors and Shaft mounted units			2–3	4–5
High-Speed Units (See Note 4)			1	2

NOTES

1. Pour point of lubricant selected should be at least 10°F lower than the expected minimum ambient starting temperature. If ambient starting temperature approaches lubricant pour point, oil sump heaters may be required to facilitate starting and insure proper lubrication. (See paragraph 5.)*
2. Ranges are provided to allow for variations in operating conditions such as surface finish, temperature rise, loading, speed, etc.
3. AGMA viscosity number recommendations listed above refer to R & O gear oils shown in Table 2.* EP gear lubricants in the corresponding viscosity grades may be substituted where deemed necessary by the gear drive manufacturer.
4. High speed units are those operating at speeds above 3600 rpm or pitch line velocities above 5000 fpm. Refer to Standard AGMA 421.06, "Practice for High Speed Helical & Herringbone Gear Units," for detailed lubrication recommendations.
5. When they are available, good quality industrial oils having similar properties are preferred over the automotive oils. The recommendation of automotive oils for use at ambient temperatures below +15°F is intended only as a guide pending widespread development of satisfactory low temperature industrial oils. Consult gear manufacturer before proceeding.
6. Drives incorporating overrunning clutches as backstopping devices should be referred to the clutch manufacturer as certain types of lubricants may adversely affect clutch performance.

*Editors note: This refers to material outside this text.

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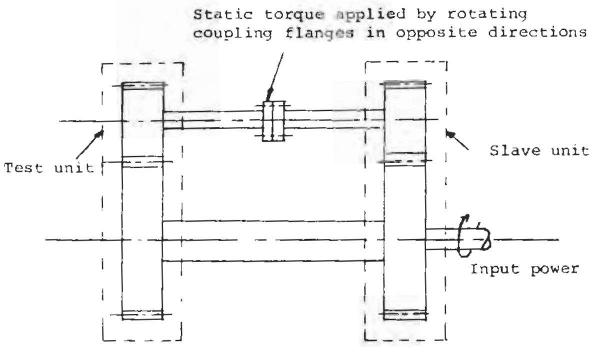


Figure 12. Back to Back Gear Test Rig

during the acceptance test:
- Oil inlet temperature
- Scavenge oil temperature
- Oil feed pressure
- Oil flow
- Shaft speed

Other parameters that can be measured during gear testing are:
- Vibration
- Shaft excursion
- Noise
- Bearing temperatures

A detailed test log should be kept making entries of each measurement at 15 minute intervals.

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

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

The full speed "light" load test is widely used since full load testing at a gear vendor's plant is costly and sometimes a prime mover of sufficient capacity is not available. If a full load test is required by the customer, the load can be applied by mechanical or hydraulic methods such as dynamometers, prony brakes, etc. Another method of full load and speed testing is back-to-back operation. Two identical units are coupled together, input shaft to input shaft, output shaft to output shaft. The desired torque is locked into the system by mechanical or hydraulic means. The unit is then operated at the desired speed by a prime mover which need only be large enough to make up the efficiency losses in the system. Figure 12 illustrates the simplest type of back-to-back test rig.

On occasion full load, low speed or static torque tests are employed to demonstrate tooth contact, load carrying capability and casing rigidity.

Initial Field Start-up

Prior to starting the equipment the following preliminary checks should be performed:
- Check oil level and ensure that proper oil is being used.
- Tighten all pipe connections.
- Check all electrical connections.
- Tighten all mounting and gearbox bolts with proper torque.
- Check the mounting of all gauges, switches, etc.
- Check all couplings for proper installation and alignment.
- Check inspection cover installations.

The following instructions pertain to the initial start-up:

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

with the recommended lubricant to the proper level.

7. After the initial 50 hours of operation check all coupling alignments and retorque all bolts. Check all piping connections and tighten if necessary.

