A General Discussion of Gear Configurations

The earliest gear drives utilized a spur gear tooth design. Although modification to the addendum, improves the smooth transfer of load from tooth to tooth with the introduction of a helix angle the transfer of load from tooth to tooth occurs at a slower rate. At the same time load sharing (more than one tooth in contact at any moment which shares the load) is increased due to the increase in the number of teeth in contact at any one time. With the tooth form optimized, this increases the contact or bearing area of the gear rotor consequently increasing surface load carrying capability. However a helix angle design introduces an axial component of force. There are a number of arrangements, which can be used to accommodate the axial force:

1. Variations to Helical Gears

A. **Single Helical Gears**

a. Thrust Bearings

The most traditional method for a single helical design is to utilize a tilting pad thrust bearings or if powers are low enough, an integrally constructed tapered land thrust bearing. The gear thrust is then absorbed by the thrust bearings incorporated within the gearbox. The use of the tapered land type is based on limits of either rotor of .5 N/mm sq. or approximately 4000 HP and/or 4000 RPM.

b. Thrust Collars

One very successful method is to incorporate a thrust collar on the pinion.

This solution eliminates internal thrust. The tangential velocity between the collar and bearing surface is relative and so the resultant sliding velocity is small. A continuous oil film provides a hydrodynamic boundary, which eliminates metal to metal contact thereby providing an infinite life bearing. The disadvantage to this design is that the rotor needed to accommodate the thrust collar or collars increases the span support.

Another problem with very high operating speeds for large power units involves keeping the collars on the shafts as these collars are shrunk on after machining. An example of an extreme gear successfully operating with this type design is a MAAG model type GBX-47 rated for 40000 HP and a speed increaser ratio 4950/9178 RPM. This gear has been in continuous service since 1984. Cost is another factor here however as thrust collars add cost to the gear and may result in a somewhat higher selling price.

c. Elimination of Thrust Bearings in Single Helical Units

An alternate arrangement is to locate the gear train off of a thrust bearing located in the driver or the driven equipment. It is important these thrust bearings be arranged in such a way so they do not oppose each other without consideration to axial flexibility in thermal growth or shaft run out.

B **Double Helical Gears**

Another method is to split the helices into a left and right hand helix.

This method eliminates internal thrust without thrust bearings while still maintaining a helical design for load sharing and smooth transfer of load from tooth to tooth. Without the need for thrust bearings double helical gears tend to be higher in efficiency and require less lube oil than single helical gears.

However, this style gear is subject to a mismatch of the helices between the pinion and the gear. In the case of a single helix, the pinion can be adjusted to properly mesh with the gear whereas in the case of two helices a compromise must be made between left and right hand helices. Mismatch is not necessarily symmetrical but can be accumulative which further complicates mesh alignment. In double helical gears there is a certain error of the apex formed by the helices commonly referred to as apex wander resulting in an axial runout over one revolution of the rotor. This causes a shuttling of the pinion axially with respect to the gear during operation. This is evident as an axial vibration. It is particularly evident during a no load full speed mechanical running test. Under these, no load conditions the pinion is most subject to the internal errors of the gearset. Without any load to dampen or stabilize the pinion, inherent manufacturing errors take over and produce a vibration. In earlier times, double helical gears were constructed of softer through hardened gears, the pinion usually somewhat harder than the gear. This results in the pinion working the gear during operation by wearing it in. This process wears down the uneven loaded surface across the gear face width resulting from the mismatch until even load sharing across the gear face width is achieved. Oftentimes, at first, because the loading can be quite uneven, highly loaded localized areas of the toothing begin to show some initial pitting. Due to the higher sliding velocities of high-speed gears, this pitting could become quite severe. However, as the pinion and gear wear in, this condition slowly heals itself and the gearset develops a high degree of polish. But early day manufactured double helical turbo gears ran at lower speeds than what is demanded of today's highly loaded designs and so this condition was not serious. Also, because of slower speed conditions, the shuttling of the pinion due to apex runout was not a serious concern and this too worked itself out under load as the gearset wore in.

