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Design recommendations for bevel gears

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Design recommendations for bevel gears

Recommandations pour le dimensionnement d'engrenages coniques

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Foreword

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ISO/TR 22849 was prepared by Technical Committee ISO/TC 60, *Gears*, Subcommittee SC 2, *Gear capacity calculation*.

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Design recommendations for bevel gears

1 Scope

This Technical Report provides information for the application of bevel and hypoid gears using the geometry in ISO 23509, the capacity as determined by ISO 10300 (all parts) and the tolerances in ISO 17485.

This Technical Report provides additional information on the application, manufacturing, strength and efficiency of bevel gears for consideration in the design stage of a new bevel gear set.

The term “bevel gear” is used to mean straight, spiral, zerol bevel and hypoid gear designs. Where this Technical Report pertains to one or more, but not all, the specific forms are identified.

The manufacturing process of forming the desired tooth form is not intended to imply any specific process, but rather to be general in nature and applicable to all methods of manufacture.

This Technical Report is intended for use by an experienced gear designer capable of selecting reasonable values for the required data based on his/her knowledge and background. It is not intended for use by the engineering public at large.

2 Symbols, descriptions and units

The symbols and descriptions used in this Technical Report are, wherever possible, consistent with other International Standards on bevel gears. As a result of certain limitations, some symbols and descriptions are different than in similar literature pertaining to spur and helical gearing.

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Symbol	Description	Unit
A_g	Arrangement constant	—
a_v	Centre distance of virtual cylindrical gears	mm
α_n	Generated pressure angle according to ISO 23509	°
α_{vat1}, α_{vat2}	Pressure angle at tip of virtual cylindrical gear	°
α_{vt}	Pressure angle in transverse plane of virtual cylindrical gear	°
b_{eff}	Face width in contact with mating element	mm
β_e	Outer spiral angle according to ISO 23509	—
β_{m1}, β_{m2}	Mean spiral angle	°
β_v	Spiral angle of virtual cylindrical gear	°
C_1	A constant	—
D	Outside diameter of the considered rotating element	mm
d_{ae1}, d_{ae2}	Outside diameter	mm
d_{v1}, d_{v2}	Reference diameter of virtual cylindrical gear	mm
d_{va1}, d_{va2}	Tip diameter of virtual cylindrical gear	mm
$\Delta\alpha_{t1}$, $\Delta\alpha_{t2}$	Change in pressure angle from pitch point to outside	°
δ_{a1}, δ_{a2}	Face angle	°
δ_1, δ_2	Pitch angle	°
ΣP_{GW}	Sum of power losses regarding churning	kW
f_g	Gear dip factor	—
φ	Friction angle	°
h_{am1}, h_{am2}	Mean addendum	mm
η_{ffc}	Sum of element churning efficiency (see 6.1.6)	—
η_{ffl}	Lengthwise sliding efficiency (see 6.1.5)	—
η_{ffp}	Profile sliding efficiency (see 6.1.4)	—
j_{et}	Outer transverse backlash	mm
j_{en}	Outer normal backlash	mm
K	Load intensity for calculating the coefficient of friction	N/mm^2
L	Length of the element of the considered rotating element	mm
m_t	Transverse tooth module of the gear considered	mm
μ_m	Coefficient of friction (see 6.1.6)	—
n	Rotational shaft speed	r/min
υ, υ_κ	Kinematic oil viscosity at operating temperature, kinematic viscosity at 40 °C	mm^2/s (cSt)
v_{et}	Pitch line velocity at outside diameter	m/s
P	Design power	kW
P_{GWi}	Power loss for each individual element	kW
R_{m1}, R_{m2}	Mean cone distance	mm
R_f	Roughness factor	—
T_{o2}	Output torque, wheel, per unit force	mm
T_{i1}, T_{i2}	Input torque per unit force, pinion and wheel	mm
t_{B1}, t_{B2}	Pinion back angle distance	mm
t_{E1}, t_{E2}	Pinion crown to back	mm
t_{F1}, t_{F2}	Pinion face angle distance	mm
T_1	Pinion torque	Nm
z_1, z_2	Number of pinion, wheel teeth	—

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3 Application

3.1 Geometry

For the purposes of this Technical Report, the geometry of bevel and hypoid gear pairs is assumed to be calculated according to ISO 23509. These calculations need at least a set of initial data. If these data are not completely given or known from similar applications, a rough estimate of the gear dimensions can be determined by means of the power to be transmitted (see Annex B of ISO 23509:2006).

In any case, a complete geometry calculation has to be successfully executed before any other of the following considerations makes sense.

3.2 Rating

3.2.1 General

To make a rating of a pair of bevel gears one should have a mathematically correct set of geometry (see 3.1). This enables the designer to proceed to more detailed calculations which complete the design insofar as the transmitted torque is concerned. Additional rating criteria for bending strength and pitting resistance should also be considered. The method for calculating the bending strength and pitting resistance of bevel gears except hypoid gears is stated in ISO 10300 (all parts).

3.2.2 Bending strength

Bending strength as a criterion of bevel and hypoid gear capacity can be defined as the ability of the gear set to withstand repeated or continued operation under nominal load without fracture of the teeth in their roots by fatigue in bending. It is a function of the bending (tensile) stresses in a cantilever beam and is proportional to the applied load. It also involves the fatigue strength of the gear materials and the shape of the teeth. Therefore, either the pinion or the wheel can be the limiting member of the pair.

3.2.3 Pitting resistance

Pitting resistance as a criterion of bevel and hypoid gear capacity can be defined as the ability of the gear set to withstand repeated or continued operation under nominal load without suffering destructive pitting of the tooth surfaces. The experienced gear designer recognizes that moderate, non-destructive pitting of the tooth surfaces can occur during the early stages of operation, especially on non-hardened or through-hardened gears. In these cases, the pitting ceases to progress after the asperities have been removed by the initial operation. This process, called initial pitting, should not affect the gear life.

Destructive pitting, although attributable in principle to the same phenomena, progresses widely enough to destroy the geometry of the flank surfaces and ultimately leads to failure. The distinction between initial and destructive pitting is defined more thoroughly in ISO 10825.

Pitting is a function of several factors; the most significant is Hertzian contact (compressive) stresses between the two mating tooth surfaces and is proportional to the square root of the applied tooth load. The ability of bevel and hypoid gear teeth to withstand repeated surface contact under load without destructive pitting involves the resistance of the gear material to fatigue under contact stresses. The smaller gear is usually the limiting member of the pair because the teeth receive more stress cycles per unit time. In some cases, the smaller gear is made harder than its mate, to increase its surface durability so that the limiting capacity can exist in either member.

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3.2.4 Other forms of bevel gear tooth deterioration

The rating standards are not applicable to other types of gear tooth deterioration such as micropitting, case crushing, wear, plastic yielding and welding.

Information on scuffing can be found in ISO/TR 13989-1.

3.3 Materials

The quality of materials and methods of heat treatment required are governed by the application. Care should be taken to choose the proper material for each application to transmit the load and obtain the life desired. Heat treatment is usually needed to develop the necessary hardness, strength and wear resistance.

For information about materials and heat treatment, see ISO 6336-5.

3.4 Gear tolerances

Bevel gears are manufactured to suit many engineering applications. In order to satisfy these needs properly, it is necessary to analyse the conditions under which these gears should operate. Reasonable tolerances should then be established to ensure that the gears perform satisfactorily in the application.

Tolerance values for unassembled bevel gears, hypoid gears and gear pairs are provided in ISO 17485. Additionally, information about bevel gear measurement methods is given in ISO/TR 10064-6.

3.5 Gear noise

3.5.1 General

The gear noise can be produced by the vibration of the gear unit caused by the transmission error of the gear pair. The flank form deviations of the teeth, a misalignment between the gears, and the elastic deformation of the teeth under load affect the transmission error. Table 1 shows typical values of transmission errors for different gear applications.

Table 1 — Typical values of transmission error

Application	Recommendation value µrad
Passenger car	<30
Truck	20 to 50
Industrial	40 to 100
Aircraft	40 to 200 (80 average)

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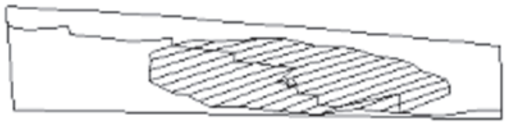
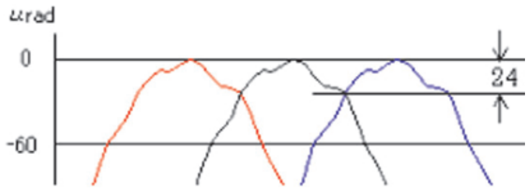
3.5.2 Tooth flank form corrections

The tooth flank form of bevel gears is corrected in order to prevent edge contact of tooth bearing during operation. Figure 1 a) shows the tooth flank form deviation of a spiral bevel gear after lapping. The amount of deviation between adjoining contour lines is 2 µm. It turns out that a crowning of remarkable size occurs in face width direction. On the other hand, the amount of deviation in the profile direction is small. Figure 1 b) shows the pertaining tooth bearing and Figure 1 c) the waveform of the transmission error. The peak-to-peak value of the transmission error is 24 µrad.

a) Tooth flank deviation



b) Tooth bearing



c) Transmission error

Figure 1 — Example of a gear pair finished by lapping process

Figure 2 shows the effect of profile crowning and flank twist where the amount of lengthwise crowning is fixed at 20 µm. In the case of 5 µm profile crowning in Figure 2 a), the width of tooth bearing is wide, and the transmission error is 27 µm. On the other hand, in the case of 20 µm profile crowning in Figure 2 b), the width of tooth bearing is narrow, and the transmission error increases to 43 µm. This means that excessive profile crowning should be avoided. However, in the case of Figure 2 c) with a flank twist correction of 80 µm, the transmission error decreases to 24 µrad. This shows the effectiveness of flank twist modifications if the profile crowning is enlarged.

