

HIGH SPEED GEARS — DESIGN AND APPLICATION

by

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ABSTRACT

High speed gear drives can be reliable if all the factors influencing their design, application, and operation are considered during the design stages. This paper will discuss the parameters used by gear engineers and the influence of design variations. Design will be analyzed from gear tooth geometry to installation considerations. Manufacturing processes, lubrication, allowable vibration levels, along with a simple gear comparison method, will be discussed.

INTRODUCTION

High speed gearing has in many instances developed a bad name with users due to problems encountered. It must be admitted that gear manufacturers have made mistakes, and there will continue to be mistakes in the future.

In all fairness to gear manufacturers, we should approach the solution to gear problems as a team realizing that gears occupy a unique position in a drive train. Some of these problems are caused by the following factors:

1. The gear supplier has less information about the package than any participant.
2. The system is not compatible due to spring rates and masses. In other words, the system is not tuned.
3. The gear is usually the only item required to operate with metal parts in such intimate contact as the gear teeth. Any disturbance produces a varying stress in the teeth, which results in early failure.
4. No other part of the system requires the operating parts to carry a stress varying from zero to maximum at 5,000 to 40,000 cycles per minute. At 10,000 RPM, a gear tooth receives approximately 10^7 cycles in 16.7 hours of operation. The old textbooks assumed that 10^7 cycles was infinite life, but this is not necessarily the case.

Gear drives are a series of compromises in the design and application, and the problem should be recognized by the OEM, engineering and construction contractor, and the user.

GEAR DESIGN

The gear engineer has many decisions to make when designing a gear drive to accomplish a specific task. Some of these considerations will be briefly explained.

Unless otherwise stated, the following discussion will concern either single or double helical parallel shaft gearing since other gear types are seldom used for high speed applications.

Appendix A outlines basic gear terminology and definition of terms.

Pressure Angle

The first decision concerns the tooling (determining pressure angle and tooth proportions) available for generating the gear teeth. A designer has considerable latitude in this area since tool suppliers will produce special cutting tools as required.

Pressure angle for the gear teeth is probably the first selection to be made. Figure 1 shows a transverse section of a set of gears in mesh which can be spur, single helical, or double helical. From this drawing you can visualize the effect of the pressure angle on gear design. It should be pointed out that the pressure angle shown in Figure 1 is in the transverse plane, and the transverse pressure angle increases with helix angle if normal pressure angle is held constant.

Good gear design dictates that the normal pressure angle be between 14.5° and 25° . Figure 2 shows the change in contact ratio with change in pressure angle. This is an indication of the number of teeth in contact in the transverse plane. Figure 3 shows the variation of length of line of action with pressure angle. Notice that as the pressure angle increases the contact ratio and the length of line of action decreases.

One of the most important considerations when selecting a pressure angle is tooth strength. The variation of the gear tooth geometry factor is shown in Figure 4: the larger the pressure angle, the higher the tooth strength.

As a general rule, the higher the contact ratio, the less noise the gears will generate assuming equal accuracy. However, tooth strength must be considered and, as a result, the most common normal pressure angles in use today vary between 17.5° and 22.5° .

Higher pressure angles increase the bearing loading as shown in Figure 5, but this is not usually a determining factor when selecting a pressure angle.

Figures 2 through 4 assumed a tooth working depth of 2.0/diametral pitch (generally referred to as full depth), but other depths can be used, such as stub (depth = 1.6/diametral pitch) or extended (depth = 2.3/diametral pitch). The stub and extended forms have advantages and disadvantages; for the sake of brevity, they will not be discussed.

Helix Angle

Helix angles vary from approximately 5° to 45° . Single helical gearing is usually in the 5° to 20° range, and double helical gearing in the 20° to 45° range. Generally, the helix