With the increase of speeds in today's turbo market, the resulting PLV (pitch line velocity) has increased due to larger powers and higher speeds. The pinion moves too quickly for it to compensate for toothing errors and therefore does not have sufficient time to shuttle back and forth. As PLV levels exceed 25000 fpm, it is almost impossible for any movement to occur at all. The initial pitting may be severe enough that the quality of the tooth surface becomes questionable resulting in a condition where pitting is now in a destructive realm incapable of repairing itself. Thus the pitting continues until gear failure.

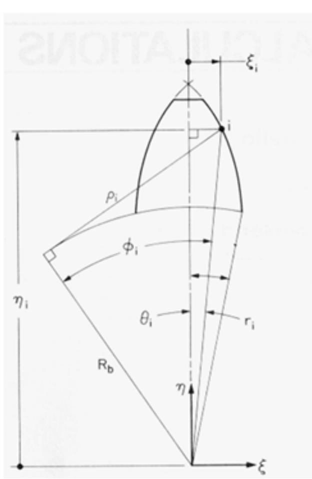
2. How Gear Geometry affects the Pitch Line Velocity of a Gearset

It was evident very early in the history of the development of gear tooth geometry that as higher speeds were demanded from gears, a tooth form that provided quiet rolling contact was essential if the gears were to operate reliably. Hence the need to meet the following goals:

- Lower the sliding action in the gear mesh thereby allowing greater capability for the gearset to carry load. If the pinion/gear tooth surfaces could roll through engagement the surface wear will be reduced.
- Quieter running gears where lower noise is a needed quality. This was demanded of gear design in the earlier history of electric passenger railcar systems where quiet running gears were greatly desirable.

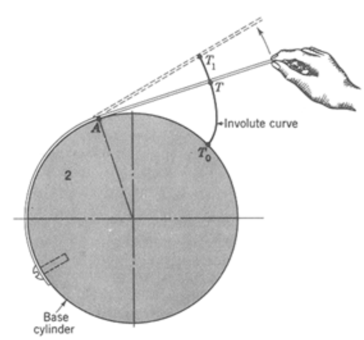
Both of these factors fostered the development of the **involute** tooth form in gear design. This resulted in the optimum tooth form. Today the involute gear tooth form is the accepted design for of all high speed gear manufacturers in the world.

The earliest drives utilized a straight spur gear tooth configuration with an involute tooth form. Modifying the tooth form assist the load entry and load distribution beyond a single tooth engagement while maintaining the involute profile. With the use of a helix angle the effective engagement beyond a single tooth is achieved by overlapping one tooth mesh with another. This added further load sharing of the gear teeth and the number of engaged toothing is referred to as the contact ratio.



$$\phi_i = \cos^{-1}\left(R_b / r_i \right)$$

- R_b – base radius
- r_i – involute radius to point i



The combination for both is referred to as the total contact ratio. This results in an overlap ratio beyond a single tooth.

The total contact ratio is the sum of the transverse contact ratio + the overlap ratio:

$$\epsilon_\gamma = \epsilon_\alpha + \epsilon_\beta$$

The PLV (Ft/min) is calculated by:

$$V = 0.262\left({(2 * a)}/{(u + 1)}\right) * n$$

v = PLV (ft/min)
a = gearset center to center distance, inches
n_1 = pinion rpm
u = gear ratio

From this relationship it can be seen that if the CD can be reduced the PLV will also be reduced. Thus if a gearset can be manufactured with higher load carrying capability, the elements are therefore reduced in size. This is achieved by gear hardening.

The power rating capacity of a carburized gearset is considerably higher thereby reducing the size of the rotor geometry and lowering the PLV. This can be referenced by the allowable stress numbers listed in ANSI/AGMA 2101- D04 Sec. 5.1.3. Both single and double helical gearing benefits from the utilization of case carburized gearing with reduced rotor size and PLV for the same transmitted power. Great care must be given to gear accuracy to mitigate the aforementioned toothing errors.

The purpose of reducing the PLV is to reduce the relative sliding action that occurs in the gear mesh. Although theoretically the involute form follows a line of contact nevertheless as the gear enters into contact and exits from contact with its mating part there exists some relative sliding velocity along the involute profile on the path of contact. The sliding velocity is detrimental to the gearset since it results in a power loss that translates into heat. This heat then increases the temperature of the gearset elements which expands the rotors and distorts the tooth form. The greater the distortion, the poorer the actual form the gear tooth in service becomes which further exacerbates the condition making for a poorly operating gearset.