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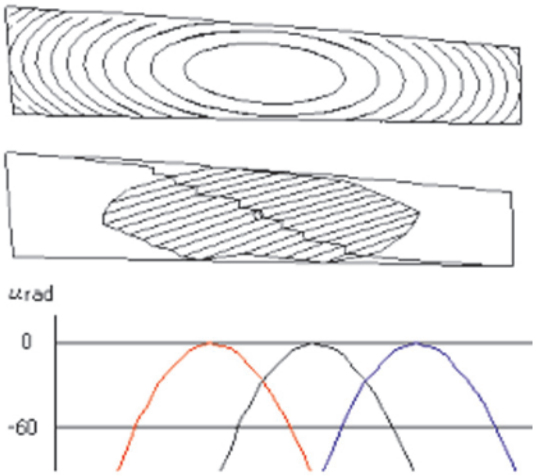
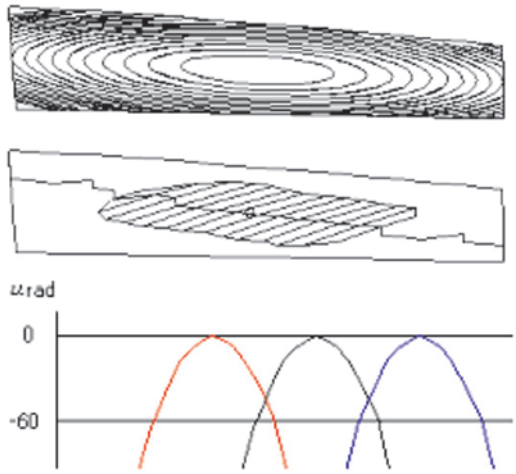


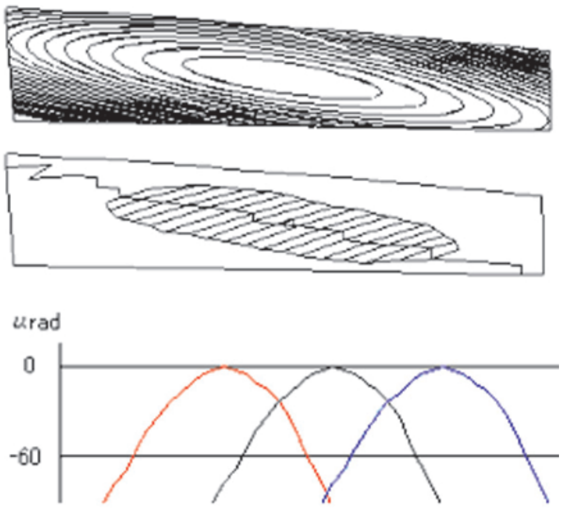
NOTE The transmission error is 27 µrad.

a) Profile crowning of 5 µm



NOTE The transmission error is 43 µrad.

b) Profile crowning of 20 µm



NOTE The transmission error is 24 µrad.

c) Profile crowning at 20 µm and flank twist of 80 µm

Figure 2 — Effect of profile crowning and flank twist on transmission error — Lengthwise crowning of 20 µm

Since tooth flanks are subject to elastic deformations under load, this needs to be considered for flank form corrections. However, as the noise of a gear set in many cases becomes a problem under light load, the measure indicated above is rather effective.

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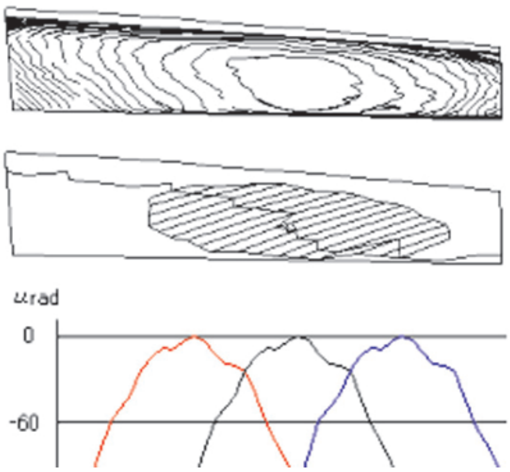
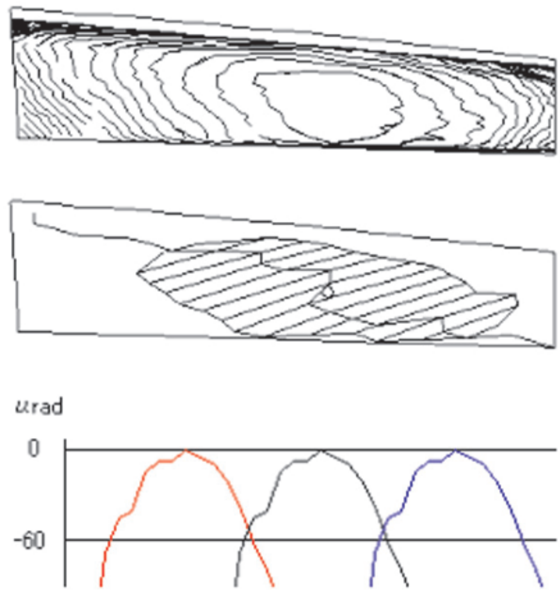
3.5.3 Design contact ratio



NOTE The transmission error is 24 µrad.

a) Number of teeth of 10/43 and contact ratio of 3:02



NOTE The transmission error is 52 µrad.

b) Number of teeth of 7/30 and contact ratio of 2:23

Figure 3 — The effect of design contact ratio on transmission error

The design contact ratio is the angle of transmission of one pair of teeth divided by the angular pitch. It is favourable, therefore, to enlarge the design contact ratio of a gear set in order to reduce gear noise. To get a higher design contact ratio, it is effective to increase the number of teeth, to enlarge the working tooth depth and to increase the spiral angle. However, if the number of teeth is increased, the mean normal module becomes smaller and reduces the load carrying capacity. Moreover, there is a risk that undercut can occur in the pinion root or the topland can become too small if the tooth depth is enlarged too much. Caution is required in those points.

Figure 3 shows the effect of design contact ratio on the transmission error. In the case of smaller tooth numbers of 7/30, the contact ratio which is less than that of 10/43, the size of the tooth bearing increases, but the transmission error also increases from 24 µrad to 52 µrad, although the tooth flank deviations are the same.

The actual contact ratio can change under load by deformation of the flanks and deflections of teeth and shafts.

3.5.4 Other noise consideration

Where a large misalignment is in the mountings of a gear pair or the misalignment produced by deflection under load is considerable, the tooth flanks can have edge contact and the transmission error can increase. Therefore, caution is advised to make the mountings of the gears accurate and the rigidity of the gearbox high.

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4 Manufacturing consideration

4.1 Outline of production methods and their features — Face milling and face hobbing

In principle, there are two different methods used for manufacturing spiral bevel gears and hypoid gears: single indexing, which is also called face milling (FM) and continuous indexing, which is also called face hobbing (FH).

For the FH method, the rotation of the tool and of the workpiece are coupled in a fixed ratio so that one blade group of the cutter head enters one tooth gap and the next blade group enters the next gap, etc. This method is called continuous indexing, where all gaps are cut at the same time and which produces an epicycloid in the lengthwise direction. Generally, the tooth depth is constant along the face width so that root angle and face angle are equal. The tooth geometry results in a tapered topland and a tapered slot width. With a reasonably sized cutter radius, the tooth gap at the toe is slightly smaller than at the heel. If the cutter radius is too small, the inner tooth end becomes thicker than the outer. Therefore, too small cutter radii should be avoided.

In the FM method, the cutter blades are set in a circle on the cutter head. This method is used for cutting and for grinding. The tooth gaps are manufactured by single indexing which means that one gap is finish cut (or ground), then the gear blank is rotated by one pitch and the next gap is cut. Consequently, this method produces a circular arc in tooth lengthwise direction and in its standard geometry the tooth depth is tapered as well as topland and slot width. However, if root angle and face angle are specifically changed by a tilted root line depending on the cutter radius, the tapered tooth gets constant slot width and nearly constant topland, while maintaining proper space width taper (see 5.3.2.1 of ISO 23509:2006). The advantage of this measure is that FM bevel gears can also be completely cut in one single clamping.

Although there is an obvious difference between face hobbed gears and face milled gears, it does not lead to a general rule that one or the other method gives better results. The only fact is that for hardened spiral bevel gears no grinding process exists with continuous indexing, but instead precision hard cutting. Moreover, for bevel gears with diameters of more than 1 000 mm, there is no other way than continuous hard cutting because such big grinding machines are not available.

Regarding operating properties and load carrying capacity, no difference between FH and FM bevel gears can be found, if all crucial parameters are kept the same. These findings are also promoted by modern tooth contact analyses and FM calculation programs by which bevel gear designs can be checked and optimized. These programs also allow the study of detailed tooth flank modifications prior to manufacturing.

Historically, the choice of the manufacturing method was determined by the cutting machine available from a particular distributor. Nearly all new machines are 6-axes CNC machines, which can realize both face milling as well as face hobbing, and most of the current submethods.

Any heat treatment distortions are independent of the FH or FM method. With small distortions, lapping is the usually applied finishing process which also works equally with FH and FM bevel gears. However, in the case of larger distortions, a grinding or cutting process is required. Then, it is obvious to use grinding for FM gears and hard cutting for FH gears as their respective geometries are identical. Face hobbed gears can also be ground, however by single indexing, and both flanks should be ground separately for a correct engagement.

Unfortunately, hardening distortions hinder a geometrically stable lapping process. If these distortions are known exactly in advance, the flank form can be modified in the cutting process to compensate for the distortion so that the lapping process can be used more effectively. Contrary to grinding and hard cutting, lapping is a process without high geometric consistency but with the advantage of less noise emission by reducing relative tooth profile error between pinion and wheel, if heat treatment distortion is limited. Lapping does not eliminate any deviation in pitch and runout. Generally, the lapping process cannot be used to apply any designed flank or profile modification. This is possible with grinding and precision hard cutting only.

The cutting method and finishing method that should be used depends on the intended use of the respective gears, the equipment available and a lot of other aspects.

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4.2 Blank design and tolerances

4.2.1 General aspects

The quality of any finished gear is dependent on the design and accuracy of the gear blank. A number of important factors which affect cost, as well as performance, should be considered.

Bores, hubs, and other locating surfaces should be in proper proportion to the gear diameter and module. Small bores, thin webs, and any condition that results in excessive overhang and deflection, should be avoided.

4.2.2 Clamping surface

Nearly all bore-type bevel gears are held by means of a clamp plate at the front face of the hub when the teeth are being cut; therefore, the blank should incorporate a suitable surface for this purpose, as shown in Figure 4.

Key

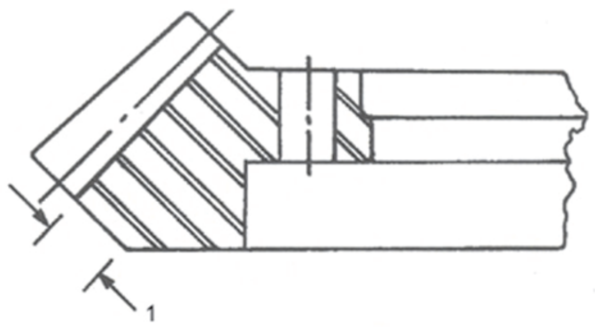
1 no surface provided for clamping, not recommended

2 clamping surface, as recommended

Figure 4 — Recommended clamping surface of the blank

4.2.3 Tooth backing

Sufficient thickness of metal should be provided under the roots of gear teeth to give proper support for the teeth. It is suggested that the minimum amount of material under the teeth not be less than the whole depth of the tooth. Highly stressed gears can require additional backing. This material depth should be maintained under the small ends of the teeth as well as under the middle (see Figure 5). In addition, on webless-type wheels the minimum stock between the bottom of the tap drill hole and the gear root line should be one third the tooth depth.



Key

1 tooth backing, as recommended

Figure 5 — Tooth backing

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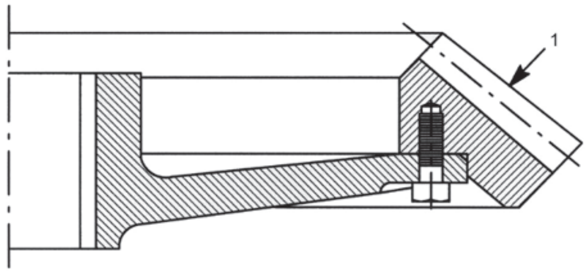
4.2.4 Load direction

A gear blank should be designed to avoid excessive localized stresses and serious deflections within itself. For heavily stressed gears, a preliminary analysis of the direction and magnitude of the forces is helpful in the design of both the gear and the mounting. Where possible, the direction of the web should coincide with the direction of the resultant tooth load in an axial section. Gear sections should be designed in such a way that a component of the tooth load is directed through the section as shown in Figure 6. See ISO 23509:2006, Annex D, for detailed discussions of tooth loads.