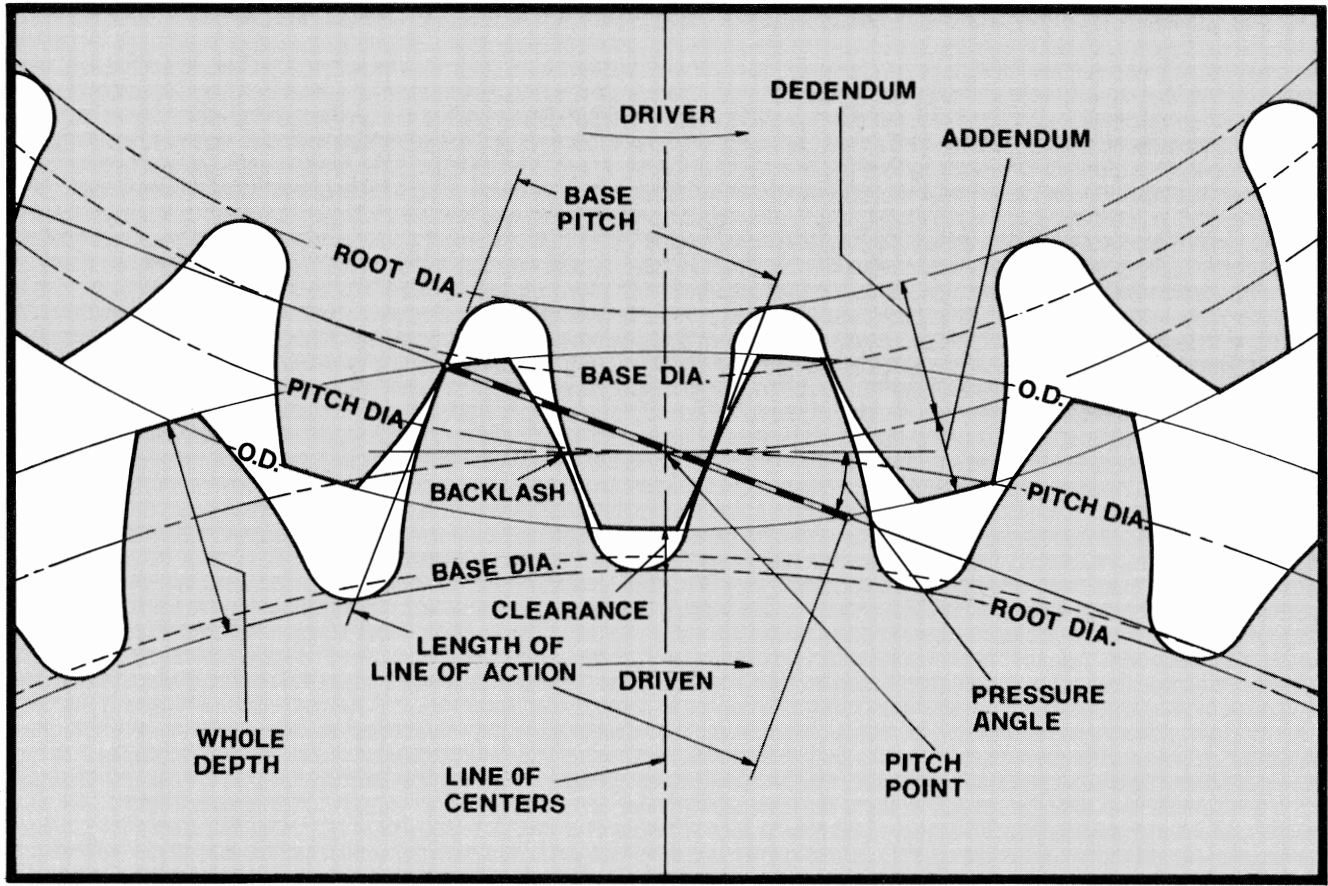


Figure 1. Transverse Section Through a Gear and Pinion in Mesh.

angle is selected to obtain a minimum face overlap ratio of 2.0 (number of teeth in contact in the axial direction) for good load sharing.

Figure 6 shows how face overlap ratio increases with helix angle and the more teeth in contact, the less noise the gears generate since the errors tend to average out.

Referring to Figure 5, generated thrust is plotted as a percentage of transmitted tangential load on the gear teeth. This is the primary reason for lower helix angles in single heli-

cal gearing. Lower helix angles reduce thrust bearing size and the overturning moment.

Single or Double Helical?

This question cannot be answered in a few words. Some of the advantages and disadvantages of each type are as follows:

1. Advantages of Single Helical
 - a. Can probably be made more accurate, one gear instead of two gears on the same blank.

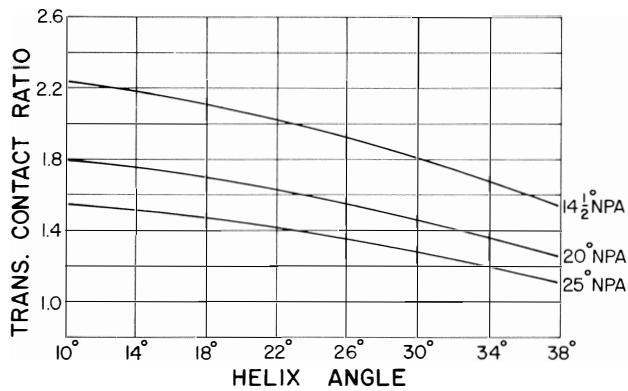


Figure 2. Showing Variation of Transverse Contact Ratio with Pressure Angle and Helix Angle.

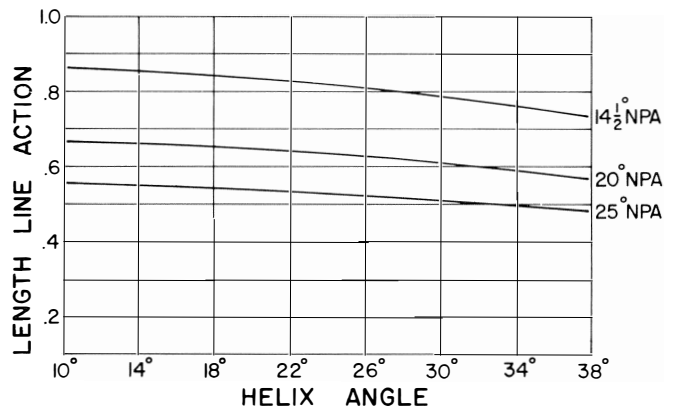


Figure 3. Length of Line of Action Change with Pressure Angle and Helix Angle.

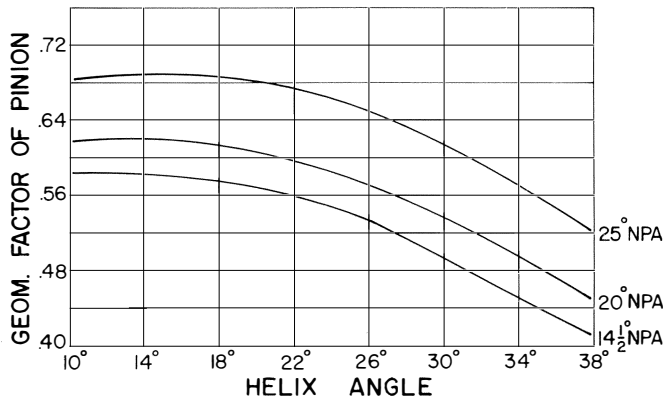


Figure 4. Tooth Strength (J Factor or Geometry Factor) Change with Pressure Angle and Helix Angle.