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

Condition Monitoring

The most efficient method of maintaining a piece of equipment is to base repairs on the machine's condition rather than overhaul at an arbitrary time period. This type of predictive maintenance requires dependence upon instrumentation and the proper interpretation of the data it provides. Reference [6] provides a discussion of basic dynamic motion, vibration and rotor position parameters, that should be measured and analyzed. In addition to vibration and shaft position other parameters that are important to monitor are oil and bearing temperature, oil condition, and oil pressure.

Vibration Monitoring

Placement of vibration pick-ups at 90° to one another is common on gearbox casings particularly near the high speed shaft. The parameters monitored are generally amplitude of vibration in peak to peak mils displacement, velocity in inches per second or peak G's acceleration. Casing measurements are utilized to attempt to lead to the early discovery of malfunctions. A normal operating machine will generally have a stable amplitude reading of an acceptable low level. Any change in this amplitude reading indicates a change of machine condition and should be investigated. In analyzing machine operation spectrum analysis of the vibration pick-up can be used. Figure 13 [6], illustrates a comparison between two identical gearboxes. The top spectrum plot depicts a gearbox in good mechanical condition with reasonably low acceleration levels and a normal mixture of components. A similar measurement on a unit that had cracked pinion teeth is presented on the bottom plot. Note the high amplitudes at gear mesh frequency, the seventh harmonic of pinion rotational speed, and even the pinion running speed (X_{hs}) at 4.2 G's.

Position Measurements

Proximity probes, when mounted rigidly to a bearing cap or casing of a gearbox, provide a vibration measurement of the relative motion between the shaft and the mounting of the proximity probe. Bearing wear, both radial and axial, can be monitored by mounting two radial vibration probes adjacent to each radial bearing and one axial position probe for each shaft. Reference [5] gives the following allowable level for the double amplitude of unfiltered vibration in any plane measured on the shaft adjacent to a radial bearing:

2.0 mils or the value defined by the following equation, whichever is less.

$$\text{Double amplitude including runout in mils} = \left(\frac{12000}{\text{RPM}}\right)^{1/2}$$

Temperature Measurement

Monitoring of the oil in and out temperatures can provide information about the gearbox and lubrication system condition. Also, bearing scavenge oil temperature and babbitt temperature can be monitored. Drastic changes in temperature parameters require investigation.

Pressure Measurements

Oil pressure into the gearbox should be monitored to insure proper flow. The oil filter should have a differential pressure indicator which is monitored to determine whether the filter is becoming clogged.

Oil Inspections

Laboratory inspection of the lubricating oil can provide an indication of oil deterioration and gearbox condition. An oil sample can be analyzed for proper viscosity and acid number. Also, a spectographic analysis can be performed which indicates the quantity and type of metallic particles in the lubricant. This type of analysis, if performed periodically, can identify a component that is wearing.

Shut Down Switches

In some cases gearboxes are provided with instrumentation that automatically shuts down the equipment when predetermined levels are exceeded. Alternatively, warning lights can be activated. Parameters that are generally monitored are:

- High oil "in" temperature
- Low oil "in" pressure
- Filter differential pressure

Correlation of vibration and position data, temperatures, pressures and other external parameters which effect gearbox operation will lead to a good predictive maintenance program. Analysis of all the data will provide an indication not only of equipment malfunction but also which component is deteriorating.