The combination of further distortion & relative tooth sliding could locally overload limited sections of the mating surfaces. This highly localized condition could fail the lube film thickness bringing the surface teeth into direct contact. The results may be surface distress (scuffing, micropitting, macropitting), thereby shortening gear life.

For high pitch line velocities (greater than 100 m/s (19700 f/m) reliability of gears is challenged by higher face width thermal gradients. Gear tooth distortion, commonly referred to as barreling is a process combination of tooth & rotor deformation observed in service. During gear unit operation there is a high velocity axial flow of lube, oil and air across the helix(s) of the mesh. Through mesh engagement, oil and air are mixed and compressed resulting in an asymmetrical (uneven) tooth flank temperature gradient creating a linear distortion of the tooth lead across the face.

3 An Overview on Gear Tooth Lead Corrections

When a pinion having uniform meshing at no load, is torqued at one end, it bends and twists according to a known algebraic combined deflection, the load distribution is proportional to the tooth deflection.

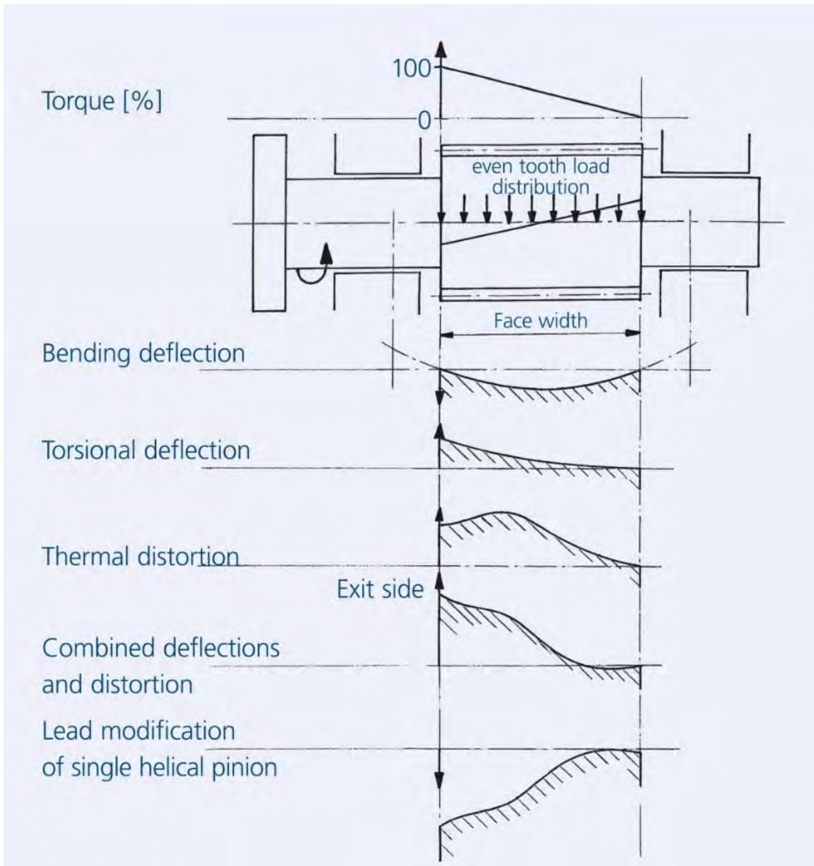
For applications that require large gearboxes high pitch line velocities will increase scuffing risk due to higher sliding velocities and greater tooth quenching losses due to the increased velocity of oil/air combination traveling axially along the tooth face. This will require deeper lead and profile modifications which increase sensitivity to proper tooth contact. Gear teeth distortion, commonly referred to as barreling is a process combination of tooth & rotor deformation as described above.