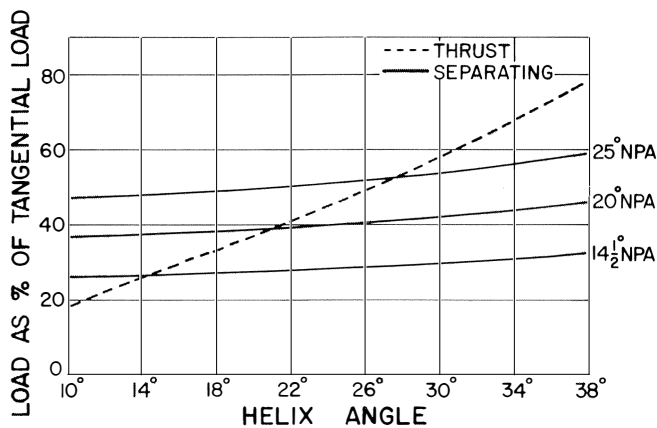


Figure 5. Thrust and Separating Load as a Percentage of Transmitted Tangential Load.

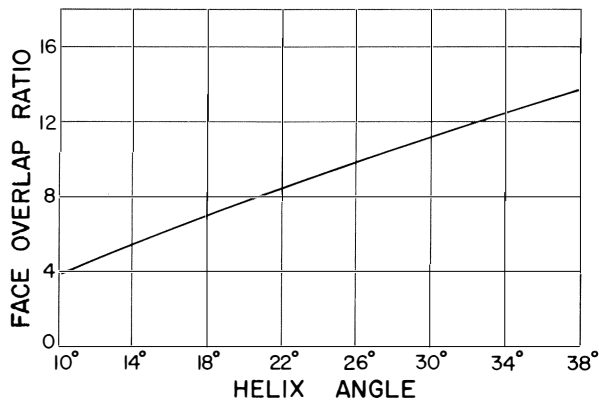


Figure 6. Face Overlap (Contact) Ratio Variation with Helix Angle.

- b. Not as sensitive to coupling thrust.
- c. Do not have a measurable apex runout.
- d. Gear tooth-wise the single helical is easier to design from the standpoint of modification of helix angle for deflections.
- e. Less expensive to cut teeth.

- f. Teeth can be finish ground more easily.
2. Disadvantages of Single Helical
 - a. Requires expensive thrust bearings and thrust faces.
 - b. Less efficient due to heat load of thrust bearings.
 - c. The active face width cannot be as great as a double helical due to possible alignment and deflection problems. Assuming the same face width and line up errors, the single helical will produce approximately twice the tooth deflection and stress as the double helical. Assuming equal hardness, the single helical will have a higher PLV than the double helical at the same speed and ratio when L/D ratio enters the design.
 - d. Torsional vibration can produce an axial vibration which is very hard to isolate.
 - e. The thrust loads produce an overturning moment on the gear and pinion which must be accounted for when making light load contact checks.
 3. Advantages of Double Helical
 - a. Very simple to design and manufacture due to absence of thrust faces and thrust bearings.
 - b. Very little thrust produced by gearing.
 - c. More efficient than single helical due to thrust bearing losses.
 - d. Allows more face width to pinion diameter as discussed above.
 - e. More service experience.
 - f. Coupling lockup is not as likely due to slight axial movement of pinion. Coupling teeth tend to break free.
 4. Disadvantages of Double Helical
 - a. All double helical gearing has some apex runout which produces additional loading.
 - b. Sensitive to coupling lockup (thrust).
 - c. Hard to modify the tooth longitudinally, but not required as often as for single helical.
 - d. Slightly more expensive to cut teeth due to set up and tool change.
 - e. Not as adaptable to finish grinding due to wide gap required for wheel runout.

Tooth Hardness

Gears are available today ranging in hardness from 225 BHN to 60 Rc and each have advantages. Some considerations that must be made when determining hardness are as follows:

1. Gears in the medium hard range (225 BHN to 350 BHN) are generally more forgiving of operational errors and will wear slightly before failing.
2. Heat treatment on medium hard gears is simple compared to surface hardened gearing.
3. Surface hardened gearing has lower pitch line velocities due to the higher allowable unit loading.
4. Very high horsepower drives in many cases require the very hard gears to transmit the power and stay within allowable pitch line speeds.
5. Very hard gears have the disadvantage of being susceptible to scoring due to high load intensity and sliding velocities.

6. Medium hard gears tend to give more warning when gear failure is approaching by an increase in noise level.
7. When weight and space are at a premium, the hard gearing may have to be used.
8. Tooth deflections under load are higher on hard gears and as a result, may be noisier. Tooth deflections can be accounted for in the design and manufacturing process but the higher the deflections, the greater the problem with tooth form modifications.
9. When case hardened gearing is installed as the original equipment selection and a decision is made later to increase the transmitted power, usually the equipment must be moved to increase the gear size. With medium hard gearing, this can usually be accomplished in the available space by installing harder gearing.
10. Case hardened gearing when carburized must have a finish grinding operation. When nitriding is used, finish lapping or honing can be used.
11. The "quickie" repair shops cannot usually repair the case hardened gears on an emergency basis since they do not have grinding equipment.