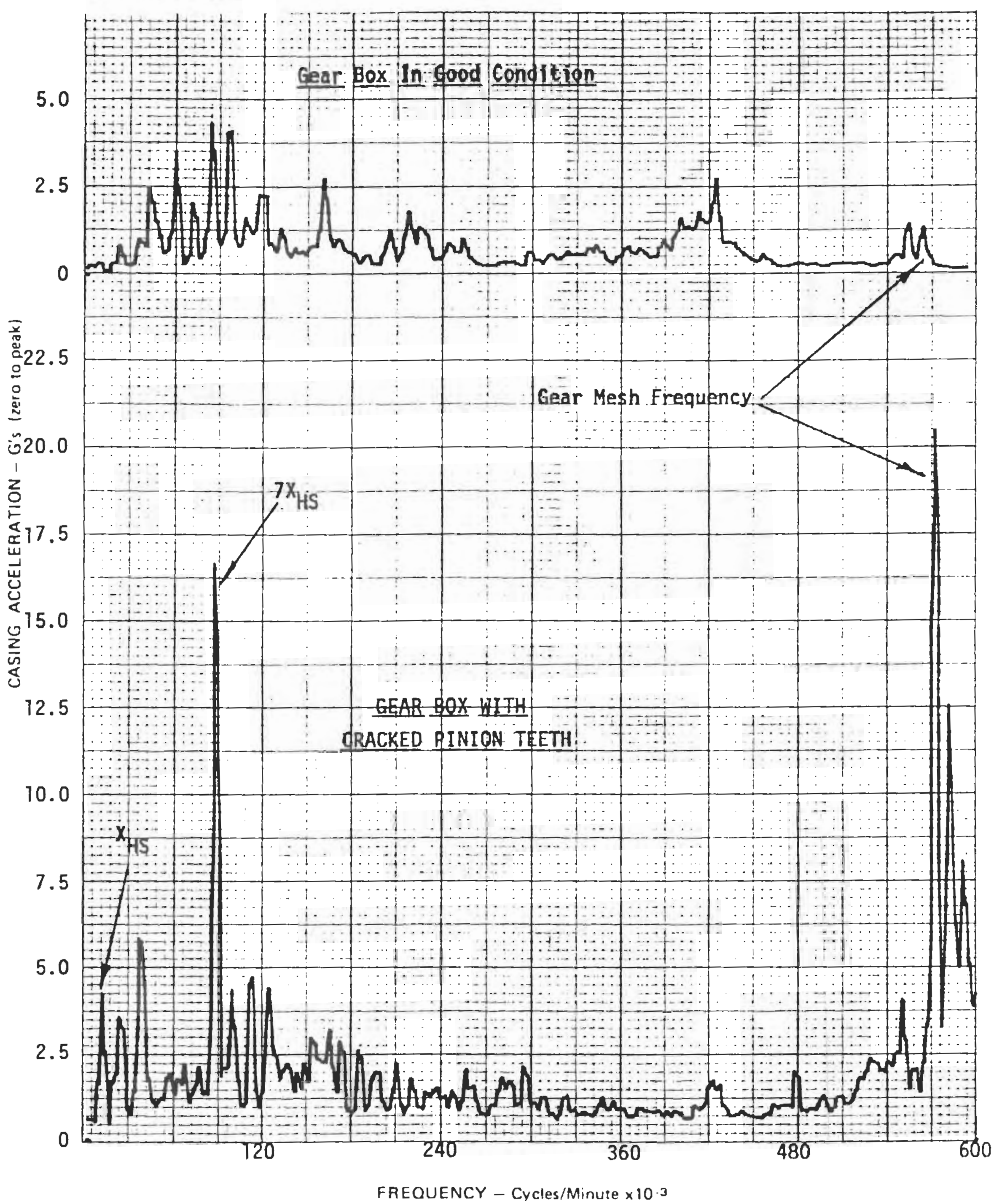


Figure 13. Comparison Between Gearbox Casing Acceleration Characteristics

MAINTENANCE

The objective of a maintenance program is to insure satisfactory gearbox performance at all times and maintain the transmission in a state of readiness if not in operation. A program should be planned which includes regular maintenance actions and monitoring of operating characteristics to determine whether the gearbox condition is deteriorating.

There are several maintenance actions that should be effected during initial operation of the gearbox. After approximately 50 hours of operation check coupling alignments and retorque all bolts. Check all piping connections and tighten if necessary. Also, after approximately 50 hours of operation, the oil in a new unit should be changed. This oil can be filtered through an element with the same micron rating as the gearbox filter and reused. Particles may be found in the oil due to the wearing in process. At this point the sump or reservoir should be thoroughly cleaned. After draining the original oil it is recommended that the gearbox and lubrication system be flushed out with a flushing oil. If possible bring the unit up to operating speed with light load after filling with flushing oil. Shut down immediately after achieving full speed. Then drain the flushing oil and refill with the recommended lubricant to the proper level.