During gear unit operation the high velocity axial flow of lube, oil and air across the helix(s) of the mesh are mixed and compressed resulting in the asymmetrical (uneven) tooth flank temperature gradient. Lead and profile corrections to account for these deviations must be applied to the gear toothing thereby assuring uniform load distribution. Single helical gears are subject to greater thermal deformation due to lower helix angles and longer travel on the tooth face. Double helical gears benefit from the higher helix angle and shorter face travel thereby reducing the amount of thermal deformation. However the corrections for double helical gears are more complex. If less than optimum correction is discovered in during gear unit operation single helical gears are easily adjusted to align the mesh loads whereas double helical gears can only approximate the adjusted value.

A. Temperature effects

Usually the pinion operates at a higher temperature than the wheel. The pinion will expand and hence the pitch will change variably along the tooth face. The change in axial pitch is most important as this wears the teeth at one end of the helix

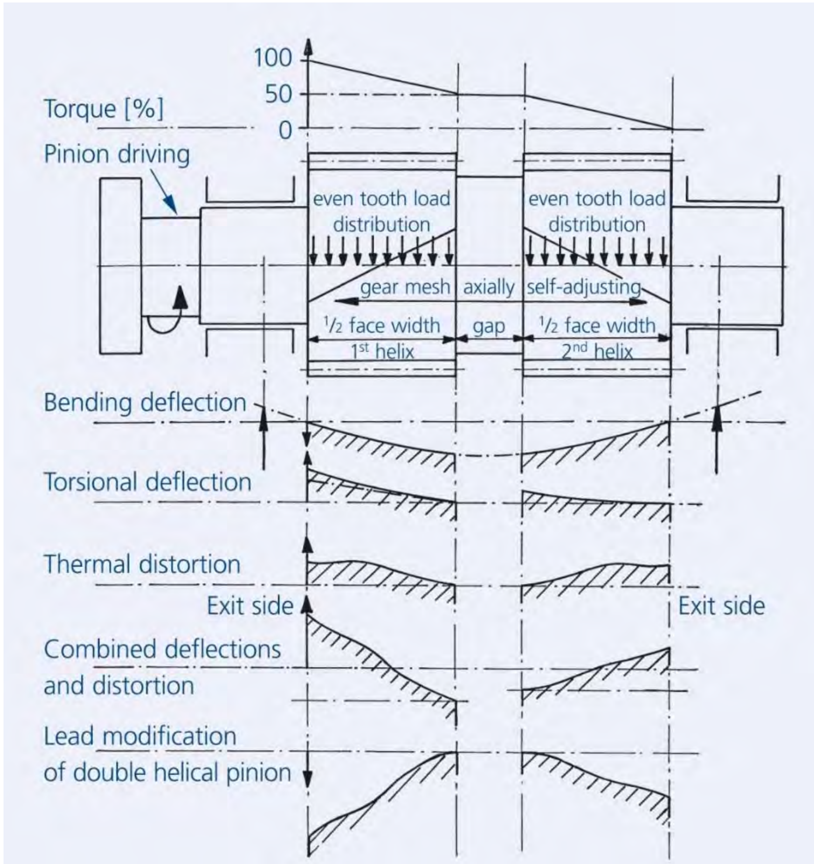
In single helical gears the combination of deflection of deflection and distortion is a single accumulated deviation from the unloaded tooth.



Single Helical Gear – Toothing Deflections of the Lead

In double helical gears the deflection is the same but the thermal distortion is split between the helices. The combined deviation is therefore more complex. However the thermal distortion is of lesser magnitude.

With double helical gears the position of the helices relative to each other is important. If the apex is trailing into the mesh, the teeth bear hard on the inner ends and with apex leading the teeth bear hard on the outer ends. Apex trailing is advantageous as apex leading teeth tend to compound the effect of heat distortion into the torque distortion



Double Helical Gear – Toothing Deflections of the Lead

4 Understanding the Thermal Rating of Gearboxes

A. Introduction

This is a complex issue and not one with easy solutions. In the gearbox or power transmission industry, oftentimes we refer to the thermal efficiency of a gearbox, whether it is an increaser or reducer as the "thermal rating" of the gear unit. This means an assigned value measured in BTU's/min (joules/min) is converted to HP (kW) (power dissipated through the gear unit structure). If the gear unit utilizes a circulating lube oil system designed to both lubricate and cool the gear unit, the residual amount of heat not eliminated by the lube oil system but rather transferred out into the atmosphere or adjacent equipment becomes the measure of the thermal rating.