Scoring

Scoring must be evaluated when a gear set is operating at very high speeds or high load intensity. Probability of scoring can be predicted using the flash temperature index. This calculation indicates the probability of scoring using a flash temperature index calculated by the procedures outlined in AGMA 217.01. Index values below 275 are considered low scoring risks, values between 275 and 335 are medium risk, and higher values are considered high risks.

As scoring probability increases, copper or silver plating on the gear teeth is sometimes used to prevent failure during start-up. This plating acts as an extreme pressure lubricant to separate the tooth surface asperities until the teeth break-in.

In Figure 7 the effect of speed and load intensity is shown. These curves are general in nature and cannot be used to evaluate a gear set since tooth size, proportions, lubrication, and actual pressure angle must be considered. It should be pointed out that AGMA 217.01 is empirical and was derived from tests on a large number of gear sets using synthetic oils; however, it can be used as a good comparison.

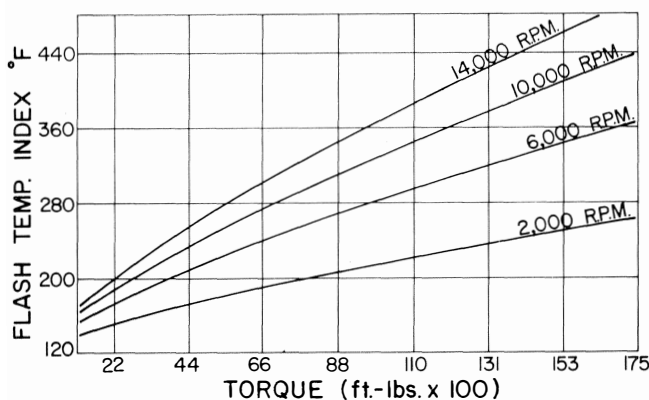


Figure 7. Scoring Based on Flash Temperature Index Related to Speed and Torque.

Bearings

Bearings of all types can be used to support gear rotors. The types used are generally:

1. Plain journal
2. Pressure dam type plain journal
3. Tilting pad journal (TPJ)
4. Elliptical

The stability types are usually applied to the pinions for light load stability. It should be pointed out that gear rotors have imposed operating loads and do not require the same degree of no-load bearing stability as a compressor or pump which has only the rotor weight applied to the bearings.

Thrust bearings vary from the ball bearing to self-equalizing tilting pad type, the most common being the babbitt lined flat face thrust bearing. The flat face bearing is sometimes modified to add tapered lands which doubles the load carrying capacity. The tilting pad bearing is becoming more popular due to its high thrust capacity and misalignment capabilities. Also, the tilting pad type thrust bearing is more efficient due to the higher allowable loading and lower rubbing speeds.

Gear Housings

Gear housings can be made from cast iron, cast steel, welded steel plate and, in some cases, aluminum. The gear housing must be stress relieved before final machining to assure dimensional stability.

Housings must be rigid enough to maintain alignment under all operating conditions but cannot be designed to resist foundation bolt forces if the mounting surfaces are not accurate.

Care must be taken to assure sufficient clearance around the gears to prevent oil choking. Also, the design must assure uniform case temperatures (as near as possible) to prevent thermal distortion.

Materials for Rotating Elements

Materials must be carefully selected for the application and suitable for the heat treating processes to be used. In other words, case hardened gear material selection is different from the through hardened materials. The 1977 edition of API 613 has a list of current materials in common use.

Dimensional stability is the most important material factor to consider after strength properties are determined. Stress relieving after rough turning and after straightening is a necessity since stress relieving (and thus distortion) can occur during operation due to dynamic loading and the operating temperature.

Gear Accuracy

This term is often discussed and AGMA has published a standard on gear accuracy. AGMA 390 has tabulated maximum allowable errors based on gear size for the different elements of gear teeth. However, this standard is to be used only for loose gearing when purchased in un-matched sets. Some of the values, such as allowable lead error (helix angle error), will lead to early failure when used for high speed wide face width gears.

As a result, there are no accuracy standards for high speed enclosed gear drives. As a general rule, manufacturers of these drives work on allowable mis-match. This is controlled by monitoring the gears and pinions for involute, lead, runout, spacing, and surface finish. To prove the accuracy, a light load blue transfer check is performed.

There is no agreement in the gear industry as to what levels are actually required. It has never been determined whether the lowest accuracy gearing shipped, which performed perfectly, or the highest accuracy shipped, which gave trouble, should be taken as a starting point for an accuracy standard.

MANUFACTURING PROCESSES

There are several methods of manufacturing good high speed gearing. The most common are hobbing; hobbing and shaving; hobbing and lapping; and grinding (usually cut by hobbing or shaping before finish grinding).

As a general rule, high speed gearing has a finishing operation after cutting such as shaving, lapping or grinding. The shaved or lapped gears have been more widely applied than grinding, but grinding is gaining in popularity. Some of the advantages and disadvantages of each are as follows:

1. *Hobbing*

Hobbing produces good tooth spacing and accurate lead. It cannot economically achieve a surface finish better than 40 microinches. The hobbing machine generates gear teeth by a continuous indexing process in which both the cutting tool and the workpiece rotate in a constant relationship while the hob is fed into the work.

The hob (cutting tool) is basically a worm which has been fluted and has form-relieved teeth. These flutes provide the cutting edges and can be sharpened and retain the original tooth profile. As the workpiece meshes with the hob, the teeth are formed by a series of cuts which is the generating process. To cut the helix angle, the rotation of the work is slightly retarded or advanced in relationship to the hob rotation and the feed is held in a definite relationship with the work and hob.

