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Methodology of Designing High Speed Gears

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ABSTRACT: Designing of high speed gears needs more variables into account since the output speed is stepped up and the relative velocity is also high at output therefore it requires more parameters need to be considered. The parameters that gear design engineers use and the impact of design variations will be covered in this paper. The design will be examined in terms of gear tooth geometry, manufacturing procedures, lubrication, permissible vibration levels along with simple gear comparison method.

I. INTRODUCTION

High speed gearing design is difficult due to various problems being encountered. Gear manufacturers should approach the solution to gear problems realizing that gears occupy a unique position in a drive train.

Some of these problems are caused by the following factors:

1. The system is not compatible due to spring rates and masses. In other words, the system is not tuned.
2. Gears, shaft and Bearings are key rotating parts of Gearbox system. Disturbance in system may produce varying stresses which can result in premature failure in system.
3. The gear supplier has less information about the application.
4. Only 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. 10^7 cycles are considered to be infinite life, but this is not necessarily the case.

II. GEAR DESIGN

The gear design engineer has many decisions to make when designing a gear drive to accomplish High speed gears. Paper will discuss either single or double helical parallel shaft gearing since other gear types are seldom used for high speed applications.

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. Fig. 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 Fig. 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° to 25° . Pressure angle increases the contact ratio followed by decrease in length of line of action. One of the most important considerations when selecting a pressure angle is tooth strength. Larger the pressure angle results show higher tooth strength. Higher contact ratio results lesser noise in gears. Considering tooth strength as major criteria for designing gears, most common normal pressure angles in use today vary between 17.5° to 22.5° . Higher pressure angles increase the bearing loading, which can lead to selection of higher capacity bearings. Tooth working depth = $2/DP$ (generally referred to as full depth), Stub depth = $1.6/DP$

Extended depth = $2.3/DP$. The stub and extended forms have advantages and disadvantages; they will not be discussed.