Key

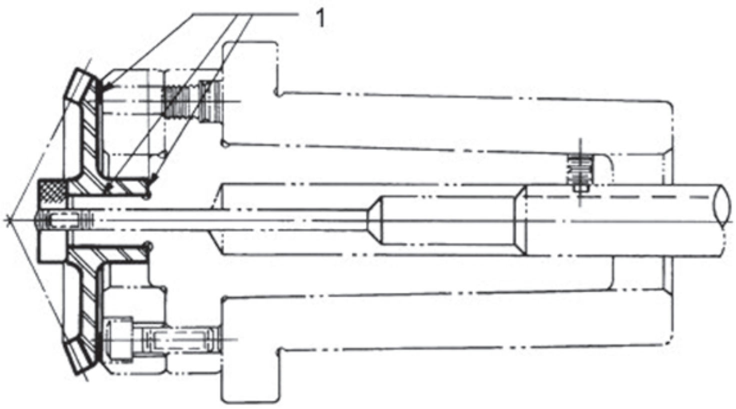
1 tooth load component

Figure 6 — Webless mitre gear — Counterbored type

4.2.5 Locating surface

The back of the gears should be designed with a locating surface of generous size. This surface should be machined or ground square with the bore and is used both for locating the gear axially in assembly and for holding it when the teeth are cut. The front clamping surface should, of course, be flat and parallel to the back surface. A flat and parallel surface also provides a convenient inspection surface after installation.

Gears with a comparatively large ratio of pitch diameter to hub diameter, greater than 2,5:1, should have an auxiliary locating surface behind the teeth as shown in Figure 7. A similar surface should also be used for thin-webbed gears where there is danger of blank distortion or vibration from cutting forces.



Key

1 suggested locating surfaces

Figure 7 — Suggested locating surfaces

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4.2.6 Solid shanks

Where gears with solid shanks are made in large quantities, a collet chuck is usually used. For small quantities, the gears should be provided with a tapped hole or external threads at the end of the shank to hold the gear securely in the chuck while cutting the teeth (see Figures 8 and 9).

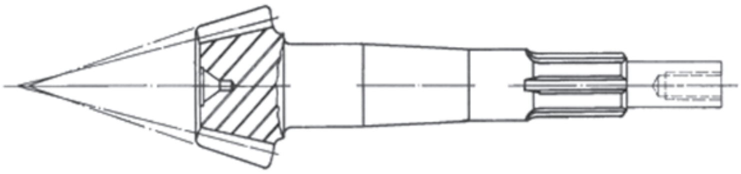
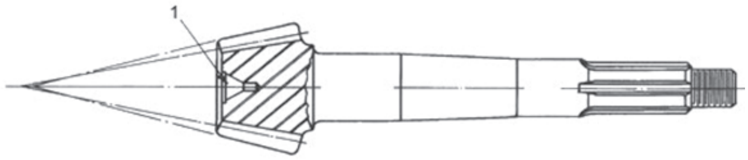


Figure 8 — Shank-type pinion with tapped hole



Key

1 centres (as large as possible and relieved as shown)

Figure 9 — Shank-type pinion with external threads

4.2.7 Flanged hub

Whether the gear is mounted on a flanged hub or is made integral with the hub, the supporting flange should be of sufficient section size to prevent deflections in the direction of the gear axis at the mesh point.

The web preferably should be made conical without ribbing to permit rough machining of the blanks for obtaining better balance, to eliminate oil churning when dip lubrication is used, and lessen the danger of stress concentration being set up within the castings.

4.2.8 Splined bores

In mounting gears with splined bores, a piloting diameter is suggested to reduce eccentricity. Hardened gears with straight-sided splines in the bore should be piloted in assembly by the bore or minor diameter of the splines, which should be ground concentric with the teeth after hardening. Unhardened gears with straight-sided splines should be piloted in assembly by the major diameter of the splines. In either case, the finish machining of the blank, cutting of teeth and the soft testing should be performed with the gear centred on the arbor by the bore, which has been machined true with the splines.

Figure 10 shows a gear with a cylindrical fit at each end of the bore, the splines being used for driving only. This type of fit is particularly applicable to aircraft gears, which often use involute splines with a full fillet radius on the major diameter. This design is an excellent solution, particularly when the splines have to be hardened, because fitting on the sides of the splines is extremely difficult when size changes and distortion takes place during heat treatment.

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Involute splines generally fit on the side of the spline only. Where gears are hardened, it can be necessary to resort to lapping or grinding of splines, or to selective assembly, or both. Even when the splines are shaped after hardening, it is difficult to obtain the accuracy of fit and the concentricity desired for precision gears. Precision finishing the teeth of the gear on involute splined arbors after the splines have been shaped, results in considerable improvement, but even then, different degrees of eccentricity are obtained by shifting the gear to different positions on a splined arbor or shaft.

Since heat treatment can introduce distortion and out-of-round conditions in the splines which cannot be corrected, it is important that the splines be of no greater length than is actually required for load transmission. Splines should be located as near the gear teeth as possible on blanks with long hubs.

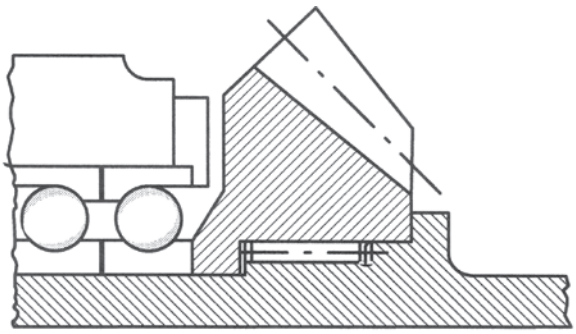
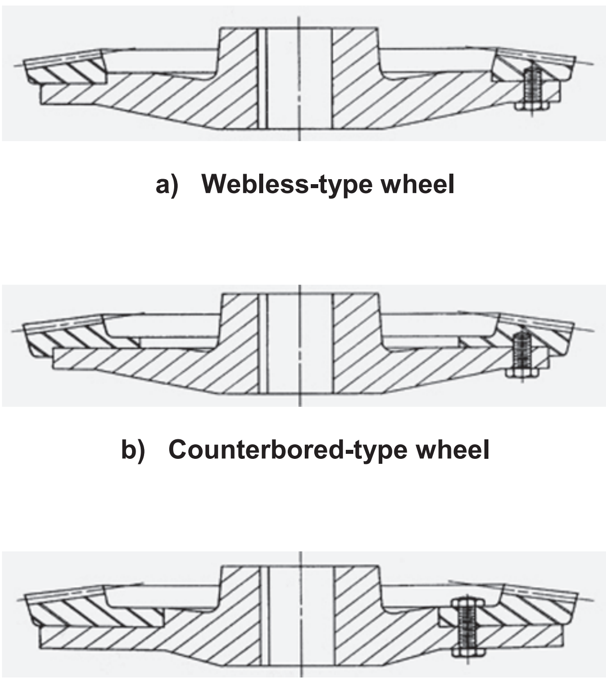


Figure 10 — Spline mounting

4.2.9 Ring-type designs

The most common wheel designs are shown in Figure 11.



a) Webless-type wheel

b) Counterbored-type wheel

c) Web-type wheel

Figure 11 — Typical wheels mounted on hubs

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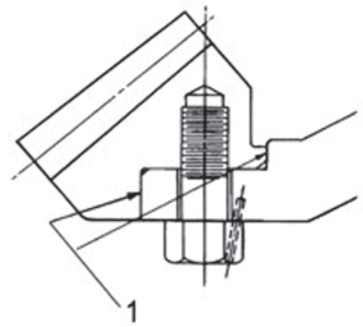
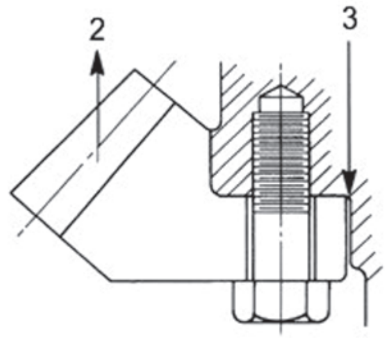
Of these, the bolted-on webless wheel design shown in Figure 11 a) is best for hardened gears larger than 180 mm in diameter. These relatively large hardened wheels are usually made in a ring shape and, subsequently, mounted on a hub or centre, because the ring form can be more effectively hardened in quenching dies.

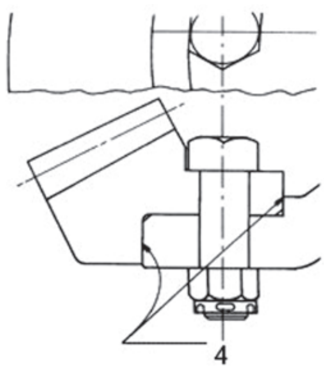
The fit of the wheel on its centring hub should either be a size-to-size fit or a slight interference fit. These gears should be mounted on the centring hub as shown in Figure 12 a) and b), or with through bolts as shown in Figure 12 c). Several methods of locking screws and nuts in place are indicated in Figure 12. The method shown in Figure 12 b) can be used for mounting gears that operate with an inward thrust only. Designs where gear loads increase screw or bolt tension should be avoided.



a) Method of centring counterbored-type gear on gear cutter



b) Method of mounting gear when thrust is inward



c) Use of bolt with castellated nut

Key

1 centre gear on one of these surfaces

2 thrust direction

3 load on inside face of web in this case; otherwise not recommended

4 centre gear on one of these surfaces

Figure 12 — Methods of mounting gear

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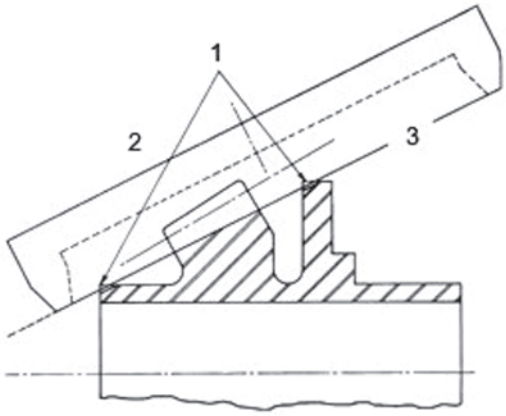
4.2.10 Dowel

On reversing or vibrating installations, separate dowel drives may be used. The use of dowels or body fitted bolts has been found unnecessary in most automotive and industrial drives. If bolts or cap screws are drawn tightly, the friction of the wheel mounting surface prevents bolt shear. Hardened gears smaller than 180 mm in diameter may be of conventional design with integral hubs.

4.2.11 Hub projections

All hub projections (front or rear), which extend above the root line, as shown in Figure 13, should be eliminated.



Key

1 blank turned off for cutter clearance

2 cutter

3 root line

Figure 13 — Example of required cutter clearance

4.2.12 Blank tolerances

4.2.12.1 General

In dimensioning bevel gear blanks, it is necessary to specify properly the items important to the functioning of the teeth. There are two accepted methods for specifying blank tolerances, which are given in 4.2.12.2 and 4.2.12.3.

4.2.12.2 Method 1

This method can be used easily and accurately on either the gear blanks or the finished gears. Items that should be checked include

a) face angle distance,

b) back angle distance, and

c) bore or shank diameter.