Hobbing machines can produce very accurate gears and are available to hob gears from 0.5 to 300 diametral pitch with pitch diameters over 200 inches.

2. *Hobbing and Shaving*

The shaving process improves surface finish, involute profile, lead, and can be used to crown the teeth. It will not improve spacing or pitch line runout. Also, shaving with inaccurate cutters will reduce as-hobbed accuracy. The shaving cutter has involute teeth and meshes with the part being shaved.

3. *Hobbing and Lapping*

The lapping process improves surface finish, involute profile, lead, and pitch line runout. The absolute accuracy is not as good on lapped gears, but the mismatch error is generally as good as other methods of production. Lapping is performed with the gears on an accurate mounting stand running at zero backlash with a cutting compound mixed with oil or grease performing the finishing operation.

Finish lapping high speed gears requires that the gear teeth be carefully hobbled to 40 to 50 microinches surface finish with good lead, profile, and tooth spacing. This will require lapping for surface finish and profile improvement only. When required, a surface finish of 8 to 15 microinches can be obtained, but, as a general rule, 20 to 30 microinches is acceptable.

4. *Grinding*

Grinding produces the best absolute values of lead and involute profile. As a general rule, tooth spacing is not

as good as gearing produced on a precision hobbing machine due to the smaller index wheel and the single space indexing procedure. The disadvantage of the grinding process is the skill and patience required of the operator. To produce gears to the same accuracy levels as shaving and lapping requires many trips from grinder to checking machine. Gears too large to be checked depend on a roll-in blue check for verification of lead and involute. Also, the grinding machines in current use have a reciprocating motion of the grinding head and require a higher level of maintenance to produce good gears.

The above is very brief and covers only the high points. A good gear set can be made by any of the above methods, and, easier yet, a poor set can be made. The secret of success in the manufacture of high speed gearing is good design, and most of all good workmanship.

LUBRICATION

The oil furnished to high speed gears has a dual purpose: Lubrication of the teeth and bearings, and cooling. Usually, only 10% to 30% of the oil is for lubrication and 70% to 90% is for cooling.

A turbine type oil with rust and oxidation inhibitors is preferred. This oil must be kept clean (filtered to 40 microns maximum, or preferably 25 microns), cooled, and with the correct viscosity. Synthetic oils should not be used without the manufacturer's approval.

For some reason, the high speed gear makes all the compromises when oil viscosity for the system is determined. Usually a viscosity preferred for compressor seals or bearings is selected and gear life is probably reduced. The bearings in a gear unit can use the lightest oils available, but gear teeth would like a much heavier oil to increase the film thickness between the teeth.

When selecting a high speed gear unit, the possibility of using an AGMA No. 2 Oil (315 SSU @ 100° F.) should be considered. In most cases, the sleeve bearings in the system can use this oil and, if not, a compromise 200 SSU at 100° F. oil should be considered.

When 150 SSU at 100° F. oil is necessary, inlet temperatures should be limited to 110° F. to 120° F. to maintain an acceptable viscosity. Oil should be supplied in the temperature and pressure range specified by the manufacturer.

Up to a pitch line speed of approximately 15,000 feet per minute, the oil should be sprayed into the out-mesh. This allows maximum cooling time for the gear blanks and applies the oil at the highest temperature area of the gears. Also, a negative pressure is formed when the teeth come out of mesh pulling the oil into the tooth spaces.

Above approximately 15,000 feet per minute, 90% of the oil should be sprayed into the out-mesh and 10% into the in-mesh. This is a safety precaution to assure the amount of oil required for lubrication is available at the mesh.

In addition to the above, in the speed ranges from 25,000 to 40,000 feet per minute, oil should be sprayed on the sides and gap area (on double helical) of the gears to minimize thermal distortion.

GEAR RATING

Determining the gear size required to perform a certain task is sometimes difficult. API, working with the gear man-

ufacturers, has published a simplified rating formula in the 1977 edition of API 613. The K factor (tooth pitting index) shown below is a universal method of determining and comparing gear size and has been used for many years.

Durability (Pitting) Rating

The following is included to show the similarity between the API method and the AGMA.

The fundamental AGMA pitting formula for gear teeth is as follows:

$$S_c = C_p \sqrt{\frac{W_t C_o C_s C_m C_f}{C_v d F I}} \quad (1)$$

where:

S_c = Contact stress number

$$W_t = \frac{P_{sc} \times 126,000}{n_p \times d} = \text{Tangential Load} \quad (2)$$

P_{sc} = Transmitted horsepower

n_p = Pinion RPM

C_p = Elastic coefficient

C_o = Overload factor

C_v = Dynamic factor

d = Pinion pitch diameter, inches

F = Net face width of the narrowest of the mating gears or the sum of the face widths of each helix of double helical, inches.

C_m = Load distribution factor

C_s = Size factor

I = Geometry factor

C_f = Surface condition factor

$$I = \left(\frac{.95Z}{P_N} \right) \left(\frac{\cos \phi_t \sin \phi_t}{2} \right) \left(\frac{m_G}{m_G + 1} \right) \quad (3)$$

Z = Length of line of action

P_N = Normal base pitch, inches

ϕ = Operating transverse pressure angle

m_G = Gear ratio

$$\text{and } S_c = S_{ac} \left(\frac{C_L C_H}{C_T C_R} \right) \quad (4)$$

where:

S_{ac} = Allowable contact stress number

C_L = Life factor

C_H = Hardness ratio factor

C_T = Temperature factor

C_R = Factor of safety

By assigning values of unity to C_o , C_s , C_f and substituting, the following equation results:

$$C_p \sqrt{\left(\frac{126,000 \times P_{sc}}{n_p \times d^2 \times F} \right) \left(\frac{m_G + 1}{m_G} \right) \left(\frac{C_m}{C_v} \right) \left(\frac{2 P_N}{.95Z \cos \phi_t \sin \phi_t} \right)} \quad (5)$$

The tooth pitting index "K" is defined as:

$$K = \left(\frac{126,000 \times P_{sc}}{n_p \times d^2 \times F} \right) \left(\frac{m_G + 1}{m_G} \right) = \text{tooth pitting index} \quad (6)$$

The K factor, which is a tooth pitting index, can be easily seen in formula 5 and the K factor is proportional to the square of the contact stress.