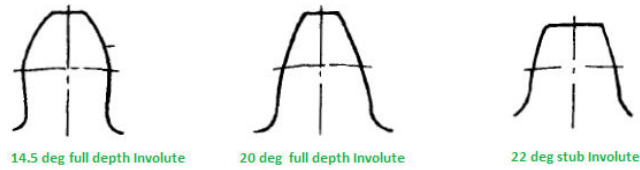


Fig – 1: Tooth Profile

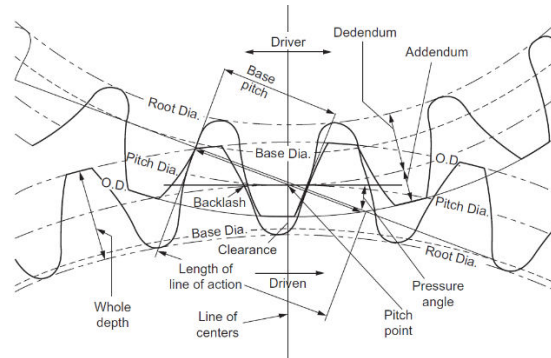


Fig – 2: Gear Mesh Approach & Recess

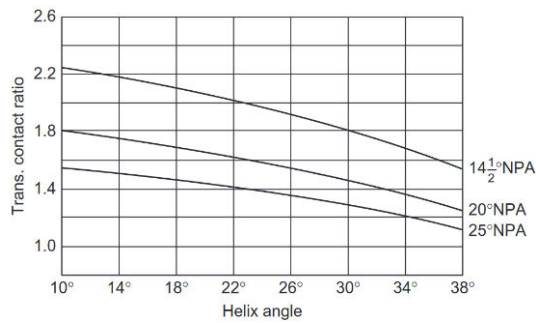


Fig – 3: Transverse Contact Ratio Vs Helix Angle

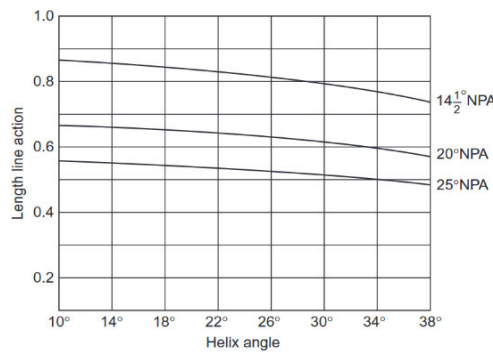


Fig – 4: Length line Action Vs Helix Angle

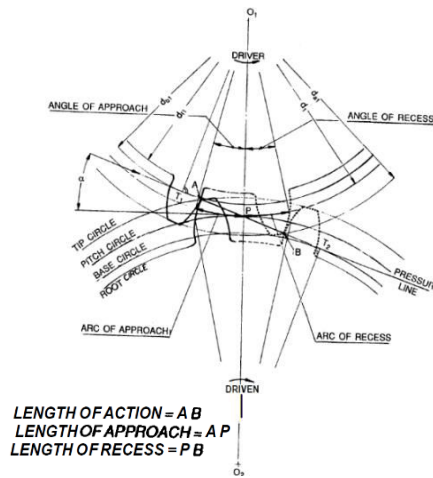


Fig – 5: Transverse section of Gear & Pinion in Mesh

Helix Angle

Helix angles vary from approximately 5°-45°. Single helical gearing is usually in the so to 5°-20° range, and double helical gearing in the 20°-45° range. Generally, the helix angle is selected to obtain a minimum face overlap ratio of 2.0 (number of teeth in contact in the axial direction) for good loadsharing. Face overlap ratio increases with helix angle and the more teeth in contact, which tends to generate lesser noise since errors tend to average out. Lower helix angles reduce thrust bearing size and the overturning moment. This is the primary reason for lower helix angles in single helical gearing.

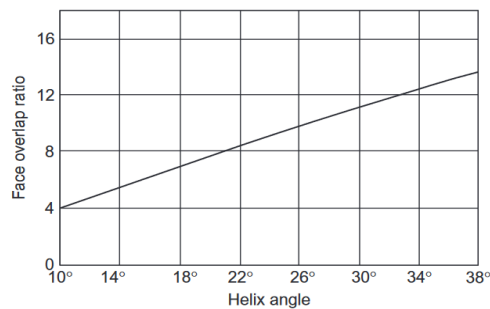


Fig – 6: Face overlap ratio Vs Helix Angle

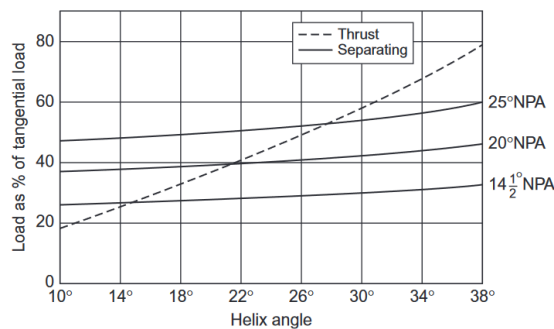


Fig – 6: Load as % of tangential load Vs Helix Angle

III. SINGLE / DOUBLE HELICAL

Advantages of Single Helical

- Can be manufactured more accurate
- Less expensive to cut teeth
- Gear tooth-wise the single helical is easier to design from the standpoint of modification of helix angle for deflections
- Teeth can be finished ground more easily

Disadvantages of Single Helical

- Requires expensive thrust bearings & faces
- Less efficient due to heat load of thrust bearings
- Single helical gears produce twice the amount of deflection & Pitch line velocity when compared to Double Helical.
- Torsional vibrations induce axial vibration which is difficult to be dampened

Advantages of Double Helical

- Simple to design due to absence of thrust faces and thrust bearings
- Little or no thrust force are produced
- More efficient than single helical as lesser heat loads produced when compared to single helical
- Higher L/D

Hardening method	Maximum L/d ratio	
	Double helical	Single helical
Through hardened	2.2	1.6
Case hardened	2.0	1.6

NOTE:
 L = face width including gap, mm;
 d = pinion operating pitch diameter, mm

Table- 1: L/d ratio for single & double helical gear

Disadvantages of Double Helical

- Due to presence of Apex point runout additional forces are subjected in gear
- Tooth Modifications are very hard to be implemented
- More expensive to cut teeth due to setup and tool change.