Gear units vary in size and weight depending on the material of housing construction; for example fabricated steel, cast iron or aluminum. Each has a different heat coefficient. Each will be constructed according to strength. Carburized and ground gears have much higher load carrying capability than gears that are through hardened. This will vary the gear set size. Thus the total physical size of a gear unit can vary greatly depending on the technology used for design and construction of the gear set for the same power and speeds. And so the thermal rating will vary considerably as well. Gears regardless of size in a parallel shaft single mesh configuration will have a mechanical efficiency of approximately 98%. So a rough rule of thumb would be a 2% loss per mesh. Therefore regardless of size the heat to be dissipated is the same. And bigger gears will dissipate heat better than smaller ones. If a gearbox has no circulating system for its lube oil but is depending on the rotors dipping and splashing oil for lubrication than the heat must be dissipated entirely through the housing walls. Sometimes fins are designed onto the housing for increasing the surface area to the atmosphere and thereby increasing the thermal rating. Further capacity to dissipate heat is sometimes accomplished with the addition of a fan or blower.

B. **On High speed gears**

The physical sizes of the gearset rotors are determined by the torque transmitted through the gearset. Torque is really the defining measure of size not power. A low speed machine of relatively modest horsepower can be physically larger than a high speed machine of lesser horsepower. However the inefficiency of the gear unit remains a percentage of the transmitted power. So high speed gears with high horsepower's produce a great amount of heat in a relatively small gear unit. In high speed units therefore the thermal rating becomes a small amount of the unit’s ability to dissipate heat. Also windage losses from the rotors operating with high peripheral speeds will increase the losses and hence increase the amount of produced heat. The unit becomes entirely dependent on the lubrication circulating system for cooling. Much has been done in testing gears for a measurement of their efficiencies and thermal ratings. This includes gears from fractional HP up to 94,000 HP (70 MW).

Here is an approximate breakdown of how the energy is dissipated in high speed gears:

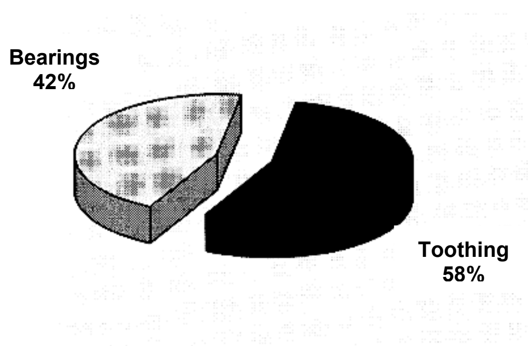


Fig. 1: Origin of gearbox losses

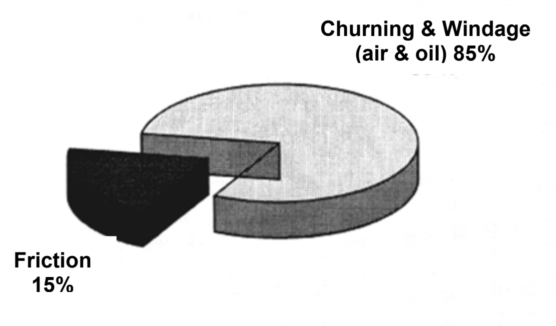
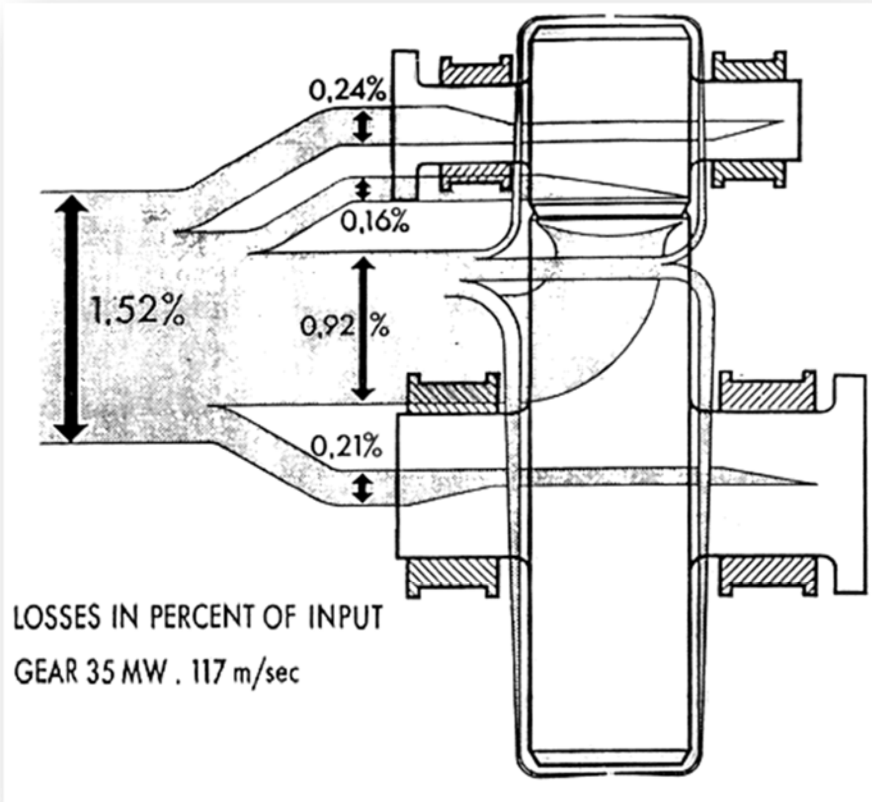
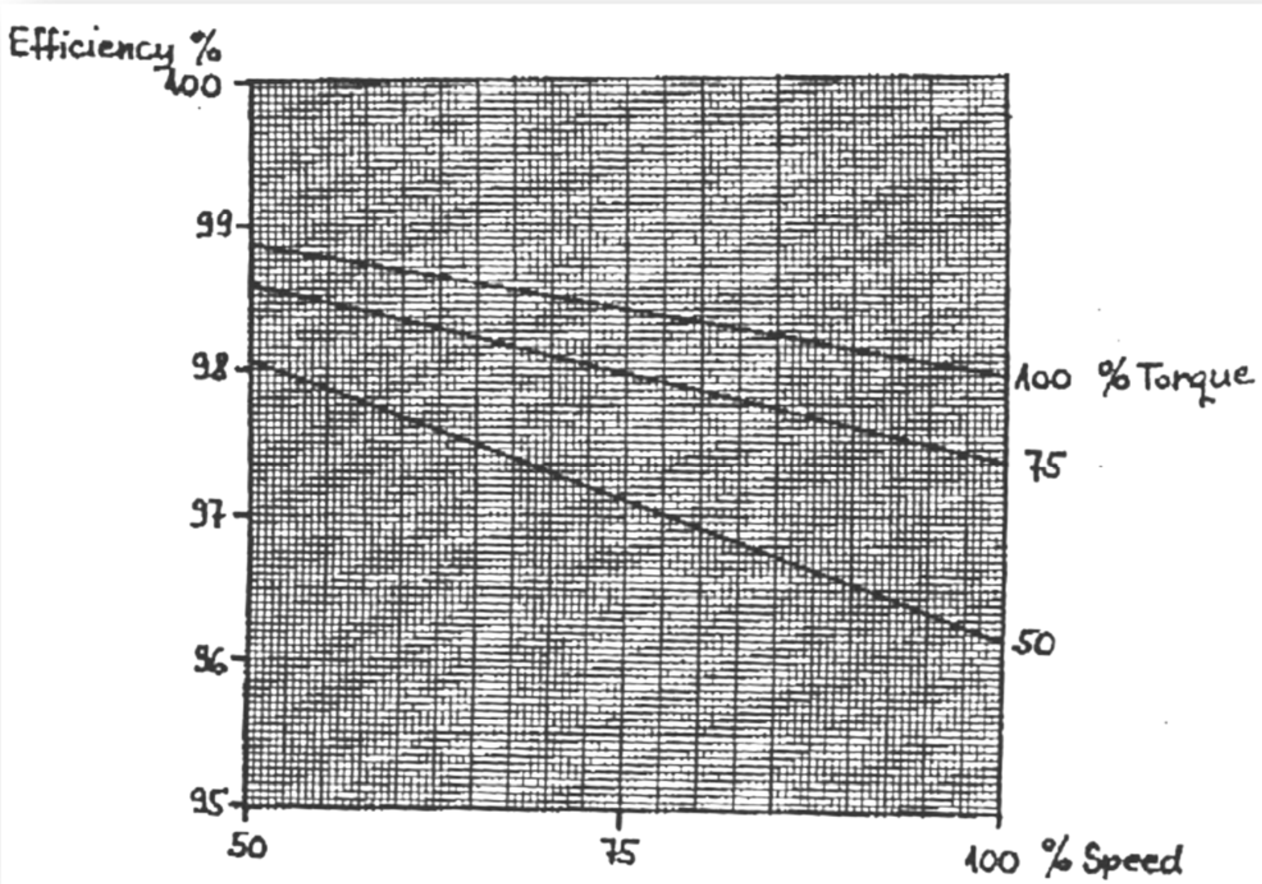