By assigning conservative values to the factors C_p , C_m , C_v , P_N , Z , ϕ_t , S_c , C_L , C_T , and C_R , maximum allowable K factors can be determined for simple gear selection and comparison.

API 613, 1977 edition, has assigned material index factors based on hardness and certain geometry limitations. These are further modified using a service factor to account for driving and driven equipment characteristics. So that:

$$\text{Allowable K} = \frac{\text{Material Index Number}}{\text{Service Factor}} \quad (7)$$

The above formulas are included only to show that the new API method is consistent with AGMA 211.

Strength Rating

This new standard also uses a modified version of the AGMA 221 bending stress formula which is as follows:

$$S_t = \frac{W_t \times P_n}{F} \times (\text{SF}) \times \frac{1.8 \cos \psi}{J} \quad (8)$$

where:

S_t = Bending stress number

ψ = Helix angle

J = Geometry factor (AGMA 226)

SF = Service factor

P_n = Normal diametral pitch

The above formula has been simplified in a similar manner to the contact stress formula. Using these methods, anyone with a hand calculator and the basic gear geometry can check gear rating. The only problem would be in determining the geometry factor J, but a good estimate of this can be found in AGMA 226. If this is not available, the gear supplier will generally be happy to furnish the correct value.

It should be pointed out that gears selected by the above methods are generally much more conservative than those sized by standard AGMA methods using a service factor of 1.5 to 2.0.

INSTALLATION

Gear unit installation is one of the most important factors to be considered for long trouble free operation. No matter how accurately the gear unit is manufactured, this can be destroyed in a few hours of operation when improperly installed.

1. Shaft Alignment

This is a subject too complex to discuss in detail in this paper, but it is very important for long gear life. Poor alignment can cause unequal distribution of tooth loads and distortion of the gear elements due to overhung moments. Also, it should be pointed out that a 2.0 mil shaft vibration level on the gear unit produced by misalignment is equivalent to a gear pitch line runout of 2.0 mils.

2. Housing Installation

The gear housing must be properly supported to maintain proper internal gear alignment. When a gear unit is installed, the support pads must be maintained in the same plane as used by the manufacturer during assembly when

gear face contact was obtained in the plant. Before start-up, gear face contact should be checked using high spot blueing and rotating through mesh.

At start-up, the teeth should be coated with layout blue and operated for approximately 30 minutes to an hour under light load. Inspection of this blued area should show approximately 90% face contact. If this contact is not obtained, the gear housing can be shimmed under the proper corner until acceptable face contact is achieved.

It should be pointed out that many large high hardness or wide face width gears are manufactured with helix angle modifications to account for torsional and bending deflection. When the helix angle has been modified, good face contact will not be obtained under light load. In this case, the gear supplier should furnish data on percent of face contact versus load to be used as a guide during installation and start-up. Also, many gears have a short area of ease off on each end of the teeth to prevent end loading, and this area usually will not show contact under light load.

The larger the gear unit, the more important this check becomes since larger housings tend to be more flexible. Also, the use of base plates furnished by the OEM does not eliminate face contact problems, and these inspection procedures should be carried out.

THE SYSTEM INFLUENCE

1. *Vibration*

The gear is a part of the rotating system and all aspects of critical speeds (torsional, lateral, and axial), must be accounted for. The gear usually fails first due to these disturbances since the gear and pinion teeth operate within a few hundred microinches of each other.

2. *Couplings*

The coupling may be a minor part of the system cost-wise, but it is very important. Couplings are a constant cause of unbalance vibration and critical speed response due to spacer shift and wear.

Coupling lockup is a problem which has not been solved. On the test stand and in field tests, it has been shown that coupling lockup can cause severe housing vibration, while shaft vibration levels remain low. For this reason, monitoring shaft vibration is not the total answer for instrumentation. Usually, coupling lockup can be monitored with a housing velocity pickup sensing axial vibration.

3. *Operating Conditions*

The gear must be sized to handle the maximum possible output of the prime mover. Good operators will obtain the maximum output from a system, as they should. Surge in centrifugal compressors causes severe overload and can lead to early failure if not controlled.

The gear vendor quite often is given only the design horsepower of the driven machine. Actual transmitted loads are much higher due to proximity of torsional or lateral critical speeds, prime mover overload factor, product variations, or overspeed.

GEAR NOISE

As we all know, industrial noise levels are going to be reduced in the future. This decree has been handed down.

Unfortunately, the gear is a noise source. Our company and other gear companies are working on solutions, but we have not found an economical answer.

Gear noise can be caused by, but not limited to, the following:

1. Tooth spacing error.
2. Involute error (mismatch).
3. Surface finish (not a great influence).
4. Lead error.
5. Wear on tooth flanks.
6. Pitting (not a great influence unless severe).
7. Gear, shaft, or housing resonance.
8. Tooth deflections.
9. Improper tip or root relief.
10. Pitch line runout.
11. Excessive backlash (only with load reversals).
12. Too little backlash.
13. Noise transmitted from driving or driven machine.
14. Load intensity on gearing.
15. Rolling element bearing.
16. Clutches and couplings.
17. Face overlap ratio.
18. Contact ratio.
19. Lube oil pump and piping.