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

A recommended maintenance schedule is presented in Table 6. Logs should be kept of instrument readings and maintenance action to keep a running account of gearbox condition.

The major causes of gear unit failure are improper lubrication and overload. Care must be taken to check for proper oil level before operation. Excessive oil volume can be as detrimental as lack of lubrication. Too much oil will result in churning and overtemperature of components.

Overload can be a result of vibration, shock, high torque at low speed, etc. If there is a possibility that operating loads will exceed rated unit loads the manufacturer should be consulted.

In many gear units the teeth can be visually inspected by removing inspection covers. When opening these inspection covers care must be taken to insure that no foreign material enters the gearbox. Gear teeth should be examined under good lighting and be wiped clean of oil to prevent a false diagnosis. The contact pattern should cover approximately 80% of the tooth. Each gear tooth should be examined for evidence of the conditions described in Table 7.

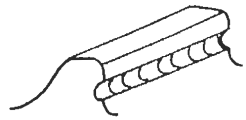
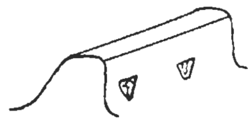
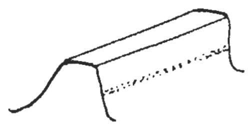
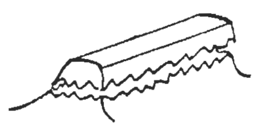
Gear teeth should be inspected for nicks, burrs

Table 6. Scheduled Maintenance Actions

FREQUENCY	MAINTENANCE ITEM	CORRECTIVE ACTION
Daily	Check oil temperature and pressure at operating conditions.	If there is a drastic change from previous readings stop unit and determine cause.
	Check for noise, vibration and oil leaks. Check sump oil level.	Add oil if necessary.
Weekly	Check oil filter.	Change filter element if necessary.
Monthly	Check lubricating oil for contamination.	Drain and refill lube system if necessary. Change oil filter.
	Check all gauges, controls, alarm systems. Clean breather element.	
*Every 2500 hours or 6 months	Change lubricating system oil.	

*If operating conditions are unusually severe such as high temperature or high moisture atmospheres, oil change requirements might be more frequent. Changes can be based on inspection of the oil for viscosity or acid number in such cases.

Table 7. Visual Inspection of Gears

CONDITION	DESCRIPTION	LIMITS	ILLUSTRATION
Wear	A general term describing loss of material from contacting surfaces of gear teeth.	Heavy wear to the extent that the working tooth surface is distorted is unacceptable. Light wear is acceptable.	
Pitting	A failure of the material characterized by the removal of metal and the formation of cavities. These cavities may be small and not progress; they may be small initially and then combine or increase in size by continued fatigue; or they may be of considerable size at the start.	Light pitting is acceptable. Any other condition of pitting is unacceptable.	
Spalling	Large cavities which may result from a material or processing defect or the forming of a large pit from several smaller pits.	Spalling is unacceptable.	
Frosting	A surface distress condition caused by welding of high spots during the break in period.	Light frosting is acceptable.	
Scoring	The removal of metal in direction of sliding due to a breakdown of the lubricant film.	Light scoring is acceptable.	
End Loading	A wear, pitting or scoring condition that is predominantly on one end of the tooth caused by misalignment of the mating tooth surface.	Light end wearing, scoring or pitting is acceptable.	
Breakage	Bending fatigue failure due to overload, misalignment, or from inadvertent stress concentrations such as notches in surface or sub-surface material defects.	Breakage is unacceptable.	
Cracking	Cracking is due to metallurgical or process problems.	Unacceptable	

NOTE: If conditions such as pitting, scoring or wear are noted, periodic inspections should be scheduled to determine the progression of these conditions.