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The face angle distance and back angle distance are obtained in the following manner:

$$t_{F1} = 0{,}5\, d_{ae1} \cos\delta_{a1} + t_{E1} \sin\delta_{a1} \tag{1}$$

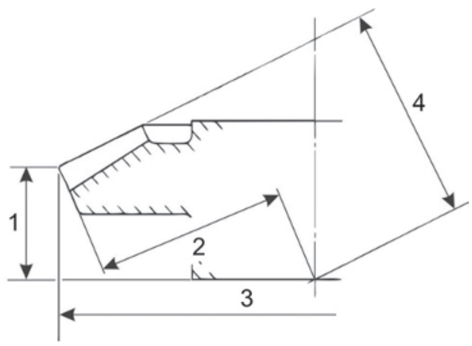
$$t_{F2} = 0{,}5\, d_{ae2} \cos\delta_{a2} + t_{E2} \sin\delta_{a2} \tag{2}$$

Back angle distances:

$$t_{B1} = \frac{t_{F2} - \dfrac{t_{E1}}{\sin\delta_1}}{\tan\delta_1} \tag{3}$$

$$t_{B2} = \frac{t_{F2} - \dfrac{t_{E2}}{\sin\delta_2}}{\tan\delta_2} \tag{4}$$

Figure 14 shows method 1 for dimensioning the gear blanks when this method of specifying tolerances should be followed.



Key

1 crown to back (ref.)

2 back angle distance

3 outside diameter (ref.)

4 face angle distance

Figure 14 — Method 1 for specifying blank tolerances on bevel gears

Tables 2 and 3 give suggested tolerances for face angle and back angle distances and bore or shank diameter.

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Table 2 — Face angle and back angle distance tolerances

Mean normal module, m_{mn} mm	Tolerance mm	
	Face angle distance	Back angle distance
0,3 and finer	+0,00 −0,03	+0,03 −0,03
0,3 to 0,5	+0,00 −0,08	+0,05 −0,05
0,5 to 1,25	+0,00 −0,10	+0,08 −0,08
1,25 to 10	+0,00 −0,10	+0,10 −0,10
10 and coarser	+0,00 −0,13	+0,13 −0,13

Table 3 — Suggested tolerances for bore or shank diameter

Nominal locating bore or shank diameter mm	Suggested tolerance mm					
	Accuracy grades 2 and 3 according to ISO 17485		Accuracy grades 4 and 5 according to ISO 17485		Accuracy grades 6 to 9 according to ISO 17485	
	Shank	Bore	Shank	Bore	Shank	Bore
Up to 25	+0,000 −0,005	+0,005 −0,000	+0,000 −0,013	+0,013 −0,000	+0,000 −0,030	+0,030 −0,000
25 to 100	+0,000 −0,008	+0,008 −0,000	+0,000 −0,013	+0,013 −0,000	+0,000 −0,030	+0,030 −0,000
100 to 250	+0,000 −0,013	+0,013 −0,000	+0,000 −0,025	+0,025 −0,000	+0,000 −0,050	+0,050 −0,050
250 to 500			+0,000 −0,025	+0,025 −0,000	+0,000 −0,080	+0,080 −0,000
500 and larger			+0,000 −0,050	+0,050 −0,000	+0,000 −0,100	+0,100 −0,000

4.2.12.3 Method 2

This method can be used easily and accurately on bevel gear blanks only and is mostly used for large components and for single piece manufacturing purpose. The gear blanks should be machined to the shapes shown in Figure 15 (left: pinion type, right: wheel type), i.e. using flat, straight cylindrical forms, but no radii, in order to get proper measuring results. Also, in this method the crown point is lost.

This checking procedure needs calculated reference dimensions L1, L3, L4, D2, D4, based on selected dimensions D5 and L4 (wheel) as well as L1 and L3 based on selected dimensions D5 and L4 (pinion).

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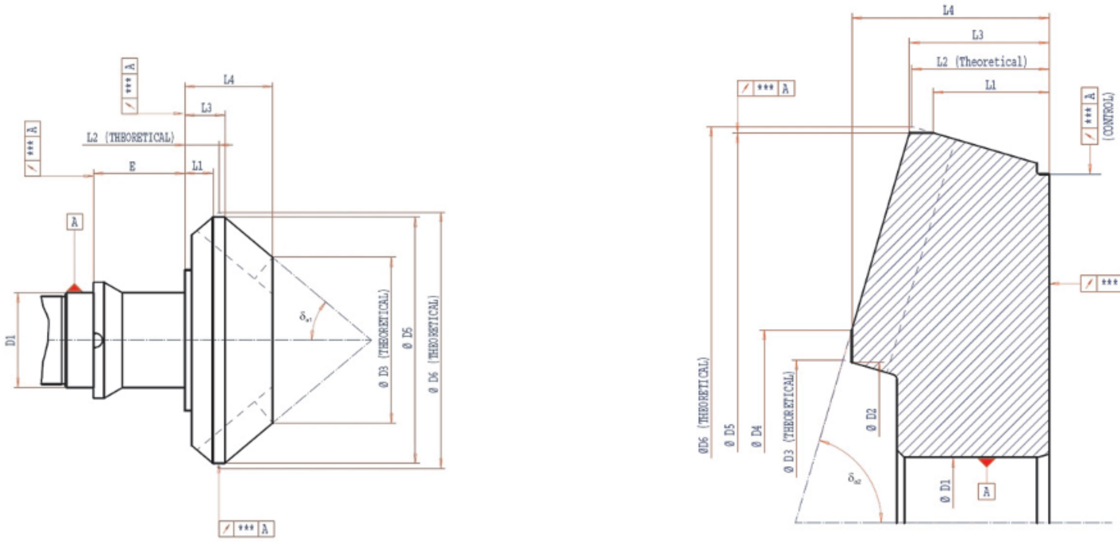

Figure 15 — Method 2 for specifying blank tolerances on bevel gears

Dimension E determines the location of the pinion holding device, which is important for the blank cone positioning. Items that should be checked include the following.

a) Wheel: L1, L3, L4, D2, D4 and D5.

These dimensions ensure the positioning of the wheel for tooth cutting.

b) Pinion: L1, L3, L4, D3, D5 and E.

These dimensions ensure the positioning of the pinion for tooth cutting.

Table 4 gives suggested tolerances for the dimensions L (1, 3, 4), D (1 to 5) and E.

Table 4 — Suggested tolerances for blank dimensions

Dimension	Module 2 to 5	Module greater than 5 to 10	Module greater than 10
L1	js12	js12	js12
L3	js12	js12	js12
L4	h8	h8	h8
E (pinion)	h12	h12	h12
D1[a]	see Table 3	see Table 3	see Table 3
D2	js12	js12	js12
D3 (pinion)	js12	js12	js12
D4	js12	js12	js12
D5	h8	h8	h8
Runout	0,03	0,05	0,07

[a] Or to suit available tooling.

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4.2.13 Drawing specifications for blanks

Values for blank parameters under inspection should be specified on the drawings. Some of these features are the following:

— face angle and, if method 1 is used, back angle;

— outside diameter;

— crown to back, or mounting surface;

— bore or shank diameter.

These latter dimensions are used in place of the face angle distance and back angle distance.

4.3 Assembly

4.3.1 General

The quality that is designed and manufactured into a set of bevel gears can only be achieved by the correct mounting of the gears at assembly. To be correctly mounted, each gear should be located axially at a position that provides the tooth contact pattern and backlash specified.

4.3.2 Correct assembly

It is important that gears be assembled carefully to meet the mounting pattern specifications. Gears assembled with an incorrect mounting can wear excessively, operate noisily, scuff and possibly break.

Generally, the only adjustments the assembler can control are those which axially position the pinion member and wheel member at assembly. In certain designs, the assembler is not provided with means of shimming or other methods for positively locating the axial positions of the members. The assemblies resulting from such designs are affected by maximum tolerance accumulations and, in many cases, do not exhibit a good tooth contact pattern.

When mounting distances are marked on the gears, and when provisions are made for shimming, the assembler should shim to achieve these mounting distances. These adjustments eliminate the effects of axial tolerance accumulations in both the gears and mountings. Shimming cannot correct for shaft angle deviations or offset deviations.

4.3.3 Markings

4.3.3.1 General

Before installing a set of bevel gears, it is necessary to examine and understand all the markings on the parts and on any tags which can be attached (see Figure 16). If no markings appear on the gears, the necessary information should be obtained from design specifications.

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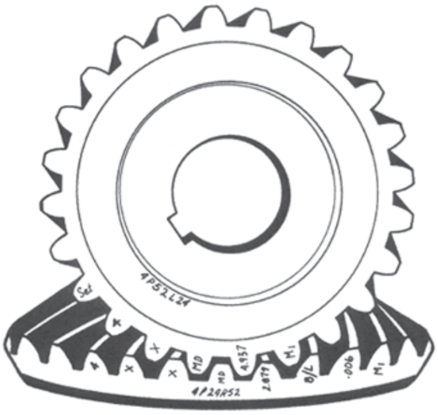

Figure 16 — Typical gear markings

4.3.3.2 Mounting distance

The mounting distance is usually shown as “MD” followed by the actual dimension.

4.3.3.3 Backlash

The minimum amount of total backlash of a pair of bevel gears is measured at the tightest point of mesh with a dial indicator or bevel gear testing machine (see Figure 17). This value is usually marked on the wheel. The amount of backlash is denoted by the markings. Unless otherwise specified, backlash is assumed to be normal backlash and cannot be measured in the plane of rotation.

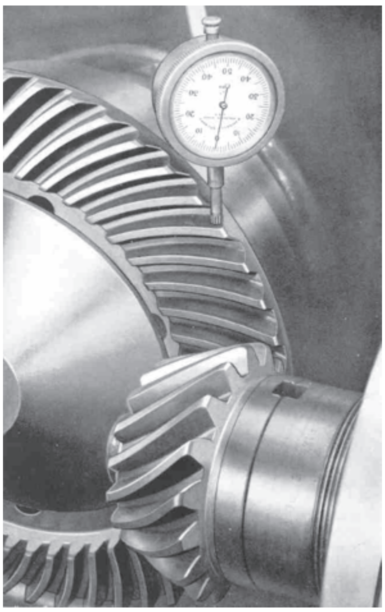

Figure 17 — Measurement of normal backlash

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4.3.3.4 Matched teeth

Some bevel gears are lapped in sets to improve their operation. These gear sets, especially those having tooth counts with a common factor, have marked teeth to assure proper assembly engagement. At assembly, a tooth marked with an “X” on one member should be engaged between two teeth marked with an “X” on the mating member. It is also important when checking backlash to rotate the set of gears to a position where the marked teeth are engaged.

4.3.3.5 Set number

While the teeth of bevel gears are manufactured to close tolerances, slight characteristic tooth form changes do occur from gear to gear, due to tool wear in manufacturing and distortion in heat treating. In most cases, a wheel and pinion are operated under light load in a bevel gear test machine, and sets are selected for a predetermined tooth contact pattern. Therefore, it is important to mark a serial number on each member of a set of gears to assure matched identification; for example, set number 4. Gears which are identified by such a number shall be assembled with the correct mate.

4.3.3.6 Part number

Most gears are identified by a part number. It usually appears in an area away from the marking previously mentioned.

4.3.3.7 Other markings

Other markings can appear which do not affect the assembly procedure. Among these are manufacturer's trademark, material identification, gauge distance, head distance, date of manufacture, and inspector's or operator's symbol. Manufacturer's instructions should be provided to explain the markings.

4.3.4 Positioning bevel gears

4.3.4.1 General

Provisions should be made to aid the assembler in positioning the gears.The desired contact pattern cannot be obtained if the assembler cannot properly position both the pinion and the wheel. Two methods may be used: position by measurement or by contact pattern.