Fig. 2: Origin of toothing losses


0,24%
0,16%
1,52%
0,92 %
0,21%
LOSSES IN PERCENT OF INPUT
GEAR 35 MW . 117 m/sec

Typical illustration of gear efficiency:

In this example: Gearbox is rated @ 29,000 HP - speeds: 4670 rpm/9022 rpm



These efficiency curves are primarily theoretical. If some of the actual operating conditions of torque and speed are points along these curves, then the conditions diverge considerably from the nominal operating conditions for which the gear has been laid out. This means that various points which diverge must be recalculated, taking into account the gear tooth modifications for the nominal condition, for tooth load capacity. The above curves were calculated not considering the effect of the modifications, therefore it brings into question their practical use. Basically one should consider, for instance, that 100% load at 50% speed is only a theoretical point. If this is an actual point, then it should be known at the time the gear is designed to assure the operating conditions are optimized for actual service duty.

Efficiency calculation methods:

- For the use of an analytical method see ANSI/AGMA 6011-I03 Annex F.
- An alternate method can be found in the latest revision of AGMA 6123-B06.

For a simple means of calculating the gearbox oil flow requirement based on a given efficiency the following can be used;

$$Oil\ Flow\ (gpm) = \ power\ loss\ (HP)x\left(\frac{14.27}{\Delta T\ (degF)}\right)$$

Or

$$Oil\ Flow\ (lpm) = \ power\ loss\ (kW)x\ \left(\frac{2.37}{\Delta T\ (dec\ C)}\right)$$

$$\Delta T = T_1 – T_2$$

Where:

T_1 = gearbox oil inlet temperature

T_2 = gearbox oil outlet temperature

6. Epicyclic Gears

Powers and speeds are ever increasing requirements of gear drives increasing the physical size of the gear elements which in turn has resulted in higher pitch line velocities, higher mesh sliding velocities and wider face widths. This in turn places greater emphasis on accurate lead and profile corrections in order to compensate for the operating thermal deformation needed to assure good load distribution and reduced scuffing risk. But the thermal effects generate higher operating temperatures demanding closer attention to the quality of the lubricant employed, size and quality of the lubricating system and the frequency of maintenance schedules. Further metallurgical considerations which affect the behavior of the material in its operational environment is extremely important such as the heat treatment procedures and, relative amount of retained austenite in carburized elements.

The use of epicyclic gears has been avoided by API due to the complexity of service and repair. However there are alternative solutions available which make these concerns more tolerable. Best of all epicyclic gearing directly provides alternative layouts that address the consequences listed above when compared to single mesh power paths in parallel shaft gear units.
The advantage is a significant reduction in the pitch line velocity for a comparable parallel shaft unit with a single mesh power path thereby reducing the negative impact created by thermal affects.
There are commercial advantages as well:
1. Envelope size (smaller than parallel shaft for same power) therefore less installed space required.
2. Coaxial Shafts (in line system) resulting in more compact installation
3. Lower weight
4. Lower cost for the entire train layout

Epicyclic units is an alternative to applications where special considerations can take advantage of the epicyclic layout. The mesh rating of the gears in epicyclic units can apply to the same rating calculations for parallel shaft gears.