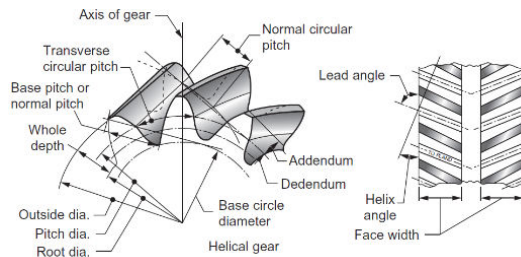


Fig – 7: Geometry of Single Helical & Double Helical



IV. TOOTH HARDNESS

- Surface hardened gears are generally suitable as it is less susceptible to manufacturing errors and will wear slightly before failing
- Surface hardened gears has lower pitch line velocity due to higher allowable unit loads
- Higher power in many drives require surface hardened gears to transmit higher power and stay within allowable pitch line speeds
- Medium hard gears are subjected to scoring, so preference is given to surface hardened gears
- Medium hardened gears tends to give warning when gear failure is approaching by increase in noise level
- Weight & space are premium, Medium hardened gears are preferred
- Tooth deflections when loaded in hardened gears result nosier, these must be addressed in design.
- Case hardened gearing when carburised must have grinding operation for finish. When nitriding is used finishing operation such as lapping, honing or no final machining is preferred.
- Generally, surface hardened gears are difficult to be repaired when compared to medium hardened gears due to elimination of grinding operations.

V. SCUFFING OR SCORING

AGMA 6011-I03 is concerned with two failure modes in gear teeth. They are surface pitting and root bending fatigue failure of the tooth material for a given number of stress cycles. There is another known failure type: scuffing (sometimes referred to as scoring). From the values of tooth loading, pitch line velocity, and viscosity of the lubricant, a condensed load function, F (load), is formed, which, to assure scuffing resistance, must be less than (or equal to) the geometric function, F (geometric). The geometric function is based on gear characteristics such as number of teeth of the pinion and gear, centre distance and gear set ratio. As long as the value of the load function, F (load), does not exceed that of the geometric function, F (geometric), there is adequate safety against scuffing.

$$F(\text{load}) \leq F(\text{geometric})$$

$$F(\text{load}) = \left[\frac{w'}{C_w} \right] [v']^{0.25} \left[\frac{46}{v_{40}} \right]^{0.22}$$

Sym	Description & Units
w'	Specific tooth load on pitch circle (N/mm)
v'	Pitch Line velocity (m/s)
v_{40}	Viscosity of Lubricant @ 40° C (mm ² /s)
C_w	1.10 (Conservative Value)
C_w	1.15 (Nominal value)
C_w	1.20 (Maximum value)

Table– 2: Load function

ISO viscosity grade VG	Nominal viscosity at 40°C, mm ² /sec (cSt)
VG - 22	22
VG - 32	32
VG - 46	46
VG - 68	68

Table– 3: Lubricant viscosities

Geometric Function, $F(\text{geometric})$

$$F(\text{geometric}) = \frac{(50 + z_1 + z_2)(a)^{0.5}}{A} [C_u]$$



z1	Number of teeth of the pinion
z2	Number of teeth of the gear
a	Centre distance (mm)
A	Refer table-4
C _u	Refer table-4

α_p	A	C _u @ 1 ≤ u < 3	C _u @ 3 ≤ u ≤ 10
15	350	95 + 28.6 (3 - u)	130 - 10 [(112.5 - (13 - u) ²) ^{0.5}
17.5	300	90 + 30 (3 - u)	120 - 10 [90 - (12 - u) ²) ^{0.5}
20	300	100 + 33.3 (3 - u)	130 - 10 [109 - (13 - u) ²) ^{0.5}
22.5	250	95 + 28.5 (3 - u)	130 - 10 [(112.5 - (13 - u) ²) ^{0.5}
25	250	105 + 31.4 (3 - u)	140 - 10 [(133.5 - (14 - u) ²) ^{0.5}

NOTE
 α_p is pressure angle, degrees;
 u is gear ratio (z_2/z_1).