For additional detailed guidance for both methods, see ANSI/AGMA 2008-B01.

4.3.4.2 Positioning by measurement

If the mounting distance has been marked on either or both members, measurement is the preferred method. Direct measurement may involve measuring all the components between the gear's mounting surface and its shimming location. Shims are used to make adjustments in position. This method includes locating the surface for the shims on the housing. The housing dimension can easily be obtained during the machining process. Either the actual dimension or the deviation from the mean can be marked on the housing for use at assembly.

In order to minimize possible accumulation of errors, the least number of measurements necessary to calculate the shims should be made. Gauges can often be designed to reduce the number of measurements required.

Both the pinion and the wheel should be positioned by this method. However, if the wheel's mounting distance is not marked, and the pinion has been positioned by measurement, the wheel's correct axial position may be determined at the point where the proper backlash is measured at the tightest point of mesh between the mating members.

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4.3.4.3 Positioning by contact pattern

In the absence of proper mounting distance marking, the assembler should mark the teeth with gear marking compound and rotate both members in mesh under light load. Adjustments to the axial position of both members are made until the desired contact pattern and backlash are obtained. This method often requires considerable time and experience to correctly interpret the contact patterns.

4.3.5 Backlash measurement

The outer normal backlash, j_{en}, of a pair of bevel gears may be measured with a dial indicator. The stem of the indicator should be mounted perpendicular to the wheel tooth surface at the extreme heel. Backlash is then measured by rotating the wheel back and forth, making certain that the pinion is held motionless (see Figure 18). The outer normal backlash measured at the tightest point of mesh or at the matched teeth should be held within the values in Table 5, if not specified.

Table 5 — Suggested normal backlash tolerance at tightest point of mesh

Outer transverse module, m_{et} mm	Outer normal backlash, j_{en} mm	
	Accuracy grades 2 to 5 according to ISO 17485	Accuracy grades 6 to 11 according to ISO 17485
1,00 to 1,25	0,03 to 0,05	0,05 to 0,08
1,25 to 1,50	0,03 to 0,05	0,05 to 0,10
1,50 to 2,00	0,05 to 0,10	0,08 to 0,13
2,00 to 2,50	0,05 to 0,10	0,08 to 0,13
2,50 to 3,00	0,08 to 0,13	0,10 to 0,20
3,00 to 4,00	0,10 to 0,15	0,13 to 0,25
4,00 to 5,00	0,13 to 0,18	0,15 to 0,33
5,00 to 6,00	0,15 to 0,20	0,20 to 0,41
6,00 to 7,00	0,20 to 0,28	0,25 to 0,46
8,00 to 10,00	0,25 to 0,33	0,33 to 0,56
10,00 to 12,00	0,30 to 0,41	0,41 to 0,66
12,00 to 16,00	0,38 to 0,51	0,51 to 0,81
16,00 to 20,00	0,51 to 0,66	0,69 to 1,07
20,00 to 25,00	0,61 to 0,76	0,81 to 1,17

To calculate outer backlash in the plane of rotation, use Equation (5):

$$j_{et} = \frac{j_{en}}{\cos\alpha_n \cos\beta_e} \tag{5}$$

If the backlash does not fall within the recommended limits, the individual components should be thoroughly evaluated to determine the cause.

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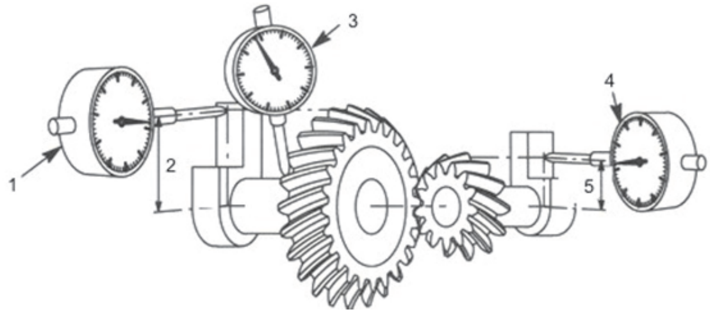

Key

1 transverse backlash
2 outer pitch radius, wheel
3 normal backlash (normal to the tooth surface)
4 transverse backlash
5 outer pitch radius, pinion

Figure 18 — Bevel gear backlash — Normal and transverse

4.3.6 Amount of axial movement for a limited change in backlash

The amount of axial movement for either pinion or wheel necessary to obtain a change in backlash may be determined by the applicable graph in Figure 19, or by the following formulas:

Desired amount of total change in backlash:

$$\Delta j = \Delta j_1 + \Delta j_2 \tag{6}$$

Calculated change in backlash for pinion:

$$\Delta j_1 = \frac{\Delta j \tan\delta_1}{\tan\delta_1 + \tan\delta_2} \tag{7}$$

Calculated change in backlash for wheel:

$$\Delta j_2 = \frac{\Delta j \tan\delta_2}{\tan\delta_1 + \tan\delta_2} \tag{8}$$

Required amount of axial movement of pinion:

$$\Delta\alpha_1 = \frac{\Delta j_1}{2 \tan\alpha_n \sin\delta_1} \tag{9}$$

Required amount of axial movement of wheel:

$$\Delta\alpha_2 = \frac{\Delta j_2}{2 \tan\alpha_n \sin\delta_2} \tag{10}$$

NOTE Equations (6) to (10) are exact for bevel gears, but can also be used for hypoid gears as a first approximation. If there are different generated normal pressure angles on drive side and coast side $\alpha_{nD} \neq \alpha_{nC}$ (see ISO 23509), use $\alpha_n = 0{,}5\,(\alpha_{nD} + \alpha_{nC})$.

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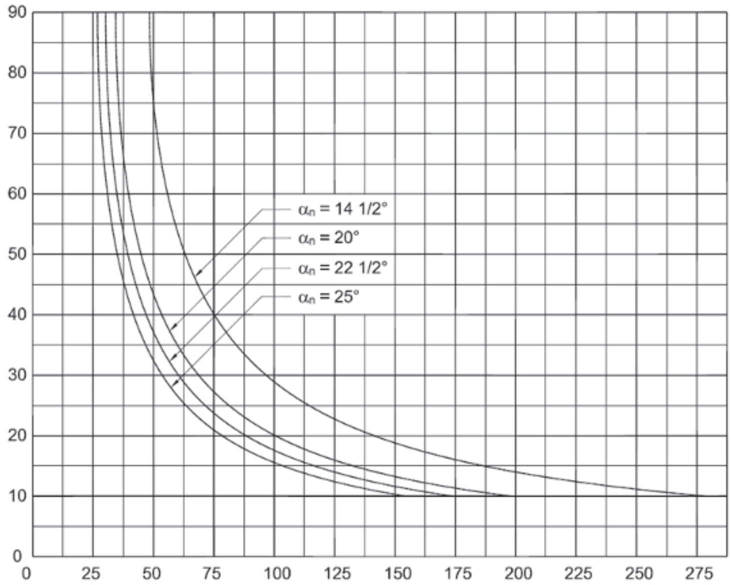

Figure 19 — Influence of axial movement on backlash

For higher ratios, the effect of pinion axial movement on backlash is small. When adjusting backlash for lower ratios, it can be necessary to move both pinion and wheel to maintain acceptable tooth contact. Calculate the amount of axial movement for each member using Equations (6) to (10). If the shaft angle is 90°, the ratio of pinion and wheel mounting distance change is equal to the gear ratio z_2/z_1.

4.3.7 Endplay

If either member of a pair of bevel gears is assembled with allowance for bearing end play and not held to a fixed position, it can be necessary to check for minimum backlash when the floating member is moved axially to its foremost position toward the crossing point.

4.4 Tooth contact pattern

4.4.1 General

The position and the size of the tooth contact pattern is an important contributor to bevel gear quality. Depending on the amount of load applied to bevel gears, deflections occur, and changes appear in the tooth contact pattern. It can be desirable to modify the unloaded contact pattern to allow for stresses which are present under operating conditions.

The location of the contact pattern is directly affected by the relative position of the gear members in assembly. The contact pattern is that portion of the tooth surface which actually makes contact with its mate. It can readily be observed by painting the teeth with a marking compound and running the gears for some revolutions under a light load.

4.4.2 Typical contact patterns

For bevel gears in rigid mountings, typical non-loaded contact patterns are shown in Figure 20 a) and b). It is common practice in industry to evaluate the contact pattern on the wheel flank. Therefore, the contact pattern illustrated in Figure 20 a) and b) relate to those on the wheel flanks only. In hypoid gears, the pattern shape of pinion and wheel can differ considerably.

The size and the location of the non-loaded pattern are influenced by tooth modifications such as lead and profile crowning. For rigid mountings, the unloaded contact pattern is generally in the middle of the flank or

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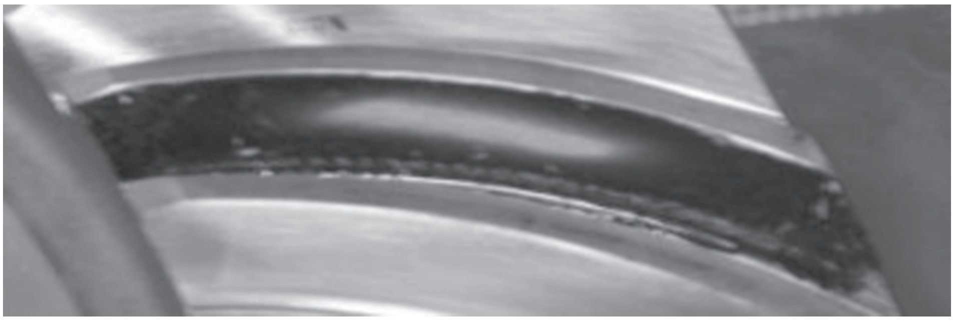
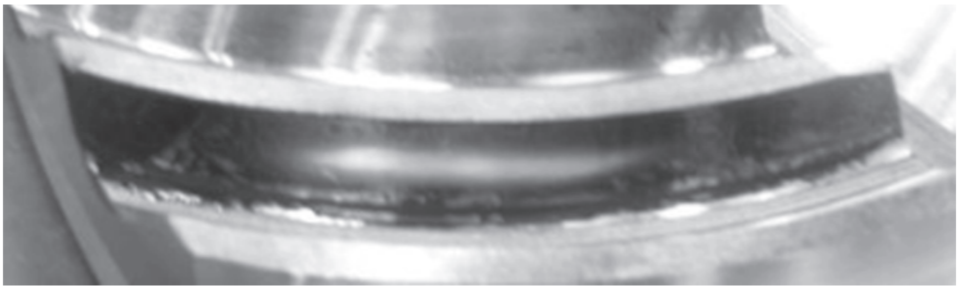
nearer the toe of the tooth than the heel. Consideration should be given to the deflection of shafts, mountings and housing which can affect the location of the contact pattern.



a) Contact pattern — Concave wheel flank



b) Contact pattern — Convex wheel flank

Figure 20 — Typical non-loaded contact patterns

With operating load, the pattern extends. Operating load also leads to deformations of shafts, bearings and housing of the gearbox as well as to displacements of the shafts caused by bearing clearance. This can affect the location of the loaded pattern on the tooth flank. The effect can be different for drive side and coast side flank. If displacements of the mating flanks are considerably large, this should be taken into account by the design. In this case, the non-loaded pattern may not be located in the centre of the flank. The determination of the effects of elastic deformations and displacements on the pattern requires detailed tooth contact analysis software.