Table 8. Journal-Bearing Trouble-Shooting Chart

CAUSE	REMEDY
I. WEAR, SCORING, GALLING:	
1. Dirt	Change oil, install new filter, remove source of contaminant, provide flushing chamfers at end of oil grooves, use softer bearing material to embed small amounts of dirt or a harder journal to reduce its effect.
2. Misalignment	Align journal with bearing.
3. Excessive journal deflection	Provide adequate journal support, reduce load causing deflection.
4. Boundary-film condition	Increase oil viscosity, use antiwear or extreme-pressure oil or grease, change to bearing material with better compatibility, change bearing design for greater oil-film thickness.
5. Too low oil viscosity	Use heavier-viscosity oil. Reduce bearing operating temperature by cooling oil, increase oil flow to lower the bearing temperature, modify bearing design to give higher load capacity.
6. Rough journal	Polish or grind journal.
7. Poor bearing-journal material combination	Select compatible materials, increase shaft hardness or use hardened sleeve on shaft. Use additive-type oils.
8. Load too high	Increase oil viscosity, increase bearing size, use stronger bearing material with hardened shaft, use antiwear or extreme pressure oil.
9. Vibration	Balance rotor, increase oil viscosity, use pressure bearing, three-lobe or elliptical type.
10. Journal eccentricity	Replace journal.
11. Improper grooving	Take grooves out of loaded zone in bearing.
12. Electrical pitting	Remove source of current flowing through bearing, electrically insulate bearing, ground journal.
13. Insufficient lubrication	Add oil; check pump, filter cooler, piping; increase oil-inlet pressure.
II. CORROSION	
1. Corrosive Contamination	Remove contamination source, atmospheric or otherwise, seal bearing, use corrosion resistant lubrication.
2. Moisture	Use rust inhibited oil, remove source of moisture.
3. Corrosive Lubricant	Change oil periodically to avoid build up of corrosive oxidation products, avoid possibly corrosive extreme pressure oils if not necessary, change to non-corrosible bearing material such as tin babbitt or aluminum.
III. OVERHEATING:	
1. Insufficient oil film	Increase bearing clearance, decrease oil viscosity, add more oil-distributing grooves, increase supply pressure, increase oil orifice size, increase chamfer size.
2. Improper oil viscosity	Change to proper viscosity for bearing load, speed and temperature.
3. Too little clearance	Scrape or machine out bearing bore.
4. Heat from surroundings	Provide circulating-oil system with external cooling, increase oil flow, use more viscous oil, water cooling, insulate bearing, reduce ambient temperature with better ventilation.
5. High load	Reduce load and belt tension, use compounded oil containing oiliness or extreme pressure agents. Change to stronger bearing material and a bearing designed for higher loads.
6. High speed	Decrease oil viscosity, increase clearance and oil flow. Change to elliptical, overshot, or multigroove bearing; decrease journal size.
IV. FATIGUE:	
1. Overload	Use larger bearing, reduce load, use stronger bearing material.
2. Edge loading	Correct shaft deflections, correct bearing alignment, check machining of bearing and journal.
3. Local high stress concentration	Check for corrosion factors, electrical pits, rough surface finish, use higher-viscosity oil
4. High bearing temperature	Reduce operating temperature, use heavier oil, change bearing design for higher oil flow, change to higher-temperature bearing material.
5. Rotor vibration	Balance rotor, use stabilizing bearing design of elliptical, pressure, or three-lobe type. Change rotor support to damp out vibration.
6. Cavitation erosion	Relocate oil holes and grooving, use higher viscosity oil and lower bearing temperature, use fatigue-resistant bearing material.
V. SEIZURE:	
1. Too little clearance	Provide adequate clearance, taking into account any temperature effects and thermal expansion.
2. Insufficient lubrication	Supply larger amounts of oil, use higher viscosity oil; use extreme-pressure lubricant; use proper design of oil-feed grooves and orifices
3. Too high load	Increase bearing area and modify bearing design to carry load; use higher-viscosity oil, use compounded oil.
4. Overheating	Provide external cooling and ample oil flow.
5. Unsuitable bearing material	Use more compatible bearing material with satisfactory strength and thermal properties.