Table– 4: Geometric function

VI. BEARINGS

Bearings of all types can be used to High speed.

1. Plain journal
2. Angular Contact or Spindle bearings
3. Tilting pad journal (TPJ)

The stability types are usually applied for light load stability. Thrust bearings vary from the ball to self-equalizing tilting pad type. the most common Babbitt lined flat face thrust bearing. The flat fact bearing is sometimes modified to add tapered lands which doubles the load carrying capacity. Tilting pad bearing is becoming more popular due to its high thrust capacity & misalignment capabilities. Also, the tilting pad type thrust bearing is more efficient due to the higher allowable loading & low rubbing speeds.

Type of bearing	Projected unit load, ³⁾ N/mm ²	Minimum lubricant film thickness, mm	Bearing metal temperature, ^{2) 3)} °C	Maximum velocity, ³⁾ m/s
Radial bearing				
- Fixed geometry	3.8	0.020	115	100
- Tilting pad	4.2	0.020	115	125
Thrust bearings				
- Tapered land	2.5	0.020	115	125
- Flat face	0.5	N/A	115	50
- Tilt pad	3.5	0.015	115	125

NOTE: Table limits will generally not occur all together; one parameter alone may dictate the design.
¹⁾ Limits are for babbitt on steel backing. When other materials are used, established limits for these materials are permissible. Bearing clearances should be chosen to yield proper temperature, high stiffness and stability, as well as to ensure adequate clearance to cope with thermal gradients, whether steady, static, or transient. The average ratio of diametral clearance (S_d) to the nominal bore size (D_n), S_d/D_n , for radial bearings is approximately 0.002 mm/mm.
²⁾ Bearing temperature measurements are taken in the backing material within 3 mm of the backing material/babbitt interface at the hottest operational zone of the bearing circumference.
³⁾ Higher values are acceptable if supported either with special engineering or testing and field experience.

Table– 5: Radial & Thrust bearing details

VII. GEAR HOUSINGS

Gear housings can be made from cast iron, cast steel, welded steel plate and in some Aluminium. The gearhousing 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 mounting is 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 so as to prevent thermal distortion.



VIII. 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 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.

Pitch line velocity, m/s	ANSI/AGMA 2015-1-A01 accuracy grade
35 – 100	A5
100 – 160	A4
Over 160	A3

Table– 6: Pitch line velocity Vs Accuracy grade

IX. MANUFACTURING PROCESS

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 has proved its importance.

Shaving

This 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.

Lapping

This 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.

Grinding

Grinding

Grinding produces the best absolute values of lead and involute profile. 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. 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 secret of success in the manufacture of high speed gearing is good design, and most of all good workmanship.

X. LUBRICATION

The oil furnished to high speed gears has a dual purpose: Lubrication of the teeth, bearings & cooling. Usually 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. 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.

A normal lubricant inlet pressure of 1 to 2 bar is an industry accepted value.

The size and location of the spray nozzles is critical to the cooling and proper lubrication of the gear mesh. Spray nozzles may be positioned to supply lubricant at either the in-mesh, out-mesh, or both sides of the gear mesh (or at other points). At 15,000 fpm oil should be sprayed out-mesh which helps in max. cooling of gear blanks, for 15,000 – 20,000 fpm 90% oil to be supplied to out-mesh & 10% oil for in-mesh this helps minimum amount of oil is supplied to in-mesh, 25,000 – 40,000 fpm oil should be sprayed in sides and gap area of gears to minimize thermal distortion.



It is recommended that the initial charge of lubricant be changed or tested after 500 hours of operation.