Gear noise can be controlled to some extent by the following measures:

1. Very careful design and super quality manufacturing (most expensive). Contrary to some opinions, the perfect gear is useless for power transmission due to tooth deflections under load. The trick is to obtain a gear which has a perfect involute form under load. The harder the gear, the more deflection due to higher allowable loading; as a result, good mesh conditions are more difficult to obtain since the involute produced is distorted more.
2. Extra heavy cast iron or double wall housings used with reasonably accurate gearing. Also, detuning techniques on housings and gear blanks can be used based on calculated and experimental data.
3. Using an acoustical enclosure (least expensive). Almost any noise level can be attained if space is not a problem. Sound enclosures have a very definite disadvantage when maintenance is required.

It should be pointed out that the inability of operators or maintenance personnel to actually place their hand on the equipment or hear the noises emitted will allow total destruction instead of minor damage. No matter how sophisticated the monitoring equipment, in many cases, the sense of touch and hearing is still the best indication of a machine's condition.

CONCLUSIONS

From the foregoing it can be seen that the gear design and application is a series of compromises. The gear manufacturer does make mistakes but most are willing to correct the prob-

lems. A perfectly good design can be a disaster with poor workmanship, and a poor design can sometimes be salvaged by good workmanship. Also, a properly designed and manufactured gear unit can be destroyed by improper installation or system design. To sum up, large high speed gear units are partly a science but in many aspects an art.

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APPENDIX A

Gear Terminology

GEAR — Gears are machine elements that transmit motion by means of successively engaging teeth. Of the two gears that run together, the one with the larger number of teeth is called the "gear."

PINION — Of the two gears that run together, the one with the smaller number of teeth is called the "pinion."

SPUR GEARS — This type of gear has parallel axes and the teeth are straight and parallel to the axes.

SINGLE HELICAL GEARS — These gears have helical teeth and have teeth of only one hand on each gear. For power transmission the axes are usually parallel.

DOUBLE HELICAL GEARS — These gears have both right-hand and left-hand helical teeth and operate on parallel axes. They are also known as herringbone gears.

RIGHT-HAND HELICAL GEAR — On this gear the teeth twist clockwise as they recede from an observer looking along the axis.

LEFT-HAND HELICAL GEAR — On this gear the teeth twist counterclockwise as they recede from an observer looking along the axis.

PITCH CIRCLE — This is the imaginary circle that rolls without slipping with the pitch circle of a mating gear. It is referred to as pitch diameter.

ADDENDUM CIRCLE — This circle coincides with the tops of the teeth in a cross section. It is referred to as the outside diameter.

ROOT CIRCLE — This circle is tangent to the bottom of the tooth spaces in a cross section. It is referred to as the root diameter.

BASE CIRCLE — This is the circle from which the involute tooth profiles are derived. It is referred to as the base diameter.

TOOTH SURFACE — This is the side of a gear tooth sometimes called the "flank."

PITCH POINT — This is the point of tangency of two pitch circles and it is on the line of centers.

TRANSVERSE PLANE — This plane is the same as the plane of rotation in gears with parallel axes. It is perpendicular to the axes of the gears.

NORMAL PLANE — This plane is perpendicular to the tooth surface at a pitch point, and it is perpendicular to a plane which is tangent to the pitch circles and parallel with the axes.

LINE OF ACTION — This is the imaginary line in the transverse plane along which contact occurs during engagement of mating teeth. It is a line passing through the pitch point and tangent to the base circles.

LENGTH OF ACTION — This is the distance on the line of action through which the point of contact moves during engagement of mating teeth.

PITCH — This is the distance between similar, equally spaced tooth surfaces along a given line or curve. The use of the single word "pitch" is confusing and for this reason specific designations are preferred, like circular pitch, etc.

CIRCULAR PITCH — This is the distance along the pitch circle from a point on one tooth to the corresponding point on another tooth. It may be measured in the transverse or the normal plane.

BASE PITCH — This is the distance along the base circle from a point on one tooth to the corresponding point on

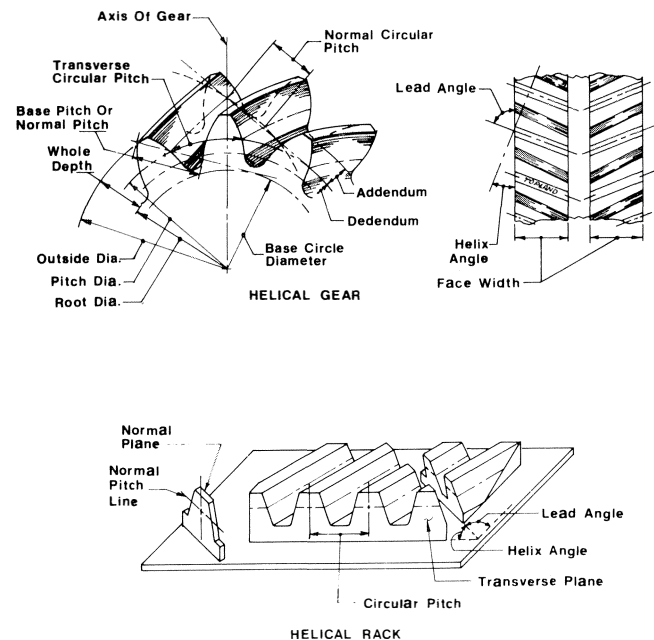


Figure 1A. Helical Gear and Rack Terminology.

another tooth. It may be measured in the transverse or the normal plane.