Figure 21 shows the calculated desired contact pattern under full load. The contact pattern should have a slight relief at the ends, and along the flank and top of the tooth profile. Generally, the contact pattern should utilize virtually the total length, without having concentration at the ends or tops of the teeth of either member.

Figure 21 — Calculated loaded contact pattern

4.4.3 Need for position control

Control of the localized contact pattern under load after assembly is essential. It contributes to smooth, quiet running bevel gears while in operation. Lack of control can lead to undetected deviations at assembly and can cause the pattern to concentrate dangerously near the toe or heel as well as tip or root of the active flank.

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4.4.4 Deflection tests

The amount and the position of the localized contact pattern should be determined to suit the specific requirements of the pinion and wheel application. If possible, deflection tests should be made on heavily loaded gears, to determine the precise amount and position of the tooth bearing in the manufacturing stage.

4.4.5 Drawing specifications

Upon completion of the deflection test and the resultant final development of the proper tooth bearing, two sketches of the desired contact pattern may be shown on the pinion and wheel drawing. One should show the contact pattern required for manufacturing; the other should show the final pattern required under normal running conditions at assembly.

5 Strength considerations

5.1 Effect of hypoid offset

A set of hypoid gears with a positive pinion offset has a larger pinion diameter and a larger pinion face width compared to bevel gears with the same transmission ratio and the same wheel pitch diameter. This leads to a higher contact ratio and thus to lower contact stresses if the same pinion load is applied. Furthermore, the pinion offset affects the sliding conditions between the mating flanks. The greater the offset value, the higher the sliding velocity in lengthwise direction of the flanks. This has a negative influence not only on the scuffing capacity of the gears but also on their fatigue life.

Due to the higher contact ratio, the tooth root strength of hypoid gears increases with the amount of pinion offset, resulting from the lower tensile stresses of pinion and wheel.

5.2 Effect of cutter radius

For spiral bevel gears and hypoid gears, the cutter radius is one of the very important design items, because it can influence greatly the characteristics of tooth bearing shift caused by misalignments. Cutter radii within the range of R_{m2} to $R_{m2} \cdot \sin \beta_{m2}$ are used (Figure 22). Figure 23 shows the directions of tooth bearing shift caused by the misalignment (ΔP) in the pinion mounting distance, and the misalignment (ΔE) in the offset direction. For further information, see ISO/TR 10064-6. The direction of tooth bearing shift caused by the misalignment (ΔE) is similar in Figure 23 a) and b). Tooth bearing moves toward toe-top on the wheel convex flank and toward heel-top on the wheel concave flank. However the direction of tooth bearing shift caused by the misalignment (ΔP) is different in Figure 23 a) and b). Namely, in Figure 23 a), tooth bearing moves toward heel-top on the wheel convex flank and toward toe-top on the wheel concave flank. On the other hand, in Figure 23 b) tooth bearing moves toward toe-top on the wheel convex side and toward heel-top on the wheel concave side.

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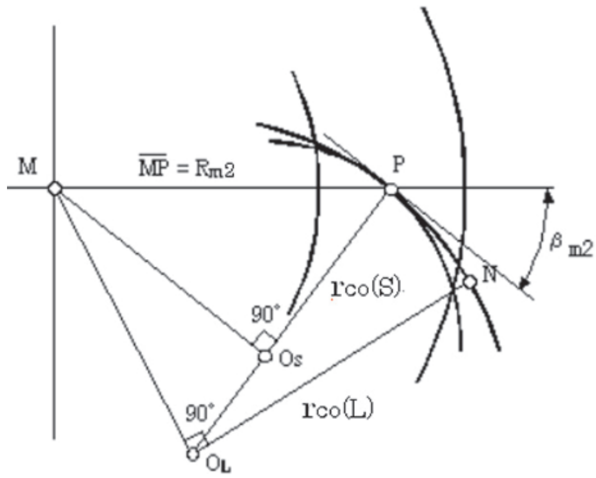
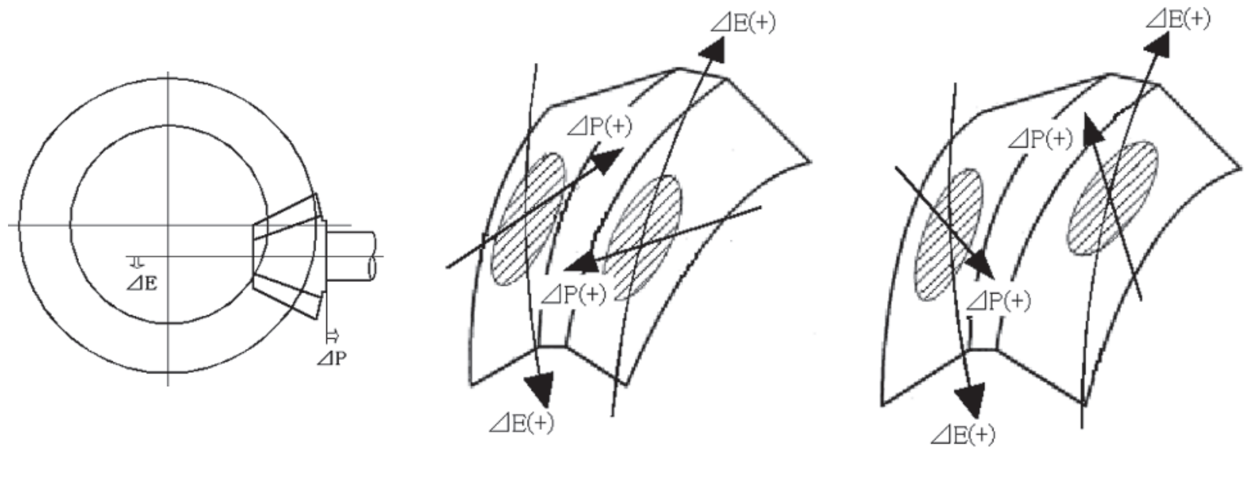


Figure 22 — Size of cutter radius

In the spiral bevel and hypoid gearing, where load is applied on the wheel convex tooth flank, ΔP has a value of plus and ΔE has a value of minus. Therefore, in the case of the gear of the large cutter radius, both of the misalignment elements move the tooth bearing toward heel. On the other hand, in the case of the gear of a small cutter radius, the effects of both misalignment elements are compensated and the amount of tooth bearing shift is decreased. Table 6 shows tooth bearing shift according to the misalignments ($\Delta P = 0{,}25$ mm, $\Delta E = -0{,}25$ mm) caused by the loading.

The tooth bearing of the gear of the large cutter radius reaches at the heel. However, that of the gear of small cutter radius remains at the middle position between centre and heel which is advantageous for the strength of the gear teeth.

Further, some caution is required for gears of small cutter radius, because it is difficult to shift the tooth bearing from heel to toe during the lapping process.



a) Large cutter radius

b) Small cutter radius

Figure 23 — Direction of tooth bearing shift caused by misalignment

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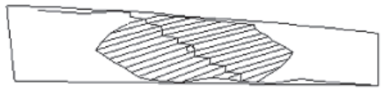
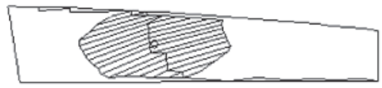
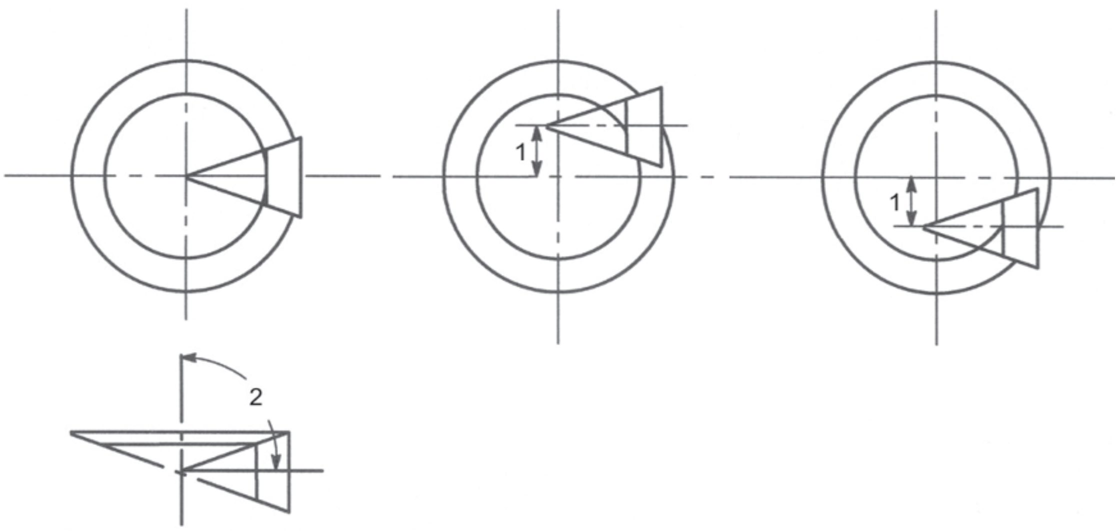
Table 6 — Effect of cutter radius in the tooth bearing shift caused by misalignment

Cutter radius		Tooth bearing
Without misalignments		
With misalignments $\Delta E = -0{,}25$ mm $\Delta P = \ \ 0{,}25$ mm	$r_{c0}/R_{m2} = 1{,}05$	
	$r_{c0}/R_{m2} = 0{,}61$	
Mean cone distance, $R_{m2} = 90{,}5$ mm.		

5.3 Bevel gear mountings

To ensure proper operation of bevel gears, the same care that goes into the design of the blanks and gear elements should be exercised in the design of the mountings.

The pinion and wheel mountings should be designed to give adequate support to the gears for all load conditions to which the gears can be subjected. Each member of spiral bevel and hypoid gear sets should be held against axial movement in both directions. Bevel gears can accommodate reasonable displacements and misalignment without detriment to tooth action. Excessive misalignment reduces the load capacity with consequent danger of surface failure and tooth breakage. Therefore, recommended positional tolerances of axes should be met (see Figure 24 and Table 7).



Key

1 typhoid offset

2 shaft angle with angle tolerance $^{+0° \, 2'}_{-0° \, 0'}$

Figure 24 — Position of axes in bevel gearing

Table 7 — Recommended positional tolerances of axes

Size range	Bevel gear axes should intersect within:	Hypoid offset dimension should be within:
Gears up to 300 mm diameter	±0,03 mm	±0,03 mm
Gears 300 mm to 600 mm diameter	±0,05 mm	±0,05 mm
Gears 600 mm to 900 mm diameter	±0,08 mm	±0,08 mm

The space suggested allowable deflection under highest sustained loads has been determined to be the following:

— the pinion and wheel axes should not separate more than 0,08 mm;

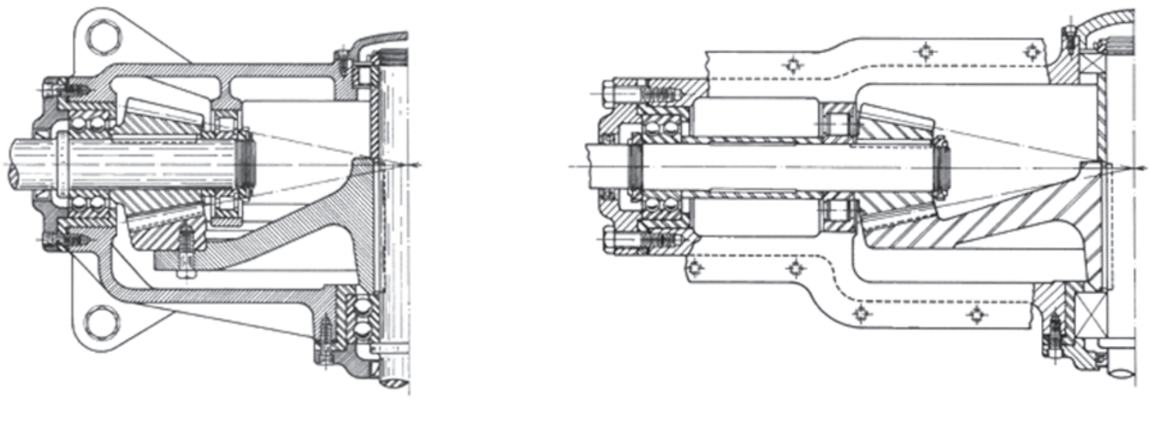
— the pinion should not move axially more than 0,08 mm in either direction;

— the wheel should not move axially more than 0,08 mm in either direction on mitres or near mitres, or more than 0,25 mm away from the pinion for higher ratios.