XI. GEAR RATING

Durability (Pitting) Resistance similar to AGMA and API

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

Sym.	Description & Units
S _c	Contact Stress Number
W _t	(P _{sc} x 126000)/(n _p x d) Tangential Load
P _{sc}	Transmitted Horsepower
n _p	Pinion RPM
C _p	Elastic co-efficient
C _o	Overload Factor
C _s	Size factor
C _m	Load distribution factor
C _f	Surface Condition factor
C _v	Dynamic Factor
d	Pinion Pitch Diameter
F	Net Face width
I	$I = \left(\frac{.95Z}{P_N}\right) \left(\frac{\cos \phi_t \sin \phi_t}{2}\right) \left(\frac{m_G}{m_G + 1}\right)$ Geometry Factor

Sym.	Description & Units
P _N	Base Pitch, Inch
Z	Length of line of action
φ	Operating Transverse Pressure Angle
m _G	Gear Ratio

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

Sym.	Description & Units
S _{ac}	Base Pitch, Inch
C _L	Life Factor
C _H	Hardness Ratio Factor
C _T	Temperature Factor
C _R	Factor of Safety

Table-7: Durability (Pitting) resistance

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

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

K = Tooth Pitting Index as per API-613
K is proportional (contact stress)²

Allowable K = (Material Index Number) / Service Factor

Strength Rating (AGMA 221)

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

Sym.	Description & Units
S_t	Bending Stress number
ψ	Helix angle
J	Geometry Factor
SF	Service factor
P_n	Normal Diametrical Pitch

Table– 8: Strength rating

J → Geometry Factor which changes for different types of gear with respect to Number of teeth, meshing teeth, etc.

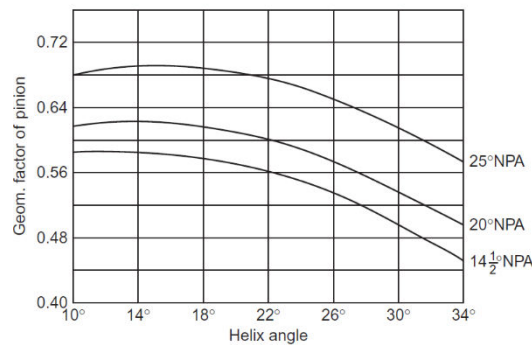


Fig – 8: Geom. Factor of pinion Vs Helix Angle

I. Seals

Where shafts pass through the housing, the housings shall be equipped with seals and deflectors that shall effectively retain lubricant in the housing and prevent entry of foreign material into the housing. Easily replaceable labyrinth-type end seals and deflectors are preferred. The seals and deflectors shall be made of non-sparking materials. Lip-type seals (PTFE) seals are also preferred as these seals are easy replaceable and can overcome shortcomings of labyrinth where oil is prone to leak from gearbox if poorly designed. Surface velocity should be kept within the seal manufacturer’s conservative recommendation.

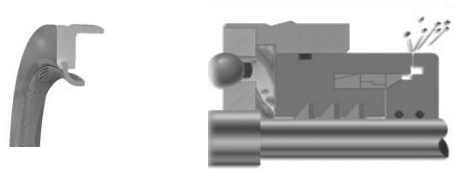


Fig – 9: Oil seal

II. System Influences

Influences have to be accounted while designing gears and shafts, proper selection of bearings for high speed gearboxes which includes Vibration, Coupling misalignments & operational conditions. Actual transmitted loads are much higher due to proximity of torsional or lateral critical speeds, prime mover overload factor, product variations or over-speed.



III. Gear Noise

Gear Noise is mainly caused by Tooth spacing error, lead error, Surface finish, tooth deflections, Improper contact ratio, Too little backlash, Improper tooth tip and root relief, etc.

However, gear noise can be controlled using careful design and improvising the quality of manufacturing. Heavy cast Iron and double walled housings which improvises dampening of vibrations and sound. Acoustical enclosure can also be considered for dampening the Noise level.

Torsional & Lateral Vibrations

When an elastic system is subjected to externally applied, cyclic or harmonic forces, the periodic motion that results is called forced vibration. For the systems in which high speed gears are typically used, two modes of vibration are normally considered:

- a. Lateral or radial vibration, which considers shaft dynamic motion that is in a direction perpendicular to shaft centreline.
- b. Torsional vibration, which considers the amplitude modulation of torque measured peak to peak referenced to the axis of rotation.

XII. CONCLUSION

This paper covered numerous factors like pressure angle, manufacturing process, lubrication, etc., that must be taken into account while developing high-speed gear drives. Also, it covers appropriate gear, seals and bearing types for high-speed gear drives. And the formulas mentioned will demonstrate how the gear's life and strength are calculated for high speed gearboxes.

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