Table 9. Rolling-Bearing Trouble-Shooting Chart

CAUSE	REMEDY
1. OVERHEATED BEARING:	
a. Inadequate lubrication	Clean oil holes, filters, and vents. Use a fresh lubricant.
b. Excessive lubricant churning	Use lower-viscosity oil, lower oil level to center of bottom ball or roller, use oil mist
c. Inadequate internal clearance	Use bearing of greater looseness, allow for differential thermal expansion, reduce interference of shaft and housing fits, correct any housing out of roundness or warping
d. High seal friction	Use reduced spring tension with seals, lubricate seals, switch from rubbing seal to low-clearance shield
e. Excessive preloading	Use gaskets or shims to relieve axial preload with opposed pair or with two held bearings on a shaft subjected to thermal expansion. Change design to use only one held bearing
f. Spinning outer ring	Use closer housing fit, use steel insert in soft aluminum housing, use garter spring or rubber holding ring
g. Misalignment	Correct alignment by shimming pillow blocks, housings, or machines to get shafts and bearings in line. Check for misalignment of bearing seats and shaft and housing shoulders
2. NOISY BEARING, VIBRATION:	
a. Wrong type oil	Check recommendation of manufacturer
b. Insufficient lubrication	(See 1a)
c. Defective bearing	Check for brinelling, fatigue, wear, groove wobble, poor cage. Replace bearing
d. Dirt	Clean bearing housing, replace worn seals, improve seal arrangement, eliminate source of dirt
e. Corrosion	Improve sealing to keep out corrosive elements, use corrosion-resisting lubricant
f. Too great internal clearance	Change to bearing with smaller clearance
g. Unbalance	Balance rotor
h. Misalignment	Align (see 1g)
i. Too loose shaft or housing fit	Build up shaft or bore with chrome plate or metallize and regrind
j. Improper mounting	Correct dirty or off-square shaft and housing shoulders and seats. Avoid brinelling caused by pounding on bearing
k. False brinelling	Use vibration mounts for machine to isolate from platform during idle periods
l. Seal rub	Check for metal bearing seal or shield rubbing on shaft, shaft shoulder, or housing
3. LOSS OF LUBRICANT:	
a. Oil leakage through seal	Adjust oil level to center of lowest ball or roller, replace seal, use double-seal arrangement with drain between, eliminate any unfavorable air flow by proper baffles and balancing channels
b. Leakage at housing split	Use thin layer of gasket cement, replace housing
c. Dry, caked residue	Use silicone or other high-temperature grease use oxidation-inhibited or synthetic oil or grease, cool oil in external cooler, cool bearing housing, increase oil flow to promote cooling
4. LOOSE BEARING:	
a. Shaft diameter too small	Turn down shaft, chrome-plate or metallize and regrind to give proper fit. Retighten adapter to get firm grip on shaft
b. Housing bore too large	Build up bore with chrome plate or metallize and regrind, bore out housing and press in sleeve to give proper bearing fit (a slip fit on OD is usually desirable to allow for differential axial thermal expansion of a shaft between two bearings)
5. HARD TURNING OF SHAFT:	
a. Excessive bearing preload	Use less interference fit on shaft or in housing, select bearing with greater internal clearance where heat conduction expands shaft and inner bearing ring. Relieve axial preloading by housing shims with either two opposed bearings or two "held" bearings on one shaft
b. Bearing pinching or cocking	Scrape housing bore to relieve pinching. Replace or remachine warped housings, check bearing seats as source of cocking

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

Troubleshooting

A step by step guide often proves useful in troubleshooting. The following group of tables lists the major problems encountered in gear drives, their causes and some corrective actions:

Table 7 Visual Inspection of Gears
Table 8 Journal Bearing Troubleshooting Chart [7]
Table 9 Rolling Bearing Troubleshooting Chart [7]
Table 10 Gearbox Troubleshooting Guide