The above-mentioned limits are for gears of from 150 mm to 380 mm outer diameter. Somewhat narrower deflections are used for smaller diameter gears and somewhat higher deflections are used for larger diameter gears. Somewhat greater deflection values are allowable in a static condition. Bearing end play is not considered.

The preferred design of a bevel gearbox provides straddle mounting for both pinion and wheel, and this arrangement is often used for industrial and other heavily loaded applications. Where it is not feasible to use this arrangement, the member having the higher radial load should be straddle mounted. Overhung mountings can be required due to gearbox space limitations.

Figure 25 shows typical mounting arrangements:



a) Straddle mounting for both members **b) Overhung mounting**

Figure 25 — Typical mounting arrangements

Ideally, the bevel gear mountings should be of good design with adequate rigidity. For the situation where the gear can move axially due to the internal clearance of the bearing, the gear should be located in its normal running position when the load pattern is checked.

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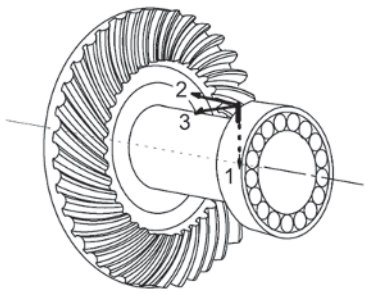
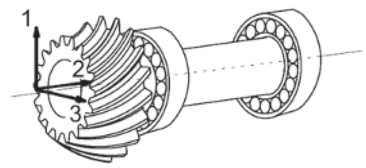
5.4 Direction of forces

The direction of forces is determined by the hand of the spiral and the direction of rotation. The direction of rotation clockwise or anticlockwise (counterclockwise) is determined by viewing from the pinion or wheel back face towards its apex as shown in Figure 26.

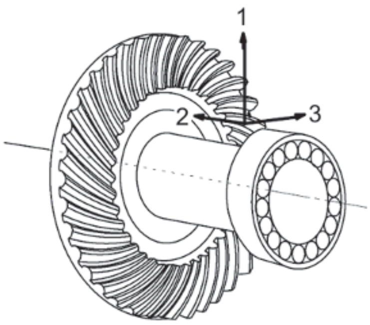
Forces acting on the gear tooth in the axial plane are shown in Figure 26. While the examples shown are typical, they are not all inclusive. Use the equations of ISO 23509:2006, Annex D, to determine the actual force directions.



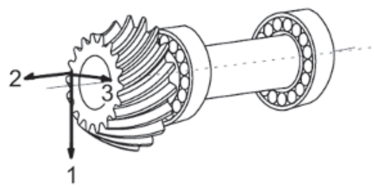
a) Load on the convex flank of the wheel



b) Load on the concave flank of the pinion



c) Load on the concave flank of the wheel



d) Load on the convex flank of the wheel

Key

1 tangential force, F_{mt}

2 axial force, F_{ax}

3 radial force, F_{rad}

Figure 26 — Typical gear tooth forces

6 Efficiency considerations

6.1 Hypoid and bevel gear mesh efficiency

6.1.1 General

The purpose of this clause is to provide a method to estimate hypoid and bevel gear efficiency. These procedures are offered in a Technical Report because there is insufficient experience available for confirmation. Actual thermal losses can differ from those calculated using the following procedures. For further information, see ISO/TR 14179-1.

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6.1.2 Gear efficiency

Friction losses are an important design consideration, particularly in highly loaded gears, in that this energy is dissipated as heat. Design and dynamic effects cannot be fully evaluated except in actual trials. The designer can theoretically evaluate friction losses based on profile sliding, lengthwise sliding and churning.

6.1.3 Efficiency of mesh

The mesh efficiency (taking into account tooth profile and lengthwise sliding and churning), η_{ff}, expressed as a percentage, is calculated using Equation (11):

$$\eta_{\text{ff}} = 100\,[\eta_{\text{ffp}} + \eta_{\text{ffl}} + \eta_{\text{ffc}} - 2{,}0] \tag{11}$$

6.1.4 Profile sliding

The calculation of profile sliding is based on virtual cylindrical gears defined in the mean transverse plane using the following formulas [Equations (12) to (28)] where a unit force is assumed:

$$d_{\text{v1}} = 2R_{\text{m1}} \tan\delta_1 \tag{12}$$

$$d_{\text{v2}} = 2R_{\text{m2}} \tan\delta_2 \tag{13}$$

$$d_{\text{va1}} = d_{\text{v1}} + 2\,h_{\text{am1}} \tag{14}$$

$$d_{\text{va2}} = d_{\text{v2}} + 2\,h_{\text{am2}} \tag{15}$$

$$\beta_{\text{v}} = \left(\beta_{\text{m1}} + \beta_{\text{m2}}\right)/2 \tag{16}$$

$$\alpha_{\text{vt}} = \arctan\left(\tan\alpha_{\text{n}} / \cos\beta_{\text{v}}\right) \tag{17}$$

$$\cos\alpha_{\text{vat1}} = d_{\text{v1}} \cdot \cos\alpha_{\text{vt}} / d_{\text{va1}} \tag{18}$$

$$\Delta\alpha_{\text{t1}} = \alpha_{\text{vat1}} - \alpha_{\text{vt}} \tag{19}$$

$$\cos\alpha_{\text{vat2}} = d_{\text{v2}} \cdot \cos\alpha_{\text{vt}} / d_{\text{va2}} \tag{20}$$

$$\Delta\alpha_{\text{t2}} = \alpha_{\text{vat2}} - \alpha_{\text{vt}} \tag{21}$$

$$T_{\text{i1}} = \left(d_{\text{v1}}/2\right) \cos\alpha_{\text{vt}} \tag{22}$$

$$T_{\text{i2}} = \left(d_{\text{v2}}/2\right) \cos\alpha_{\text{vt}} \tag{23}$$

$$\text{a}_{\text{v}} = \left(d_{\text{v1}} + d_{\text{v2}}\right)/2 \tag{24}$$

$$R_2 = \left[\left(d_{\text{va1}}/2\right)^2 + a_{\text{v}}^{\,2} - 2 \cdot \left(d_{\text{va1}}/2\right) \cdot a_{\text{v}} \cdot \cos\Delta\alpha_{\text{t1}}\right]^{1/2} \tag{25}$$

$$\sin\theta_2 = \left(d_{\text{va1}}/2\right) \sin\Delta\alpha_{\text{t1}} / R_2 \tag{26}$$

$$T_{\text{o2}} = (d_{\text{v2}}/2) \cdot \cos\alpha_{\text{vt}} - \mu_{\text{m}} \cdot \cos\alpha_{\text{vt}} \cdot [h_{\text{am2}} \cdot (d_{\text{va2}}/2) \cdot \sin\Delta\alpha_{\text{t2}} + h_{\text{am1}} \cdot R_2 \sin\theta_2] / [2 \cdot (h_{\text{am1}} + h_{\text{am2}})] \tag{27}$$

$$\eta_{\text{ffp}} = T_{\text{o2}} / T_{\text{i1}} = 1 - \mu_{\text{m}} \cdot [h_{\text{am2}} \cdot (d_{\text{va2}}/2) \cdot \sin\Delta\alpha_{\text{t2}} + h_{\text{am1}} \cdot R_2 \cdot \sin\theta_2] / [d_{\text{v2}} \cdot (h_{\text{am2}} + h_{\text{am1}})] \tag{28}$$

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6.1.5 Hypoid lengthwise sliding

Calculation is made in the mean pitch plane with the following formulas [Equations (29) to (36)] again assuming a unit normal force:

$$d_{v1} = 2\, R_{m1} \cdot \tan \delta_1 \quad (29)$$

$$d_{v2} = 2\, R_{m2} \cdot \tan \delta_2 \quad (30)$$

$$\tan \varphi = \mu_m \,/\, \cos \alpha_n \quad (31)$$

$$T_{i1} = (d_{v1} \,/\, 2) \cdot \cos(\beta_{m1} - \varphi)(\cos \alpha_n \,/\, \cos \varphi) \quad (32)$$

$$T_{i2} = (d_{v2} \,/ 2) \cdot \cos \beta_{m2} \cdot T_{i1} \,/\, [(d_{v1} \,/\, 2) \cdot \cos \beta_{m1}] \quad (33)$$

$$T_{i2} = (d_{v2} \,/\, 2) \cdot \cos \beta_{m2} \cdot \cos(\beta_{m1} - \varphi) \,/\, \cos \beta_{m1} (\cos \alpha_n \,/\, \cos \varphi) \quad (34)$$

$$T_{o2} = (d_{v2} \,/ 2) \cdot \cos(\beta_{m2} - \varphi)(\cos \alpha_n \,/\, \cos \varphi) \quad (35)$$

$$\eta_{ffl} = T_{o2} \,/\, T_{i2} = (1 + \mu_m \tan \beta_{m2} \,/\, \cos \alpha_n) \,/\, (1 + \mu_m \tan \beta_{m1} \,/\, \cos \alpha_n) \quad (36)$$

For bevel gears without hypoid offset, $\eta_{ffl} = 1{,}0$.

6.1.6 Coefficient of friction

If the pitch line velocity, v_{et}, is $2 < v_{et} < 25$ m/s and the K-factor is 1,4 N/mm$^2 < K <$ 14 N/mm^2, then μ_m can be estimated by Equation (37). Outside these limits, the values for μ_m should be determined from experience.

For bevel gearing, the pitch line velocity is calculated at the larger end of the tooth.

Load intensity, K, can be calculated from Equation (38) The exponents, j, g and h modify the viscosity, υ, the load intensity, K, and the tangential pitch line velocity, v_{et}, respectively. Values to be used for the exponents, j, g and h, and the constant, C_1, are the following:

$j = -0{,}223$

$g = -0{,}40$

$h = 0{,}70$

$C_1 = 3{,}239$

$$\mu_m = \frac{\upsilon^j \times K^g}{C_1 \times {v_{et}}^h} \quad (37)$$

$$K = \frac{1\,000\, T_1 \times (z_1 + z_2)}{2 b_2 \times (d_{v1} / 2)^2\, z_2} \quad (38)$$

6.1.7 Churning efficiency

The equations for gear windage and churning power loss that follow are derived from the equations that appear in ISO/TR 14179-1:2001, 7.9. They have been modified to include the oil viscosity, υ, the gear dip factor, f_g, and an arrangement constant, A_g. In addition, the exponent for the diameter, D, was adjusted. The use of the dimensions and tooth geometry based on the large end of the teeth results in conservative values.