DIAMETRAL PITCH — This is the ratio of the number of teeth to the pitch diameter. It may be measured in the transverse or the normal plane.

ADDENDUM — The radial distance between the pitch circle and the addendum circle.

DEDENDUM — The radial distance from the pitch circle to the root circle.

CLEARANCE — The amount by which the dedendum of a given gear exceeds the addendum of its mating gear.

WORKING DEPTH — The depth of engagement of two gears, that is, the sum of their operating addendums.

CIRCULAR THICKNESS — The length of arc between the two sides of a gear tooth, on the pitch circle unless otherwise specified. It may be measured in the transverse or the normal plane.

CHORDAL THICKNESS — The straight line thickness of a tooth measured on any circle. It may be measured in the transverse or the normal plane.

CHORDAL ADDENDUM — The distance from the addendum circle to the point where chordal thickness is measured.

BACKLASH — The difference between tooth thickness and the space width in which it meshes on the operating pitch circles. The different backlash terms are normal backlash, transverse backlash, radial backlash (center distance change), and axial backlash.

FACE WIDTH — The length of the teeth in the axial plane.

PRESSURE ANGLE — An angle (at the pitch circle unless otherwise specified) between the line of pressure which is perpendicular to the tooth surface in some plane and a plane which is tangent to the pitch circles and parallel with the axes. It may be measured in the transverse or the normal plane.

HELIX ANGLE — The angle between the tooth and the gear axis as measured at the pitch circle.

BASE HELIX ANGLE — The angle between the tooth and the gear axis as measured at the base circle.

BOTTOM LAND — The surface at the bottom of a tooth space.

TOP LAND — The surface at the top of a tooth.

ANGLE OF ACTION — The angle through which a gear turns from the beginning to the end of contact between two mating teeth.

ANGLE OF APPROACH — The angle through which a gear turns from the beginning of contact between two mating teeth until their point of contact arrives at the pitch point.

ANGLE OF RECESS — The angle through which a gear turns when two mating teeth are in contact at their pitch point until contact ends.

TRANSVERSE CONTACT RATIO — The ratio of the length of line of action to the transverse base pitch.

FACE CONTACT RATIO — The ratio of the face width to the axial pitch. Sometimes referred to as "face overlap ratio."

TOTAL CONTACT RATIO — The sum of the transverse contact ratio and the face contact ratio.

TIP RELIEF — The modification of a tooth profile whereby a small amount of material is removed near the top of a gear tooth.

ROOT RELIEF — The modification of a tooth profile whereby a small amount of material is removed near the root of a gear tooth.

HELICAL GEAR FORMULAE STANDARD GEARING

$$1. \cos \psi = \frac{N + n}{2 P_n C}$$

$$2. d = \frac{n}{P_n \cos \psi} = \frac{2 C}{m_G + 1}$$

$$3. D = \frac{N}{P_n \cos \psi} = \frac{2 C m_G}{m_G + 1}$$

$$4. d_o = d + \frac{2 a_c}{P_n}$$

$$5. D_o = D + \frac{2 a_c}{P_n}$$

$$6. d_R = d - \frac{2 b_c}{P_n}$$

$$7. D_R = D - \frac{2 b_c}{P_n}$$

$$8. C = \frac{d + D}{2}$$

$$9. h_t = \frac{a_c + b_c}{P_n}$$

$$10. V_t = \frac{\pi d n_p}{12}$$

$$11. W_t = \frac{126000 P_{sc}}{n_p d}$$

$$12. W_R = \frac{W_t \tan \phi_n}{\cos \psi}$$

$$13. W_x = W_t \tan \psi$$

$$14. m_G = \frac{N}{n}$$

$$15. \tan \phi_t = \frac{\tan \phi_n}{\cos \psi}$$

$$16. P_t = P_n \cos \psi$$

$$17. P_N = \frac{\pi \cos \phi_n}{P_n}$$

$$18. \sin \psi_b = \sin \psi \cos \phi_n$$

$$19. d_b = \frac{n \cos \phi_n}{P_n \cos \psi_b}$$

$$20. D_b = \frac{N \cos \phi_n}{P_n \cos \psi_b}$$

$$21. Z = \frac{\sqrt{d_o^2 - d_b^2} + \sqrt{D_o^2 - D_b^2}}{2} - C \sin \phi_t$$

ψ = Helix angle

C = Center distance

P_n = Normal diametral pitch

P_t = Transverse diametral pitch

N = Number teeth, pinion

n = Number teeth, gear

a_c = Addendum constant of cutting tool

b_c = Dedendum constant of cutting tool

h_t = Whole depth of tooth

d = Pitch diameter, pinion

D = Pitch diameter, gear

d_o = Outside diameter, pinion

D_o = Outside diameter, gear

d_R = Root diameter, pinion

D_R = Root diameter, gear

n_p = Revolutions per minute, pinion

m_G = Gear ratio

V_t = Pitch line velocity (ft./min.)

W_t = Tangential load on tooth

W_r = Radial load on tooth (separating)

W_x = Axial load on tooth (thrust)

P_{sc} = Service or transmitted horsepower

ϕ_n = Normal pressure angle

ϕ_t = Transverse pressure angle

P_N = Normal base pitch

b = Base helix angle

d_b = Base circle diameter, pinion

D_b = Base circle diameter, gear