Overhaul

Generally, turbomachinery gearboxes do not have a specific time period where the unit is dis-

Table 10. Gear Box Trouble-Shooting Chart

PROBLEM	RECOMMENDED INSPECTION	CORRECTIVE ACTION	PROBLEM	RECOMMENDED INSPECTION	CORRECTIVE ACTION
Overheating	1. Oil cooler operation	Check flow of coolant and oil flow. Measure oil temperature into and out of cooler. Check cooler internally for build up of deposits from coolant water.	Shaft Failure (continued)	4. High transient loading	Apply couplings capable of absorbing shocks. Use couplings with shear pins.
	2. Is oil level too low or too high	Check oil level indicator.		5. Torsional or lateral vibrations	Adjust system mass elastic characteristics to control critical speed location. Possibly coupling geometry can be modified.
	3. Bearing installation	Make sure bearings are not pinched and properly adjusted.		6. Cracks due to fretting corrosions	Note cause of fretting and correct. Press fits between gear and shaft.
	4. Grade and condition of oil	Check that oil is specified grade. Inspect oil to see if it is oxidized, dirty or with high sludge content.	Oil Leakage	1. Exceed oil level	Check oil level indicator.
	5. Lubrication System	Check operation of oil pump. Make sure suction side is not sucking air. Measure flow. Check if oil passages are free. Inspect oil line pressure regulator, nozzles and filters to be sure they are free of obstruction.		2. Is breather open	Check oil breather.
	6. Coupling float and alignment.	Check coupling alignment and adjust end float.		3. Are oil drains open	Check that all oil drain locations are free and clean.
Shaft Failure	1. Type of coupling	Rigid couplings between rigidly supported shafts can cause shaft failure. Replace with coupling to provide required flexibility and lateral float.		4. Oil seals	Check oil seals and replace if worn. Check condition of shaft under seal and polish if necessary.
	2. Coupling Alignment	Realign equipment as required.		5. Plugs at drains, levels, and pipe fittings	Apply sealant and tighten fittings.
	3. Excessive Overhung Load	Reduce overhung load. Use outboard bearing or replace with higher capacity unit.		6. Housings and caps	Tighten cap screws or bolts. If not effective, remove housing cover and caps. Clean mating surfaces and apply new sealing compound. Reassemble. Check compression joints by tightening fasteners firmly.
			Unusual or increasing noise	1. Check gears and bearings	See Tables 7, 8 & 9

assembled and overhauled. It is more common to note deterioration of a bearing or gear and replace the particular component at a convenient time. Usually the gearbox is supplied with an operating and maintenance manual which describes how to assemble and disassemble the unit. If the user is not completely familiar with the equipment, it would be prudent to have a factory representative accomplish any major component replacements.

Spare Parts

Spare gears or bearings for a gear unit will not necessarily be readily available, particularly if the gearbox design was customized in some manner. When purchasing the drive the user should request a recommended spares list and determine what the availability of those parts will be. The user and manufacturer can then arrive at some agreement over what spares will be available and where they will be stored.

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BIOGRAPHY

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At American Lohmann, Mr. Lynwander is responsible for the analysis, design and development of enclosed gear drives. Product lines of parallel shaft and planetary high speed transmissions are being introduced under his direction.

From 1959 to 1978, he was with AVCO-Lycoming Division, a gas turbine manufacturer, where he headed up the Mechanical Components Group. His responsibilities included turbine engine gears, bearings, seals, clutches and lubrication systems. He established computerized procedures for the analysis and design of gear teeth and participated in numerous turbine engine power and accessory gearbox programs.

Mr. Lynwander has a BSME (1959) and MSME (1964) from the University of Bridgeport and is a registered Professional Engineer. He is the Chairman of the Tribology Division of the Aerospace Gearing Committee of the American Gear Manufacturers Association. He has authored numerous publications concerning gearing, seals, and clutches.