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Before calculating gear windage and friction losses, the gear dip factor, f_g, should be determined. This factor is based on the amount of dip that the element has in the oil. When the element does not dip in the oil, $f_g = 0$; when the element is fully submerged in the oil, $f_g = 1$. When the element is partly submerged in the oil, linearly interpolate between $f_g = 0$ and $f_g = 1$. For example, for a gear that has the oil level at the centre line of its shaft, $f_g = 0{,}50$. Use a value of 0,2 for the arrangement constant, A_g, in Equations (40), (41) and (42).

The power loss equation for the tooth surface calls for a roughness factor, R_f. ISO/TR 14179-1:2001, 7.9 provides some values based on tooth sizes. Equation (39) is a reasonable approximation of the values from Dudley.

$$R_f = 7{,}93 - \frac{4{,}648}{m_t} \tag{39}$$

Gear windage and churning losses encompass three types of loss. For those losses associated with a smooth outside diameter, such as the OD of a shaft, use Equation (40). For those losses associated with the smooth sides of a disc, such as the faces of a gear, use Equation (41). Equation (41) includes both sides of the gear; therefore, the value should not be doubled. For those losses associated with the tooth surfaces, such as the OD of a gear or pinion, use Equation (42).

For smooth outside diameters:

$$P_{GWi} = \frac{7{,}37 f_g \upsilon \ n^3 D^{4{,}7} L}{A_g 10^{26}} \tag{40}$$

For smooth sides:

$$P_{GWi} = \frac{1{,}474 f_g \upsilon \ n^3 D^{5{,}7}}{A_g 10^{26}} \tag{41}$$

For tooth surfaces:

$$P_{GWi} = \frac{7{,}37 f_g \upsilon \ n^3 D^{4{,}7} b_w \left(\frac{R_f}{\sqrt{\tan \beta_m}} \right)}{A_g 10^{26}} \tag{42}$$

NOTE If β_m is less than 10°, use 10°.

After calculating the individual elements for each shaft assembly, they should be added together for the total loss. For example, an output shaft assembly would use Equation (40) for the OD of the shaft outside of the gear between the bearings, Equation (41) for the smooth sides of the gear and Equation (42) for the tooth surfaces. The sum of power losses is then expressed as efficiency of churning:

$$\eta_{ffc} = 1 - \sum P_{GWi} \ / \ P \tag{43}$$

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6.2 Lubrication

6.2.1 Principles for lubrication

The principles suggested for the lubrication of bevel gears are similar to those followed in the lubrication for spur and helical gears. Lubrication provides a dual function:

— to prevent metal to metal contact;

— to carry away the heat generated by friction in tooth engagement.

To fulfill these functions, each tooth surface should carry a film of lubricant when it enters into engagement with a mating tooth, and there should be sufficient volume of lubricant to absorb and dissipate the heat generated by friction, without excessive temperature rise.

6.2.2 Selection criteria of lubricants

Lubricant selection should consider other factors in addition to gear design.

6.2.2.1 Environment

The operating environment should be carefully considered when selecting the lubricant system of a gearbox. The ambient temperature is the most common consideration. Contamination is also a common factor, but is frequently overlooked. Some applications, such as mining, paper mills, textile mills and printing presses produce large amounts of abrasive dust. This material can seriously affect the operation of gears if it contaminates the lubricant. Replacing a conventional air breather vent with an air filter, providing proper gaskets and seals or the use of a circulating oil system with proper filtration are common solutions against contamination.

6.2.2.2 Maintenance

The frequency and type of maintenance influence the number of lubrication options available to the gearbox designer. In situations where lubricants can be easily checked and replaced, the selections can be numerous. Mineral oils and greases are common selections. Frequently, the gears are placed in an application where maintenance is difficult or impossible, such as sealed for life consumer appliances. In this case, the designer may choose synthetic oils, greases or self-lubricating materials. Frequently, the designer has to increase the amount of lubricant and component sizes to compensate for the wear they experience due to the less than desirable lubrication conditions.

6.2.2.3 Application

The application of the gear system may dictate the type of lubricants available to the designer. Certain industries have a predetermined selection of lubricant. A common example would be a food processing plant. The lubricant selection may be limited to products that would not be harmful to the food processed, should contamination occur. Medical processes have similar restrictions. Aircraft and military lubricants should be selected from those that pass rigorous tests and qualifications. Spacecraft and satellite applications cannot contain materials that produce gases in a vacuum.

6.2.2.4 Internal components

Bevel gears depend on bearings, seals, clutches and sometimes other components to perform their intended functions. The selection of lubricants is frequently a compromise when considering all components. For example, gears typically require high-viscosity oil, bearings prefer low-viscosity oil. Some components can tolerate wider variation of lubricants than others. Changes to gear material, heat treatment, surface finish or geometry can be necessary to find a lubricant that best fits all components. A typical example is the automotive engine. The lubricant is tailored to the combustion part of the engine, yet should lubricate the gears and bearings as well. It can be necessary for the gear designer to select the gear material, heat treatment, surface finish or geometry specifically for the system-required lubricant.

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6.2.2.5 Cooling requirements

One of the primary functions of a lubricant is to carry away heat. Many drives require oil to properly remove the heat generated during operation. Several methods can be used to remove heat. The simplest method is allowing the gearbox housing to dissipate the heat. This requires that the housing be large enough to allow the heat to dissipate at least as fast as it is produced. Frequently, external cooling is required. This may be accomplished by the use of fans or a heat exchanger. The determination of the heat flow in a gearbox is beyond the scope of this Technical Report.

6.2.2.6 Efficiency and load-carrying capacity

Lubricants have a high influence on the efficiency and load-carrying capacity of gears. Therefore, the lubricant should be chosen subject to the operating conditions.

Depending on the lubrication conditions different base oil types and additives can show very different friction behaviour: At no-load, the power losses of a pair of gears are mainly influenced by the viscosity. Under load, the base oil type (mineral oil, poly-alpha-olefin, ester, polyglycol, etc.) influences the losses at elasto-hydrodynamic-lubrication, the additives in the mixed and boundary lubrication. Also, the gear failures affecting the flank surfaces, e.g. scuffing, pitting, micropitting and wear, are influenced by the used type of lubricant as well as the type and concentration of the additives, respectively.

Exact values for the influences on efficiency and load-carrying capacity can only be derived from experimental investigations.

6.2.3 Types of lubricants

6.2.3.1 General

Numerous types of lubricants are available to the designer. Oil is the most versatile and popular. Grease is the second most popular. Other materials can also be used.

6.2.3.2 Oil

6.2.3.2.1 General

Oil is by far the most frequently used lubricant for gears. It can be produced from natural hydrocarbons or by synthetics. The properties of lubricants can be changed by changing its viscosity or by the addition of chemical additives. The proper selection of oil varies drastically with application. There is no universal gear oil. Oil selected for one application can be disastrous in another.

6.2.3.2.2 Viscosity

Viscosity is the most significant property requiring specification. Usually, viscosity is selected for the gear mesh having the heaviest load and lowest speed among those served by the same lubricant.

The viscosity grade at 40 °C may be selected by the following:

$$\upsilon_K = \frac{35{,}56}{v_{\mathrm{et}}^{\;0{,}5}} \tag{44}$$

NOTE If ν_{et} is less than 2,5 m/s, use 2,5 m/s.

Equation (44) provides a guide. While the nearest viscosity grade given in ISO/TR 18792 is usually selected, satisfactory performance may be obtained with a viscosity level one grade higher or lower than indicated.

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6.2.3.2.3 Criteria for oil selection

When selecting an oil for a given application the following items should be considered:

- high and low temperature;
- viscosity index improver;
- pour depressants;
- oxidation inhibitor;
- corrosive inhibitors;
- antifoam additives;
- extreme pressure additives.

For additional information see ISO/TR 18792.

6.2.3.3 Grease

Grease is a mixture of base oil and a thickening agent. The thickening agent, usually a metallic soap, is used to control consistency. The consistency can vary from a solid to a thin semi-fluid. Since the base oil provides all lubrication, the discussion regarding oil also applies to grease.

Grease is frequently selected as a lubricant in an attempt to avoid leakage problems. Since grease does not flow as readily as oil, seals are not as critical. The oil does tend to separate from the thickener with time, so care should be taken if minor leaks cannot be tolerated.

Grease is also frequently used as an initial lube for self-lubricating gears.

At high speeds, gears cut a channel through a standing reservoir of grease and throw off any remaining grease from the gear. The semi-fluid consistency prevents the grease from flowing back to fill the channel. Poor heat transfer characteristics combine with the flow limitations to reduce the load capacity of the gearbox. For these reasons, the use of grease should be restricted to lower speed applications.

6.2.3.4 Dry lubricants

Dry lubricants refer to coatings applied to the gear tooth surface and are not intended to be replenished. They can include molybdenum disulfide, graphite or organic materials, such as polytetrafluoro-ethylene. They essentially provide an anti-wear coating between the teeth. They cannot provide cooling, which is often the primary lubricant requirement. Because of this, dry lubricants are normally used for lightly loaded applications.

6.2.3.5 Self-lubrication

The use of polymer compounds (plastic gears) has become quite popular. They include some limited self-lubrication and act in a similar way to dry lubricants. They also deflect more than metal gears. Even though the material is self-lubricating, an initial coating of grease or oil greatly improves the break-in of self-lubricating materials. Care should be exercised in the selection of lubricants applied to these gear materials to prevent undesirable chemical reactions.

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6.2.4 Application of lubricant

6.2.4.1 General

Regardless of the type of lubricant selected, proper operation requires that an adequate quantity be applied. The methods used to ensure the application of that quantity vary with the type of lubricant. For more detailed information about the lubrication of industrial gear drives, see ISO/TR 18792.

6.2.4.2 Quantity required

The amount of lubricant is dependent on several factors. When oil is the selected lubricant a minimum of 0,08 l/min per mm face width is suggested regardless of the power transmitted.

6.2.4.3 Method of application

Splash lubrication and pressure jets are the typical means of applying oil. Splash lubrication is the process of allowing a rotating component, usually a gear, to dip into the oil. Centrifugal forces fling the oil around inside the gearbox. Oil can also cling to the gear, carrying it into mesh. Increasing the oil level to bring more oil into contact with the rotating component can be detrimental. The additional oil can lead to churning and hence increased temperatures, foaming and a loss of efficiency. The use of splash lubrication is usually confined to slow speed drives and can require some experimentation for proper operation.

In pressure feed systems, oil is forced through orifices, called jets, to the gear teeth near the meshing point. The jets should be positioned with at least one jet per 25 mm of face width. High-speed gears can act as air pumps and deflect the oil coming from the jets. The position of the jets and pressure should be adjusted to assure that the mesh receives oil. Pressures at the jet can range from 0,17 N/mm^2 to 0,34 N/mm^2 depending on the flow rate and the gear speed. Table 8 is frequently used for the location of oil jets in pressure feed systems.

Table 8 — Typical oil jet location

Pitch line velocity m/s	**Jet location**	**Comment**
15	None	Properly designed splash is adequate
15 to 25	Into mesh	Lubrication is primary; cooling is secondary; splash with adequate baffles and channels can also work
25 to 60	Out-of-mesh or into mesh	Cooling is the primary function; sufficient oil adheres to the teeth which lubricates the contact

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